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THE THERMAL EVALUATION OF AIR-COOLED ELECTRONIC EQUIPMENT

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and
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THE OHIO STATE UNIVERSITY RESEARCH FOUNDATION

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WRIGHT AIR DEVELOPMENT CENTER

HEADQUARTERS

Wright Air Development Center

WRIGHT-PATTERSON AIR FORCE BASE, OHIO

IN REPLY ADDRESS BOTH COMMUNICATION AND ENVELOPE TO COMMANDING GENERAL, WRIGHT AIR DEVELOPMENT CENTER, ATTENTION FOLLOWING OFFICE SYMBOL:
WCLCD3

WCLCD3/YJ/ep
1 October 1952

TO: Recipients of Technical Report

1. The problem of cooling electronic equipment in high altitude, high speed airframes is an increasingly serious one which must be successfully solved if the goals of the Air Force are to be met.

2. Analysis of the overall situation shows that it breaks into two major parts, the ultimate and the interior cooling phases. The ultimate phase is intimately linked with the characteristics and tactical operation of the airframe; and while of great interest and importance to the electronic equipment designer, its solution is not primarily his problem. USAF Technical Report 6493 covers the initial work in this phase and continued study is being carried out under Contract AF 33(616)-147.

3. The interior cooling phase, however, is fully the responsibility of the electronics engineer. The objective of this report is to assist him in meeting this responsibility. Here, for the first time, are presented means and methods of performing thermal evaluation of electronic equipments in a systematic and engineering fashion. Insofar as possible, the report has been written in the language of the electronics man. The fundamentals of thermodynamics, heat transfer and flow of fluids have been gathered together for easy references. All methods have been illustrated with specific examples involving electronic equipment.

4. The usability and effectiveness of the evaluation methods presented can only be determined by continued use. This Center therefore earnestly solicits reports from electronic engineers on their experiences in using these methods as well as criticisms and suggestions which will permit their refinement and simplification. Please address all correspondence to Commanding General, Wright Air Development Center, ATTN: WCLCD3, Wright-Patterson Air Force Base, Ohio.

FOR THE COMMANDING GENERAL:

Richard S Carter

RICHARD S. CARTER
Colonel, USAF
Chief, Components and
Systems Laboratory
Directorate of Laboratories

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The Ohio State University Research Foundation

September 1952

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Wright Air Development Center
Air Research and Development Command
United States Air Force
Wright-Patterson Air Force Base Ohio

FOREWORD

This report was prepared in the Mechanical Engineering Department of The Ohio State University. The work was performed between October 1950 and August 1952 as one phase of the activities under Contract W33-038ac-14987 with The Ohio State University Research Foundation. It was administered under the direction of the Components and Systems Laboratory, Weapons Components Division, Wright Air Development Center, Major Y. Jacobs acting as project engineer.

This report is the second Air Force Technical Report to be prepared under the contract. It was preceded by Air Force Technical Report No. 6493, <u>Methods for the Ultimate Dissipation of Heat Originating with Airborne Electronic Equipment</u>, April 1951, and a number of reports dealing with data and methods for the internal design of airborne electronic equipment, utilizing various means for heat dissipation from individual components.

This report is intended as a manual to aid electronic designers and application engineers in the thermal evaluation of existing airborne electronic equipment by experimental and analytical methods. The treatment is limited to the performance of individual units under specified environmental conditions. A future supplement to this report will be concerned with the evaluation of systems installations and modifications of units for operation in more severe thermal environments. Other reports will deal with the evaluation of air- and liquid-cooling techniques on test units of high power density and the presentation of associated design methods.

The authors wish to express their thanks to their coworkers on the project staff at The Ohio State University for contributions to various parts of this report, in particular, to J. R. Barton, L. S. Han, and R. H. Lynch, who were responsible for much of the preliminary work and the preparation of several of the illustrative examples presented.

ABSTRACT

Air-cooled electronic equipments designed for airborne application are classified according to their cooling methods. Basic thermal test methods and techniques of making the necessary measurements are described with emphasis on bench testing. Calculation methods for the prediction of altitude performance from bench test data are presented. The analysis of altitude chamber test data and calculation methods for their correction are discussed for equipments which cannot be evaluated by bench tests only, and for other equipments for which certain data may be conveniently secured by altitude chamber test. Methods applicable to various types of equipment for evaluation of non-steady state operation are covered in detail. One chapter is devoted to the theory, performance, evaluation and selection of cooling blowers. Throughout the report, charts are presented which allow graphical means to be utilized to a great extent for purposes of analysis. All methods are illustrated by means of examples listed in the Contents. Physical property data, air flow theory, blower test methods and details of experimental apparatus are presented in the appendices.

PUBLICATION REVIEW

FOR THE COMMANDING GENERAL:

Richard S. Carter

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Colonel, USAF
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CHAPTER I

INTRODUCTION

The purpose of this report is to present methods and procedures which will aid electronic designers and application engineers in determining and evaluating the thermal operating characteristics of electronic equipment for any condition of air temperature, altitude, and installation environment. Further, it should enable the electronic equipment engineer to procure the necessary basic data and information from which personnel more expert in the field of heat transmission could analyze the cooling problems and make recommendations for the design or application of effective cooling systems for a particular electronic equipment.

Because of the preponderance of the application of air cooling to present airborne electronic equipment, this report is limited to the study of equipments which employ air cooling. Although radiation may contribute substantially to the total heat dissipation of a unit, it remains unmentioned in the classification of the unit for its cooling method. The cooling method of the unit is generally described with specific reference to the manner in which air convection around, within, or through the unit takes place. This report is concerned with evaluation of (1) pressurized units cooled externally by either free or forced convection applied to the case or a special heat exchange section, with internal circulation by natural or forced means; (2) units which are vented to the compartment, with components cooled by natural convection; and (3) units in which compartment air forced directly over the component surfaces is used for cooling.

The quantity of heat to be dissipated from a component such as a tube, a resistor, or a transformer operating at a given load, is fixed regardless of the manner in which it is cooled. The use of a more effective cooling method lowers the surface temperature of a component, but the basic internal heat transfer processes within the component remain unaltered. Therefore, internal temperatures are also reduced since the total heat dissipation is fixed. However, the temperature variation of the internal structure with changes in envelope temperature may differ greatly among different types of components. The plate structure of a vacuum tube transmitting heat to the tube's envelope principally by radiation undergoes a much smaller temperature change than the envelope because the rate of heat transfer is a function of the difference of the fourth-power of the absolute temperatures. In contrast, the innermost windings on a transformer may change in temperature almost as much as the surface when the cooling conditions are changed. This occurs because the transformer's losses are transmitted to the surface principally by conduction which is almost directly proportional to the temperature difference between the core and the surface. If a low-loss component, such as a condenser, were mounted in such a way that it received heat by radiation from adjacent high temperature components, both its surface and internal temperatures would rise until the heat gained by radiation was offset by the heat lost from the surface by convection and conduction. Its internal temperature would approach the average surface temperature. Regardless of the type of component, the surface temperature furnishes the best indication of the internal thermal conditions which determine whether or not a component will fulfill its electronic design functions.

External influences which affect component surface temperatures cannot be defined completely by mere physical properties such as ambient temperature and pressure, since the degree of motion of the surrounding air as well as the relative position, surface temperatures and radiation characteristics of adjacent components and surfaces also greatly affect surface temperature. Due to the complexities caused by these variables, extensive tests and detailed analyses are necessary to predict the surface temperatures of components installed in electronic units which are to be operated under various air temperature, altitude and environmental conditions.

Component rating data are not always available based on maximum surface temperature. For non-heat-generating components the limiting ambient temperature for which the component was designed could be taken as the limiting surface temperature. For best thermal evaluation of electronic equipment, every attempt should be made to determine the limiting surface temperature of heat generating components from manufacturers' data or specifications. Otherwise, since surface temperatures are most indicative of the thermal operation of a component, it may be necessary to determine limiting surface temperatures from laboratory tests of individual components.

The principal objectives of the thermal evaluation of an electronic equipment are the determinations of (1) its operational limitations, and (2) the effectiveness of its thermal design. The operational limitations of an electronic unit are defined by all combinations of compartment air temperature, altitude, and other environmental conditions at which the unit becomes electronically inoperative because the maximum allowable surface temperature of one or more components has been exceeded. If a single variable of a set of conditions defining the operational limit of the equipment is so altered that it tends to increase the temperature rise required for heat dissipation, and if the operational temperature limit is not to be exceeded, the effect must be compensated by a change of one or more of the other variables which would tend to decrease the temperature rise required for heat dissipation from the equipment.

Effectiveness of thermal design of a unit for specified operating conditions is expressed by the degree to which the surface temperature of each heat producing component approaches its allowable limit. The optimum in thermal effectiveness for a unit cooled by natural convection would require that all heat producing components operate at their limiting temperatures. This is necessary to achieve a unit of absolute minimum size. For a unit of given size cooled by forced convection the optimum operating condition would result in the least possible expenditure of energy for cooling purposes. This requires that all heat producing components operate at their limiting temperatures.

Results of the thermal evaluation must indicate what modifications, if any, are necessary to adjust the operational limit of the equipment to specified conditions of installation and aircraft operation, or what modifications may be feasible to improve the effectiveness of the thermal design so as to achieve for the equipment the more general objective of minimum size, weight, and cooling power requirement.

In addition to this introductory first chapter, this report consists of seven other chapters and six appendices. In Chapter II the classification of electronic equipment is discussed. An attempt is made to cover the most prevalent types of equipments from the standpoint of cooling design and to group them into common categories which would require the same experimental and analytical treatment for thermal evaluation. The chapter also contains the classification and brief description of the three basic thermal test methods which are applicable to the evaluation of electronic equipment. The types of measurements necessary for the experimental phases of thermal evaluation are discussed in Chapter III. Techniques in general and details of application to specific components are covered. Commercial and laboratory instruments suitable for thermal evaluation tests are described and their operation is discussed. The details of test procedures applicable to equipment designed to operate under steady-state conditions are described in Chapter IV. General procedures common to the study of all types of equipment and special procedures applicable to specific classifications of equipment are covered separately. In the description of procedures, reference is made to the methods of measurements described in the previous chapter. In Chapter V, the reader is introduced to the subject of blowers applicable to electronic equipment. The chapter is intended to serve as an introduction to the discussion of analytical evaluation methods in the following chapters. Because of the great importance of blowers for cooling of airborne electronic equipment, an understanding of the principles and methods presented in Chapter V is essential for use in the analytical evaluation methods. Graphical methods for the performance analysis, design and selection of blowers are contained in this chapter. The specific analytical methods used to derive thermal performance under operational conditions from bench test data are discussed in Chapter VI. The methods are intended to be applicable to the prediction of thermal conditions in flights of sufficient duration to permit the equipment to operate in a steady thermal state. The various methods of analysis are classified in accordance with the units to which they are applicable. The methods described in Chapter VI are supplemented by those contained in the following Chapter VII. The latter methods are based on altitude chamber test data and are concerned with characteristics of certain equipments which cannot be evaluated satisfactorily by analysis based on bench test data only. While Chapter VII, like Chapter VI, deals with the analysis of steady-state operation, Chapter VIII contains methods for test and analysis of equipments which are to be operated under conditions of unsteady thermal state. The methods may apply to short-range installations of relatively appreciable thermal capacity and/or inadequate cooling capacity, and to installations exposed to external heat gain under conditions of high-speed flight.

A future supplement to this report will contain Chapters IX and X. In addition to the thermal evaluation methods applicable to individual units, as described in Chapters VI to VIII, an attempt will be made in Chapter IX to outline procedures applicable to the evaluation of system installations and to provide guide lines for the analysis of the thermal interaction of individual units. In Chapter X, modifications applicable to the extension of the operational limits and improvement of thermal effectiveness of various types of equipments will be discussed. Principally major modifications will be treated which would involve more extensive redesign for cooling but would not necessarily require a change of the electronic design in the sense of rearranging components or modifying chassis configurations.

The appendices to this report are intended to provide background information and reference data. Appendix I contains reference data on the properties of air required for cooling analysis. Appendix II contains a discussion on the theory of air flow and the energy transformations involved and should aid in a better understanding of forced-air cooling. Appendix III is an introduction to the measurement of pressure by various means and is intended to convey information on characteristics of manometric devices. In Appendix IV, an apparatus for the measurement of air flow to be used in the laboratory in connection with test and development of forced-air cooled electronic equipment is described in detail and data required for the construction of its component parts are given. In Appendix V, test methods are described for the determination of operating characteristics of blowers furnished with electronic equipment. Appendix VI is a reference sheet containing a consolidated nomenclature covering all analytical subject matter treated in this report.

CHAPTER II

CLASSIFICATION OF AIR-COOLED ELECTRONIC EQUIPMENT AND TEST METHODS

Airborne electronic equipment has been designed in a variety of modifications which employ heat dissipation to the atmosphere of the compartment in which it is installed and to the aircraft structure defining the compartment in the vicinity of the equipment. These electronic equipments may be classified into two major groups which are distinguished from each other by the pressure of the atmosphere in which the electronic components are operating. The group classified as vented equipment embraces all units in which the electronic components operate at the air pressure of the installation compartment. This pressure decreases appreciably with increased operational altitude, unless the installation compartment is a pressurized cabin. The second group, classified as pressurized or sealed equipment, embraces all units in which the electronic components operate in an atmosphere of essentially constant pressure, produced by hermetical sealing or pressurization of the case. Pressurized and sealed units have found extensive application more recently in the design of equipments for high-altitude operation. Increased operational altitude, miniaturization, and thermal limitations are responsible for the decreasing use of vented equipment. The thermal evaluation of either type of equipment can be made on basis of bench tests, altitude chamber tests, or flight tests. The characteristics of the various types of equipments and their modifications, and features of test methods are discussed in this chapter.

Pressurized Equipment

Compared to vented equipment, the thermal study of pressurized equipment and its modifications is much simpler. The outstanding characteristic of pressurized equipment is that in it the mechanism of heat transfer of the electronic components is independent of altitude conditions and is only dependent on the control of the temperature of the equipment's outer heat exchange surfaces. Like in any electronic equipment, heat transfer from components within pressurized equipment takes place by radiation, conduction, and convection. Radiant heat exchange occurs between surfaces defined by the components, the chassis of the equipment, and the equipment case. Conduction of heat from components takes place through the chassis to the case or other surfaces at lower temperature. Convective heat transfer exists between components and the internal atmosphere of the equipment. Circulation by virtue of temperature differences in adjacent sections causes heat transfer by natural or free convection. The action of a mechanical device, such as a blower, causes heat transfer by forced convection. The combination of the different modes of heat transfer within pressurized equipment results in an overall internal heat transfer coefficient which is dependent on external conditions only to the extent to which they affect the surface temperature of the case or other external heat exchange surfaces.

The shape of the case has little influence on the processes of internal heat transfer. However, it does affect the processes of external heat transfer from the case, particularly if they are based on free convection and radiation. Therefore it is significant to note that pressurized equipment is usually enclosed in cylindrical cases of ellipsoidal heads.

As pointed out above, all pressurized equipments are essentially similar in regard to their processes of internal heat transfer. However, they do differ substantially in manner of external cooling. Therefore, they are classified on that basis in the following.

1. Case Cooled by Free Convection and Radiation

Equipment of this type which depends on the natural heat dissipative capacity of the outer case to maintain internal temperature within acceptable limits can usually be designed only for moderate heat loads per unit volume. This may mean a heat load of 100 to 300 watts per cubic foot, depending on the size, i.e. surface-to-volume ratio, of the unit and the operational conditions. Heat dissipation from the case of such units is not only affected by the environmental air temperature and pressure, but is also affected by characteristics of surrounding surfaces which determine radiant heat exchange. Within the limitation of these natural modes of heat transfer, maximum heat dissipation from the case can be obtained only if its surfaces are made to reach the highest temperature. Therefore, units are often built with an internal blower which serves to increase air agitation, thus reducing the temperature difference between the surface of the components and the case. Frequently, the blower serves principally for internal heat dissipation from one or several high-output components and helps to distribute the total thermal load more uniformly by preventing stratification. However, if the temperature level of most components is similar and overall size limitations are so liberal that great congestion can be avoided, free convective heat transfer, combined with conduction and radiation, can be relied upon to transmit the heat from the components to the case.

At this point, it should be mentioned that the above principles of design for this type of unit, as well as the principles of design for other types of units described in the following sections, have been more or less recognized by electronic designers, using cut-and-try methods of natural elimination. Consequently, the designs they have produced may leave much to be desired from the standpoint of cooling effectiveness. Improvements must and will be made, however, only by increasing rationalization of cooling design resulting in arrangements of electronic components derived from application of basic heat transfer data.

Since the total heat dissipation of any electronic unit operating at a given rating is constant, and since the internal heat transfer coefficient of a pressurized unit is also essentially constant, because of fixed convective and conductive conditions, the temperature difference between the internal air of the unit and the surface of the case is practically independent of the case temperature. Therefore, the actual internal air temperature, as well as the surface temperatures of low-loss and non-heat producing components are directly dependent on the case temperature and change with it at equal amounts. If conductive heat transfer occurs between the components and the case, the relationship of the temperatures is not affected because the conductive heat transfer is proportional to the temperature difference between the components and the case. The surface temperature of hot components of such units usually does not increase with the case temperature by an equal amount since they are largely cooled by radiation which is a function of the difference of the fourth-power of the absolute temperatures of component and case surfaces. Therefore, as the case temperature increases, the temperature difference between hot components and the case decreases somewhat.

2. Case Cooled by Forced Convection

The barrier to increased heat dissipation established by the available natural modes of heat transfer can be removed by providing for the forced flow of atmospheric cooling air over the surface of the case. Thus, without modifying the internal method of heat transfer from components to the case, and using the same methods as with units cooled externally by free convection and radiation, the heat load per unit volume of the case may be increased, since for the same case temperature and surface area the total heat dissipation is increased by virtue of an increased overall heat transfer coefficient. The simplest means of achieving forced convective cooling of the case is by surrounding the case with a concentric cylindrical baffle which forms an annular passage through which the flow of atmospheric cooling air is induced by means of a blower. The effect of shrouding the case completely by means of the baffle is to eliminate almost completely any radiant heat transfer to the environment. Consequently, environmental effects, except those of air temperature and pressure, are reduced to relative insignificance since the baffle forming the outer surface of the cooling passage would operate at a temperature not materially higher than that of the air in the compartment.

The relationships between the component temperatures, the internal air temperature, and the case temperature are the same as if the case were cooled by free convection and radiation, as long as the internal method of heat transfer is the same. The external cooling conditions which affect the case temperature are established by the temperature and pressure of the atmospheric air and the characteristics of the external cooling blower unit under the operational conditions of the aircraft in which the unit is installed. As mentioned above, the temperatures and surface characteristics of surrounding compartment surfaces are of little significance because of the small share which external radiant heat transfer has in the total heat dissipation from the unit.

3. Case-Envelope Heat Exchanger Cooled by Forced Convection

Units of the type described and discussed under the preceding classification, having a cooling air passage formed by the case and a simple baffle, are limited in heat dissipative capacity by the surface area of the case and the internal method of heat transfer which is assumed to be dependent on free convection or random agitation, and radiation. A better heat exchange system of greater heat dissipative capacity per unit surface of the case may be obtained by installing, similarly to the external baffle surrounding the unit, an internal baffle which is concentric with the case. A schematic design of such a unit is shown in Figure II-1. In this unit an internal peripheral passage (A) is formed which is designed to carry internal cooling air, conveyed by a blower, in heat exchange, through the wall of the case, with atmospheric cooling air conveyed through the external passage (B). The increase in heat dissipative capacity per unit volume of case resulting by the application of this method is dependent on the flow velocity past the surface of the case, since it determines the heat transfer coefficient at the internal surface and can yield values of increasing magnitude, depending on the extent to which power consumption for internal air circulation is permissible. In conjunction with this heat exchange system, the modes of heat transfer between the electronic components and the

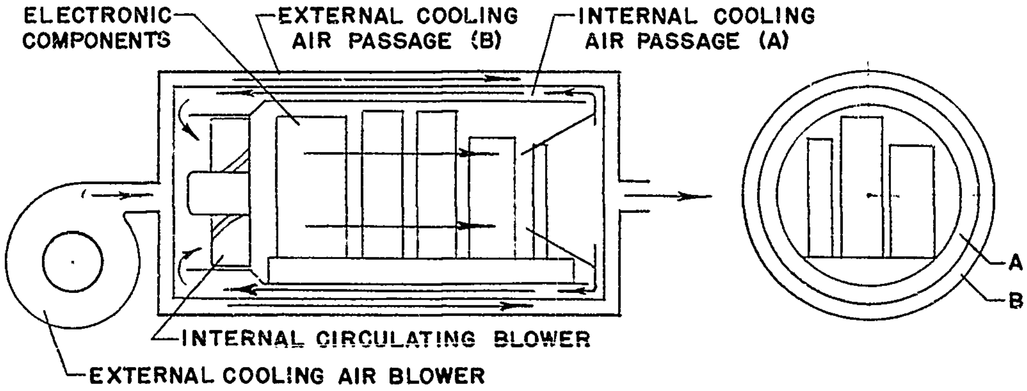


Figure II-1. Schematic Diagram of Air-Cooled Pressurized Unit with Simple Case-Envelope Heat Exchanger

internal atmosphere, as well as the case, are changed. The components are installed in baffled passages through which the internal air flow passes by virtue of the pressure differential created by the internal blower. Thus, the principal mode of heat transfer from the components is forced convection not only because of its more effective cooling action but also because direct radiation from the components to the case is practically eliminated by introduction of the internal baffle.

A more effective heat exchange surface enveloping the case can be produced by modifying the internal and external air flow passages in such a way that the heat exchanging wall of the case is provided with surface extensions so as to increase the effective case area. A cross section of one type of peripheral heat exchanger core is shown in Figure II-2. This construction further increases the heat dissipative capacity per unit volume of the case. In such a system the internal air flow should be distributed in a systematic manner so that a definite circulation pattern is established from the blower discharge, over the electronic components, through the internal heat exchanger passages, and back to the blower's inlet.

When the heat exchange surface envelopes the case as described above, the effects of surface temperature and characteristics of surrounding structures on heat dissipation are practically eliminated and the temperature of the heat exchange surfaces is principally a function of the temperature and pressure of the atmospheric air and of its flow rate. The latter is dependent on the characteristics of the external blower unit. While it would be desirable to hold the external heat transfer coefficient constant, it may not be possible to do so because of limitations of performance of the blower and, therefore, the temperature of the heat exchange surfaces would change. With forced convective heat transfer dominating internally, the temperature differences between individual component surfaces and the heat exchange surface are practically fixed if the internal cooling blower is operating continuously. Therefore, any change in the temperature of the heat exchange surface results in an equal change in the temperature of every component.

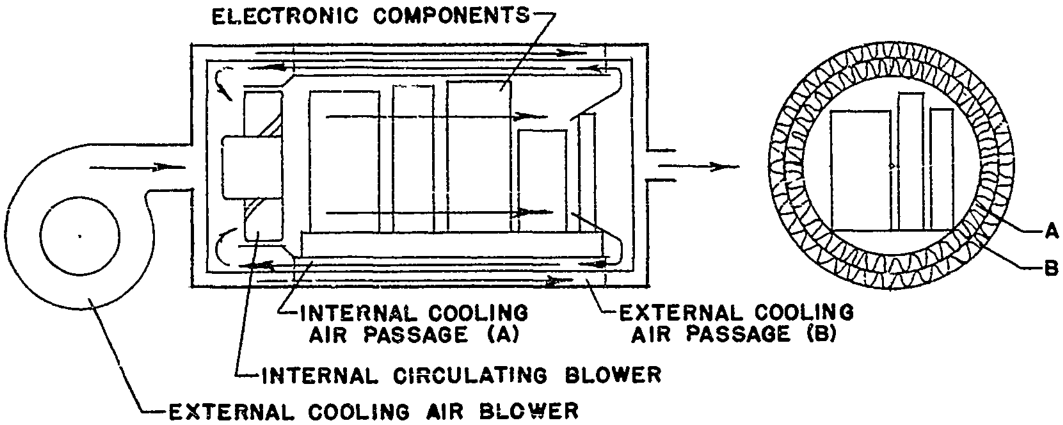


Figure II-2. Schematic Diagram of Air-Cooled Pressurized Unit with Case-Envelope Heat Exchanger Having Extended Surfaces

4. Integrated or Separate Heat Exchanger Cooled by Forced Convection

High rates of heat dissipation per unit heat exchanger volume and weight can be obtained by constructing the heat exchanger in a compact core through which internal and external cooling air circulates. Such heat exchanger cores may be so designed that their incorporation in the envelope of the case is not possible. A compact heat exchanger may be attached to each unit and may be cooled by an individual external cooling air blower as shown schematically in Figure II-3, or it may be used in form of a separate heat exchanger installed at a central location and serving more than one electronic unit, as shown in Figure II-4. In the latter type of application, the internal air flows of all units may be mixed before reaching the central heat exchanger or they may pass through adjacent sections in the heat exchanger core which is cooled by a common external air blower. Each unit, normally equipped with an individual internal air blower, has an inlet and an outlet duct which forms a closed circuit with the particular section of the heat exchanger core serving the unit. The internal air circulation system is the same as in the units in which the heat exchanger is incorporated in the case envelope. Consequently, the principal difference resulting from the use of a compact-core heat exchanger rather than an envelope-type heat exchanger is that individual component temperatures are slightly affected by external conditions since the case of the unit is exposed to free convective and radiant cooling. However, it is unlikely that the internal air temperature is materially affected by the temperature of the case, since the surface area of the heat exchanger core is much greater than that of the case and also the heat transfer coefficient between the internal air and the core is materially greater than between the internal air and the case surface. Therefore, the internal air

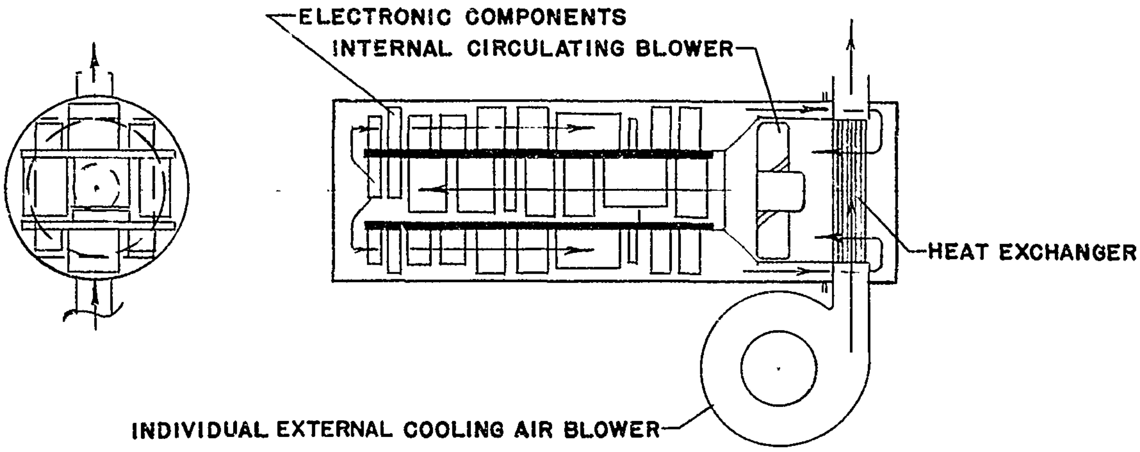


Figure II-3. Schematic Diagram of Air-Cooled Pressurized Unit with Integrated Compact Heat Exchanger Cooled by Individual Blower

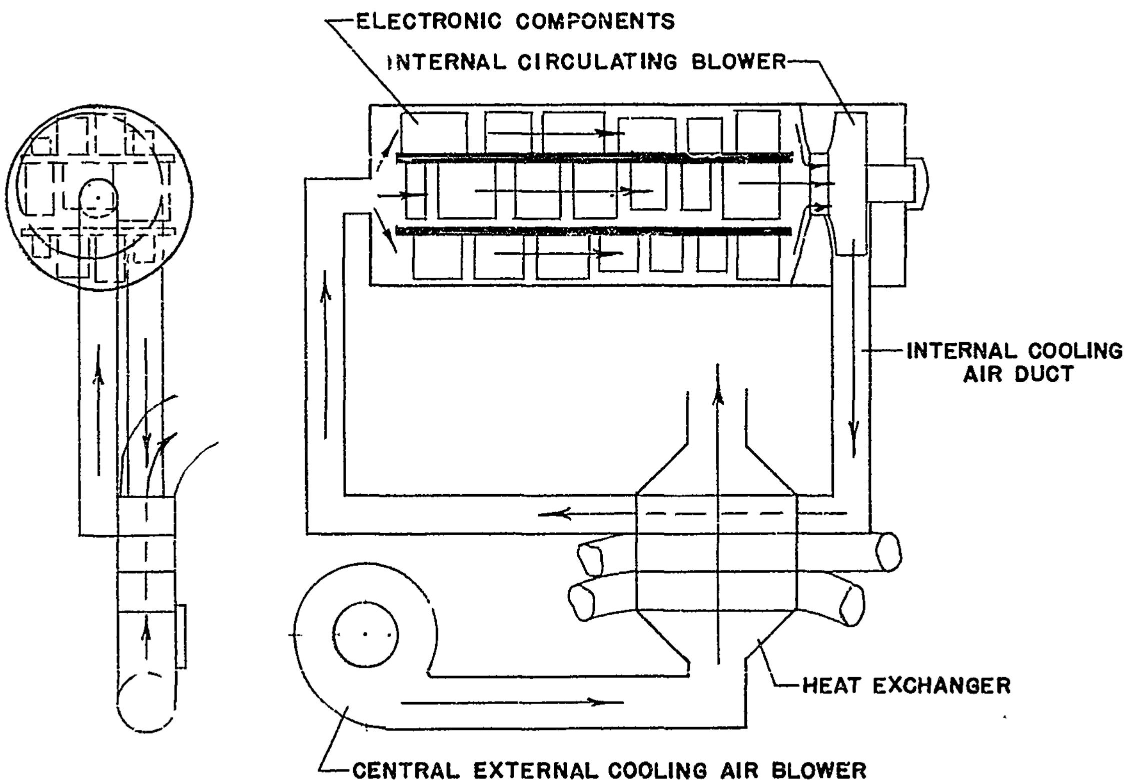


Figure II-4. Schematic Diagram of Air-Cooled Pressurized Unit with Separate Central Heat Exchanger System temperature is directly related to the air temperature at entrance or exit of the heat exchanger core. Because of high internal velocities the heat transfer from individual components occurs principally by forced convection and, therefore, the case temperature has little effect on component temperatures which are principally determined by the internal air temperature. However, there may be isolated components in close proximity to the case surface which would radiate a more appreciable portion of their total heat dissipation to the case surface. Such components are subjected to variable influences with variable case temperature and may not maintain an entirely constant temperature differential with the internal air temperature.

Vented Equipment

The modes of heat transfer contributing to the cooling of vented units and the electronic components they contain are the same as those of pressurized equipment. However, an important difference between the two major categories is that for vented equipment the overall internal heat transfer coefficient is not independent of operational conditions but varies vastly with altitude and external air temperature.

Vented equipment is usually enclosed in rectangular prismatic cases with flat surfaces. The surface temperature of the case under variable operational conditions is affected by environmental conditions and the extent to which they are conducive to radiant heat transfer. Since in vented equipment the internal electronic components are also in direct radiant heat exchange with the case surfaces, environmental conditions may have appreciable influence on the component temperatures, particularly when the other principal mode of internal heat transfer is free convection. In the following, more specific characteristics of the various classifications of vented equipment are discussed.

1. Closed Case Cooled by Free Convection and Radiation

Equipment with very low heat dissipation can conveniently be incorporated in a closed case which, however, is vented to the atmosphere. The heat load of such units may be from 50 to 150 watts per cubic foot, depending on size, i.e., surface-to-volume ratio, and operational conditions. The principal advantage in closing the case is to prevent contamination of the interior of the unit with dust. The pattern of heat transfer is exactly the same as for the analogous pressurized unit, except that a closed vented unit is unlikely to contain an internal blower. Thus, components are transmitting their heat to the case by free convection, radiation, and conduction, while the case dissipates the entire thermal output of the unit by free convection and radiation.

The relationships between component temperatures and case temperature are considerably more complicated than in the pressurized unit because the distribution between radiant and convective heat transfer from any individual component may change appreciably with environmental atmospheric pressure. While components which transmit their heat to the surfaces of the case principally by radiation and conduction change in temperature little when the operational altitude is changed, as long as the case temperature is constant, components which are dependent on free convective heat transfer would increase in temperature appreciably at higher altitudes. If part of their heat tends to be transmitted by radiation and conduction, this tendency increases at higher altitude so that the share of the latter two modes of heat transfer in the total heat dissipation from the component is increased.

2. Open Case with Through-Flow of Atmospheric Air by Natural Convection

Some increase in heat dissipative capacity per unit volume over that of the closed vented unit is achieved by equipping the case with ventilating louvres. They permit circulation of external air through the unit and over the component surfaces. Thus, the temperature of the internal atmosphere of the ventilated unit is lower than that of the closed unit and, therefore, components would tend to operate cooler at the same spacing, or would operate at the same temperature if arranged more closely. The mechanism of heat transfer from the entire unit is quite complicated in that only part of the total heat dissipation occurs from the surface of the case while the greater portion of heat is transmitted directly from the surfaces of the components to the air circulating through the unit. The volumetric rate of air circulation through the unit is a function of the temperature difference between the internal air and the ambient atmosphere, and the configuration of the passages. The weight flow of air through the unit is highly variable with altitude. For the same temperature difference between the internal atmosphere and the ambient air, the weight flow would vary in direct proportion to the atmospheric pressure, which would result in a substantial decrease in convective heat transfer from the components at higher altitude. However, there is some natural compensation built into the system which causes the convection to decrease less rapidly since the mean air temperature within the unit would be expected to increase with altitude, providing the ambient temperature remains constant.

In the distribution of total heat dissipation between the surfaces of the case and the air circulating through the unit, changes tend to occur as one or the other mode of heat transfer is altered in its effectiveness. For example, with increase in operational altitude, convective heat transfer will tend to decrease and the internal air temperature will increase. Consequently, component temperatures will also increase. As a result, the heat transfer from components to the case by radiation will increase and the contribution of the case to the total heat dissipation will also increase.

There are units which essentially fit into this category, except that they contain small blowers which are used for spot cooling. While such blowers provide a remedy to local problems of excessive temperature, their contribution to the improvement of the over-all pattern of heat transfer is usually not substantial. Their effects are difficult to isolate but counteract to a limited extent the natural tendency for decrease in convective heat transfer at higher altitudes.

3. Open Case with Forced Through-Flow of Atmospheric Air

Equipment with electrical characteristics permitting it to be designed open to the atmosphere, i.e. vented and with extensive interchange of its internal air with the environment, would usually be of minimum size if cooling is achieved by means of creating the air flow through the unit with a blower. In this manner, the pattern of cooling air circulation is controlled and is not substantially affected by external conditions, except to the extent to which the performance of the blower is influenced by the density of the ambient air in the compartment in which the equipment is installed.

Units designed for low pressure drop of the cooling air would have multiple inlets, usually distributed around the periphery of the case, and would have a centrally located blower serving to induce air flow through the openings in the case and over the components. The blower discharges the heated air through a single outlet of the unit. The flow could also be reversed so that discharge of the heated air would occur through the multiple openings. However, in the latter design distribution of the cooling air is somewhat more difficult to achieve. The use of internal baffles aids in giving the air flow desired direction. In general, the air flow pattern in such units is horizontal, while that of units relying on natural convection for circulation is vertical.

Units of greatest compactness can be produced by systematizing the air flow by means of subdividing the entire case into ducted passages in which the components would be installed. Thus, practically individual cooling control can be achieved for the various components while the pressure drop of the cooling air passing through the unit would be fairly large. Such units would have a single inlet and outlet and need not be equipped with integrated blowers. Instead, they can be cooled from a central blower system, if desired, either in parallel or in series with other units of similar design.

The cooling of components in units with forced air flow is principally convective. The effect of the installation environment on the heat dissipation from the outer case of such units is relatively small since the case temperature would not be substantially above the ambient air temperature. Heat transmitted from components to the case by conduction and radiation is largely picked up by the air flow passing through the unit. Change of the environmental conditions in respect to air temperature and pressure, affects the performance of the unit's blower so that the total convective heat dissipation is altered. Simultaneously, the temperature of the case, and with it, the total radiant heat dissipation is changed. However, an appreciable redistribution of total heat dissipation between forced convection and radiation would occur only under operating conditions where the blower capacity would be inadequate to supply the necessary volume of cooling air, thus resulting in excessive component temperatures. Otherwise, by far the major portion of total heat dissipation is effected by forced air convection from components, internal case surfaces, and radiation-absorbing internal baffles.

Test Methods

For the thermal evaluation of any electronic equipment, expected to operate under steady state or transient conditions, it is necessary that values of component temperatures, air temperatures, air pressure, air flow, and heat dissipation rate be known, as defined by compartment air temperature, altitude, and environmental conditions which may be constant or variable. As mentioned, there are three general test methods by which basic data for thermal evaluation may be obtained. They are bench tests, altitude chamber tests, and flight tests.

1. Bench Tests

Bench tests are characterized by operation of the equipment in the laboratory under prevailing atmospheric conditions. They may be so executed that data for thermal evaluation under steady state or transient conditions are produced. They are simple to execute, do not require elaborate test equipment, and are, therefore, relatively inexpensive. Although bench tests are best performed on individual units, it may be necessary to operate the entire equipment if representative electronic operating conditions cannot be produced conveniently for an individual unit. Measurements of a large number of local component and surface temperatures, requiring extensive thermocouple installation, are necessary to provide conclusive thermal data. Therefore, thermal data may be obtained conveniently for only a single unit at a time. Bench test data may be analyzed and extrapolated to provide an indication of the thermal performance of electronic equipment over a wide range of operating conditions. The accuracy of the extrapolation for a particular condition of air temperature, altitude, and type of operational environment varies somewhat with the cooling methods employed for the particular unit under consideration. The thermal performance of pressurized units may be quite accurately predicted from bench test data. For vented units cooled by either free or forced convection, the extrapolation of bench test data is less accurate due to the complexity of the variables affecting the thermal behavior of individual components. For units cooled by forced convection, a proper interpretation of the bench test data provides a means of estimating the required capacity of motor-blower units under various operating conditions.

2. Altitude Chamber Tests

Altitude chamber tests are well suited to simulate some operating conditions during flight, but generally do not completely define installation conditions. While air temperature and pressure are controlled in such tests, other environmental conditions such as air motion, or surface temperature and characteristics of surrounding surfaces are not easily simulated. Therefore, altitude chamber tests fulfill only two principal functions. One is to ascertain whether equipment will withstand specified operational limits which are only defined in terms of air temperature and altitude, the usual method in present specifications for equipment development. The other function of altitude chamber tests should be the verification of component performance, as predicted from the extrapolation of bench test data of an entire unit. Thus, it is possible to make altitude chamber tests with the minimum of instrumentation since only those component temperatures which are predicted as critical need be measured.

While the chamber test procedure is primarily applicable to steady-state operation, transient operating conditions can also be studied within the scope of the controllable variables of air temperature and pressure. The effects of rapidly varying temperatures of surrounding surfaces are difficult to simulate in altitude chambers. For units which utilize forced convection and contain integrated blower equipment, the chamber tests can be used to determine whether the motor-blower unit provides adequate cooling under critical conditions of air temperature and altitude. Quantitative information on the weight flow of air under various operating conditions is usually not obtained. Spatial arrangement of equipment in an altitude chamber differs greatly from actual installation conditions in an aircraft. Consequently, particular attention should be given to correcting the data for the environmental conditions.

For certain types of equipment, particularly vented units with natural through-flow of air, altitude chamber tests must be relied upon for evaluation of performance under aircraft operational conditions. For such tests suitable provisions must be made to simulate as closely as possible installation conditions by preventing the operational characteristics of the chamber, particularly its air circulation pattern, from affecting the heat dissipation mechanism of the unit.

3. Flight Tests

Because of high cost and the difficulty of instrumentation, flight tests should not be used for the accumulation of numerical data for the preliminary thermal evaluation of electronic equipment. They are suitable for equipment which is expected to reach steady-state conditions when operating in the aircraft. However, their use is most desirable to ascertain conditions resulting from transient operation under variable flight conditions, or under essentially constant flight conditions of short duration. They serve as final checks on the actual installation of the equipment to determine whether its operation is satisfactory under the most critical conditions. In such tests the use of extensive instrumentation for temperature explorations is not practical. Therefore, only critical temperatures are measured. A unit not operating properly can be equipped with supplementary instrumentation to determine the causes for overheating. However, if no unusual external influences are found to be present, it is usually most practical to test such a unit more thoroughly on the bench or under altitude chamber conditions, before additional flight tests are made.

CHAPTER III

MEASUREMENTS

Thermal evaluation of any electronic unit is dependent on knowledge of temperatures and heat dissipation of individual components, temperature and flow distribution of the unit's internal atmosphere, overall flow rates and temperatures of the cooling air, the total rate of heat dissipation and external surface temperatures of the unit, and the temperatures and characteristics of the environment. Thus, the bases of evaluation are the measurements of (1) electrical variables for the calculation of heat dissipation, (2) temperatures, (3) static pressures, (4) air flow, and (5) velocity distributions.

Since this report is principally written for the electronic engineer, electrical measurements are not discussed in detail. However, it should be mentioned that the measurements required for thermal evaluation rarely need to be more complete than to give the necessary data for the calculation of the total rate of heat dissipation of all circuits contained in a unit. This is essential since a primary characteristic of the unit, its heat load, is then defined. In addition, the rates of heat dissipation of individual components, as estimated from their characteristics and the design of the circuit, can be checked approximately by the overall rate of heat dissipation. Measurements for the experimental determination of individual components' heat dissipation rates are not necessary if the characteristics of the circuit and components are known. Such measurements are only indicated if overheating of individual components cannot be explained by inadequate cooling design.

The measurements of temperatures, pressures, air flow, and velocity are treated more specifically in this chapter because they are of great importance, and their techniques may not be within the field of direct experience of the electronic engineer. The measuring methods indicated for the thermal evaluation of electronic equipment are of engineering nature, and accuracy of measurement to within less than one or two per cent is not a suggested requirement. When critically examining recommended methods of measurement and instrumentation, it should be kept in mind that, in view of the qualitative and comparative nature of thermal evaluation and the necessity for simplification of procedures for reasonable speed of execution, the very precise measurement of temperatures to fractions of a degree, static pressure to thousandths of an inch of water, and air flow to more than three significant figures, are not only unessential, but also undesirable.

Temperature Measurements

The determination of temperature distribution in electronic equipment is the principal task in thermal evaluation. Even without further analysis, temperatures alone give some qualitative indication of the unit's operation and design. Among them, component temperatures are most significant. However, as previously mentioned, temperatures of the inner and surrounding atmosphere of the equipment, as well as other surface temperatures are necessary in the analysis of the thermal performance since component temperatures are directly dependent on them. Therefore, these temperatures affect the operating limits of the equipment.

1. Component Temperatures

Electronic components are composite structures with a wide variety of internal heat sources. In one case, such as the filament of a tube, heat to be dissipated may be generated within the component in a space representing a small fraction of the total volume of the component. In another case, such as a transformer, the heat source may in essence be uniformly distributed over the entire structure. Therefore, it is difficult to speak, without qualification, of the surface temperature of a component as a singular value. This has been realized in practice and the term "hot-spot" temperature, referring to the maximum surface temperature of the component, has been generally accepted as a component characteristic. In reality, this temperature is not always descriptive of the structural characteristics of a component and its internal heat distribution, since it is often greatly affected by the method by which the component is cooled. A vacuum tube develops hot spots almost independently of the cooling method by virtue of its internal structure's and heat source's relationships to the envelope. For the elimination of such hot spots, change of the general cooling method is relatively ineffective. Only application of highly localized cooling can produce an essentially uniform surface temperature on such components. On the other hand, if a component like a resistor develops a hot spot, the principal cause is furnished by the cooling method. In this case, a general change in the cooling method, possibly involving a change in the position of the resistor, can result in an essentially uniform surface temperature. This is desirable for best utilization of the heat-resistive properties of the component's structural elements.

Knowledge of the location of inherent hot spots on components by virtue of construction or installation is essential since they are usually referred to when a component's limiting surface temperature is given by the manufacturer. In that event, a single surface temperature measurement on the component is adequate for the equipment's thermal evaluation. The location of the maximum surface temperature may be mentioned in rating sheets together with its allowable value. It may also remain unspecified because certain components have, by experience, demonstrated that their maximum temperature occurs in similar locations. Should a value of the maximum allowable surface temperature be specified without its location, the latter can be estimated by experience and the measurement can then be made at that point. If heat-generating components are rated only on the basis of ambient air temperatures, reproduction of the rating conditions is desirable in order to measure the allowable hot spot temperature. For this purpose, a closed test chamber should be used.

Figure III-1 shows the suggested arrangement of a test chamber which may be used for the determination of component hot spot temperatures and their locations at rated ambient air temperature. The principal purpose of a component test in such a chamber is to establish a reference value for the hot spot temperature and to fix the location where a thermocouple should be installed in the thermal evaluation test of the unit containing the particular component. No conclusions as to the performance of the component in the unit should be drawn from the test alone since its behavior in the circuit surrounded by other components may be quite different. This particular test

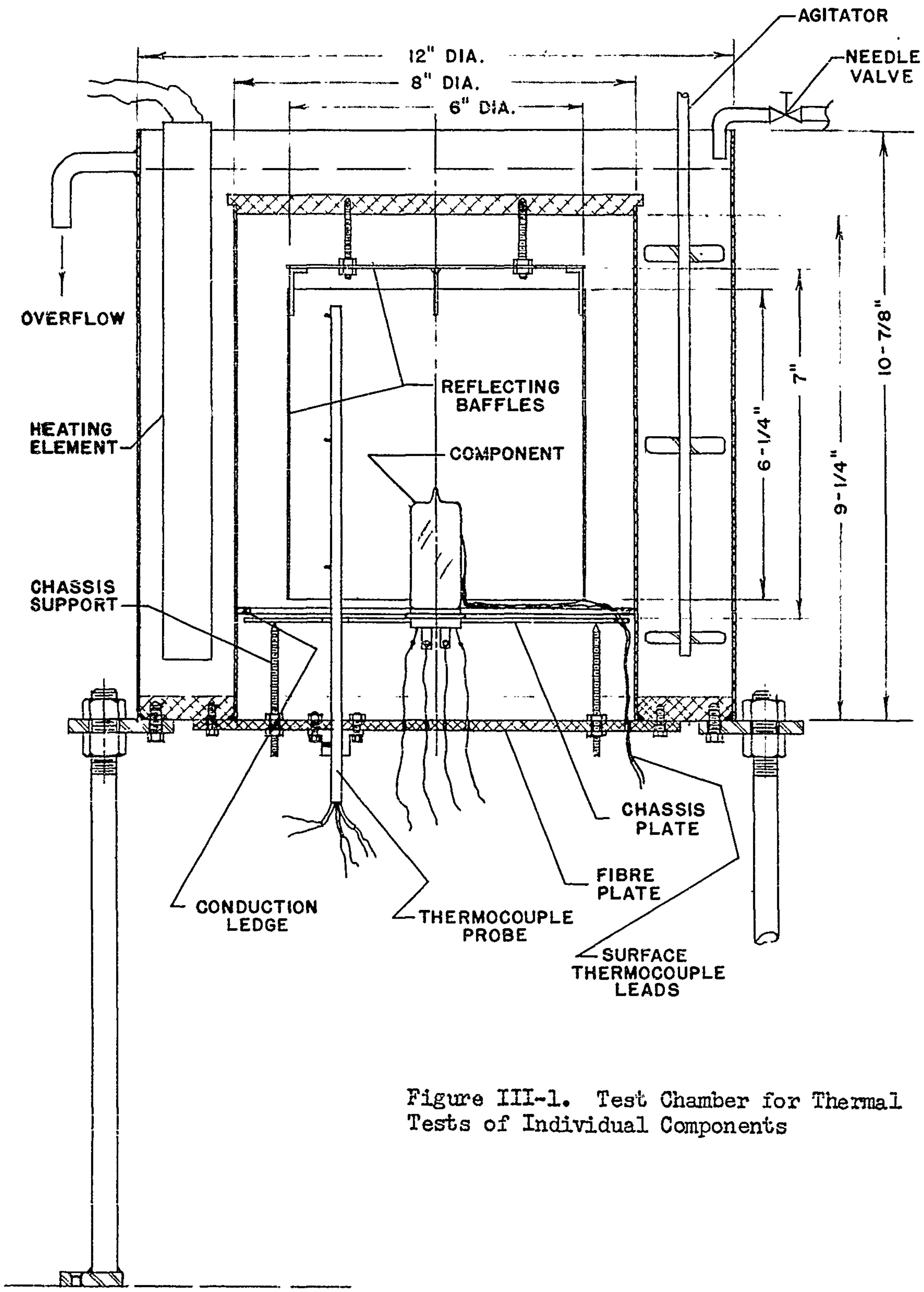


Figure III-1. Test Chamber for Thermal Tests of Individual Components chamber is so designed that it is capable of dissipating 50 watts for an internal air temperature of 85°C when the wall temperature of the inner container is about 30°C. The heat-dissipating capacity of the chamber would change proportionally to the temperature difference between the wall and the internal air, should internal temperatures different than 85°C be desired. The wall temperature of the inner chamber can be controlled by means of the agitated liquid bath surrounding it. Unless the internal air temperature is to exceed 100°C, water can be used as the bath fluid. Its temperature level can be controlled by means of the addition of fresh water at a small rate which can be adjusted manually and by the automatic control of an immersed electric heater. The operation of the heater is controlled by means of a temperature-sensing element within the inner chamber, since the eventual effect of change in the bath temperature is a change in the internal air temperature. For high-temperature conditions which do not permit the use of water as a bath fluid, a silicone fluid is best used. In that case, it is usually not necessary to cool the bath since its temperature would be sufficiently high that control can be obtained when depending on heat dissipation from the outer container wall to the atmosphere of the room. For the construction of the inner container, polished aluminum is desirable so as to reduce its thermal capacity and to provide for high reflectivity. It should be noted that the suggested arrangement calls for the installation of baffles within the test chamber. They would operate at a temperature close to that of the internal air. Their purpose, if made of highly polished aluminum, is to prevent radiant heat transfer from the component to be tested so that cooling can only occur by free convection. Thus, the most unfavorable cooling conditions would exist and the highest hot spot temperature corresponding to the rated ambient temperature would be obtained. If this hot spot temperature is to be used as a reference temperature in evaluation tests of equipment, it is possible that it may be excessive. Therefore, particularly for components such as tubes which potentially have a large radiant heat transfer capacity, it may be desirable to obtain the hot spot temperature with a modified apparatus where the internal reflecting baffles are replaced by others of the same configuration but coated with carbon black. With the latter baffles, more favorable conditions for radiant heat transfer would be produced and, therefore, a lower hot spot temperature would be obtained. If the latter temperature is used in thermal evaluation studies, conservative estimates of required operating conditions for the particular component will be obtained. The same considerations as pointed out for the internal baffles also hold for the construction of the chassis plate which may either be of polished aluminum or coated with carbon black, depending on the desired conditions. It may also be desirable to utilize a third modification for components, such as transformers, which may be cooled effectively by conduction, by providing for solid contact between the chassis plate and the cooled walls of the test chamber. The method for doing so is not obvious from the design shown in Figure III-1 since it may be noted that the chassis plate fits loosely in the chamber in order to facilitate its withdrawal through its bottom opening. However, good contact can be established with metallic shims between the edges of the chassis plate and the projecting ring shown as part of the chamber wall immediately above the plate. The ring is so located that a vertical air space of about 1/8 inch would exist between it and the edge of the plate when no conduction cooling is desired. The chassis plate is supported on a red fiber plate which bolts to the bottom of the test chamber and has small openings which permit wiring to be passed through it. The wires are not intended to be sealed in the fiber plate since the openings through which they pass should also serve as vents which permit equalization of the air pressure within the test chamber with that of the atmosphere. Thermocouples for the determination of the air temperature distribution within the chamber are supported on a vertical probe attached to the bottom of the apparatus.

Experience has shown that maximum surface temperatures on tubes occur where the internal heat-radiating structure is closest to the envelope. This is usually at two opposite points adjacent to the plate structure if the tube is subjected to essentially symmetrical cooling conditions. A single temperature measurement at either location should be sufficient to give an indication of the tube's operation. In equipment internally cooled by forced convection, non-symmetrical cooling of tubes may occur. If such conditions are ascertained by inspection, measurements at both potential hot spot locations are desirable.

A single temperature measurement on the coil surface of a transformer may describe its operating condition since it may indicate a temperature close to that of its hot spot if the transformer is so installed that conduction to the chassis can take place from the core, while heat dissipation from the surface is not great. With little conduction cooling, but extensive heat dissipation from the surface, the innermost coil winding may have a hot spot which is sometimes difficult to determine. Since the basis of a transformer's operating limitation may be an ambient temperature rating, magnitude and location of internal and external reference temperatures are best determined in a rating chamber as shown in Figure III-1. The known value of the insulation's breakdown temperature can also be used as the basis of an estimated allowable temperature which should be located by at least two spot temperature measurements, one in a readily accessible location as close as possible to the center of the windings and one at the surface of the windings. This is a feasible procedure for open-type transformers. Internal temperature measurements on transformers sealed in pitch-filled metal enclosures are not practical and, therefore, only the surface temperature of the enclosure need be measured at one location since it is usually uniform. The limiting temperature can also be assumed to be the softening or decomposition point of the impregnation material.

Considerations similar to those for transformers also apply to relays. Their maximum temperature is found on the holding coil and is limited by the heat resistive properties of the insulation. Sealed relays must be checked on the basis of surface temperature, pre-determined by single-component tests.

Single resistors installed horizontally and cooled by free convection usually develop a hot spot at top center. When installed in the vertical position the maximum temperature occurs near the top. Forced convection cooling may change the location of the maximum temperature materially. In cross-flow it may be estimated to occur at points about 60 to 90 degrees from the planes of impinging air flow, measured along the periphery. Forced convection with flow parallel to the axis yields a relatively uniform surface temperature, although probably highest at the trailing end of the air flow. The limiting temperature of the resistive coating material is best used as reference determined by a single surface measurement. At present, a representative value would be in the order of 300°C.

Non-heat-producing components, rated on the basis of the maximum air temperature in which they can operate for a specified life period, are not normally subjects of surface temperature measurement if the temperature distribution of the air within the unit is investigated. However, such components may develop a hot spot temperature above the limiting temperature because of external influences exerted on them by adjacent components. Therefore, if it appears from a study of the unit's configuration that radiation or conduction from high-temperature components may raise the local surface temperature of a component such as a condenser substantially above the air temperature, a surface temperature measurement on that area of the component must be made. The precise location is not too critical.

2. Air Temperatures

Measurement of air temperatures for evaluation purposes is necessary inside and outside of electronic equipment. When a definite flow of air exists through the unit or through passages surrounding its case, knowledge of the air temperatures at the entrance and exit is essential.

Air temperature measurements within electronic units are usually made for exploratory purposes to obtain indications of air distribution and circulation patterns. They are of qualitative nature and are desirable for overall evaluation purposes, as well as for the study of cooling effects on individual components. They are spot temperature measurements and must be determined by means of a small primary measuring element, unaffected by surface temperatures existing in the environment. Therefore, shielding of the element from incident radiation is necessary. The basic method of shielding a primary temperature measuring element most applicable to temperature measurement in the evaluation of electronic equipment is relatively simple. A principal requirement is that the shield be so installed that the natural air circulation existing in the location is not impaired. In a free convection field the shielding surfaces should be vertical. In forced air flow the shielding surfaces should be parallel to the main direction of flow. The material of the shield must be highly reflective to incident radiation. Polished aluminum and thin silvered glass are most suitable. The thermal capacity of the shield should be small so that equilibrium with the air is reached readily. The preferred shape of the shield is a cylinder, entirely surrounding the temperature measuring element. For temperature differences usually encountered between air and radiating surfaces in electronic equipment, the use of a single cylindrical shield is adequate because the shield temperature would be barely above the air temperature. Between groups of resistors operating at maximum surface temperatures in the order of 300°C a double shield should be employed. This shield consists of two concentric cylinders of reflective material mounted with an annular air gap between them. Shielded thermocouple probes are shown in Figure III-4.

Measurement of air temperature outside an electronic unit being evaluated is usually necessary to define environmental conditions. The temperature so ascertained should be quite general and should represent a fairly large region of the environment without, however, being affected by radiant heat transfer between the measuring element and the environment. Therefore, for good accuracy it is usually necessary to shield the element. If test conditions are well controlled, shielding of the element from radiation from the equipment is usually more important than shielding from radiant heat exchange with surrounding surfaces.

Measurement of air temperatures at the entrance and exit of cooling passages is very significant because the effectiveness of overall heat exchange can be calculated from it. By using the temperature rise of the cooling air and the measured rate of air flow in heat balance calculations, heat losses or gains can be determined, which otherwise cannot be accounted for, or can be estimated only very inaccurately by calculation. In order to obtain representative temperature measurement of air flowing at some velocity, complete mixing of the air is necessary. This is not always attainable at measurement stations near the entrance and exit because of stratification and combination of streams of different temperatures shortly before measurement. Therefore, such measurements require care in execution and should be preceded by a careful determination of the air temperature distribution in the passage, which can be obtained by probing with a small primary element. In conjunction with this, the velocity distribution must also be determined so that the individual temperature measurements can be properly weighted. It is usually possible to determine by this method a location in the passage where the air temperature is equal to the weighted average. Then, after the preliminary work, a single temperature measurement can be used with fair accuracy for the remaining test work.

The method for determination of the temperature distribution is based on subdivision of the flow passages into areas of equal magnitude. The temperature and velocity are measured in the geometrical center of each subdivision. It may also be possible to probe for the air temperature and the velocity stepwise along lines forming diameters, diagonals, and axes of symmetry of the flow area's cross section. The profiles can then be plotted from the data and the temperature and velocity distributions may be integrated graphically.

3. Chassis and Case Temperatures

Measurement of chassis temperatures in electronic units having low rates of heat dissipation is usually of minor importance. However, in units having high-temperature components of high heat dissipation rates, chassis temperatures should be considered in the thermal evaluation. They indicate the extent to which the chassis is being used as a heat transmission path in the cooling of the components. The chassis is capable of absorbing thermal radiation from high-temperature components and of transferring the heat to either the cooling air or areas at lower temperature within the case. The chassis may also aid in cooling certain components by conducting heat directly by metal-to-metal contact from the component to areas of lower temperature. In general, spot temperatures must be measured on the chassis. Therefore, thermometric elements of small size, easily installable in or on the material of the chassis must be used.

Case temperatures are required to permit computation of the amount of heat dissipated externally by radiation and free convection. The case temperatures are also used in estimating environmental effects and thermal performance under various air temperature and altitude conditions. Since case temperatures are usually non-uniform, a sufficient number of surface temperature measurements must be made to permit the determination of a mean case temperature. Therefore, like on the chassis, local temperature measurements must be made with the same type of thermometric elements. In most designs the case serves as a simple heat exchange surface. Measurements of surface temperatures on other heat exchanger surfaces incorporated in electronic units are of the same type as those on the case.

4. Environmental Surface Temperatures

Measurement of surface temperatures in the environment of electronic units being tested is not always necessary. If walls, screens, or surfaces of large unheated enclosures are involved, their temperatures can be assumed equal to that of the surrounding air. However, in the performance of tests, unavoidable conditions may be encountered where some surface with a temperature substantially lower or higher than the case of the unit is in radiant heat exchange with it. Then the measurement of this temperature by at least one small thermometric element imbedded in or attached to the surface is mandatory.

5. Primary Thermometric Elements. Thermocouples

From the discussions in the preceding portions of this section, it may be noted that in the thermal evaluation of electronic equipment point temperature measurements are of pre-eminent importance. Temperature measurements requiring installation of a thermometric element in intimate contact with a surface predominate. Air temperature measurements in restricted spaces are almost equally important. For all such applications thermometric elements are needed which are small, can be installed easily, and furnish an indication which can be determined in a remote location. These requirements are most satisfactorily met by thermocouples. Thermocouples are also applicable for measurements where space is not a restrictive criterion, such as in the determination of environmental surface and air temperatures. Particularly for the latter purpose, glass-stem thermometers or indicating-type expansion thermometers could also be used. By nature, they tend to integrate environmental temperatures and may, for this reason, be more desirable. However, for simplification of instrumentation and ease of reading from a central test stand, it is recommended that all temperature measurements be made by means of thermocouples.

A thermocouple is made by joining together two dissimilar metals, usually in the form of wire. If the two metals are joined in two places of unequal temperature, a current will flow in the closed circuit. This is indicated when the two leads are attached to a voltage-measuring instrument while the other ends of the wires which are joined together are subjected to a temperature different from that at which the instrument operates. Then the circuit is closed by means of the instrument and the latter will indicate an e.m.f. which is a function of the temperature difference between one junction of the wires and the temperature of the instrument. This arrangement is shown in Figure III-2. The external junction of the wires is designated as the cold junction, or the reference junction. The latter may vary with the room temperature and, therefore, the indication obtained for a given hot junction temperature may not be constant, unless compensation is made within the instrument. In commercial instruments this is usually accomplished automatically. Another method of utilizing thermocouples for more precise temperature measurement is to incorporate into the circuit another junction made between the two wires which serves as a cold junction and is maintained at a constant controlled reference temperature such as may be produced by means of ice. This arrangement is also shown in Figure III-2. It has the advantage of yielding measurements of greater precision but involves additional complication in obtaining the data. The e.m.f. values for most commonly used metal combinations, as functions of temperature difference between the hot and cold junction, are given in tables commercially available. The choice of materials used for construction of the couples depends upon the operating conditions, as defined by temperature and character of the surrounding atmosphere, and may be vastly different. Before thermocouples are used in test work it is well for the operator to familiarize himself with the principles of thermo-electricity and to review information of more detailed character available on thermocouples. Study of Reference 1, given at the end of this chapter, is recommended.

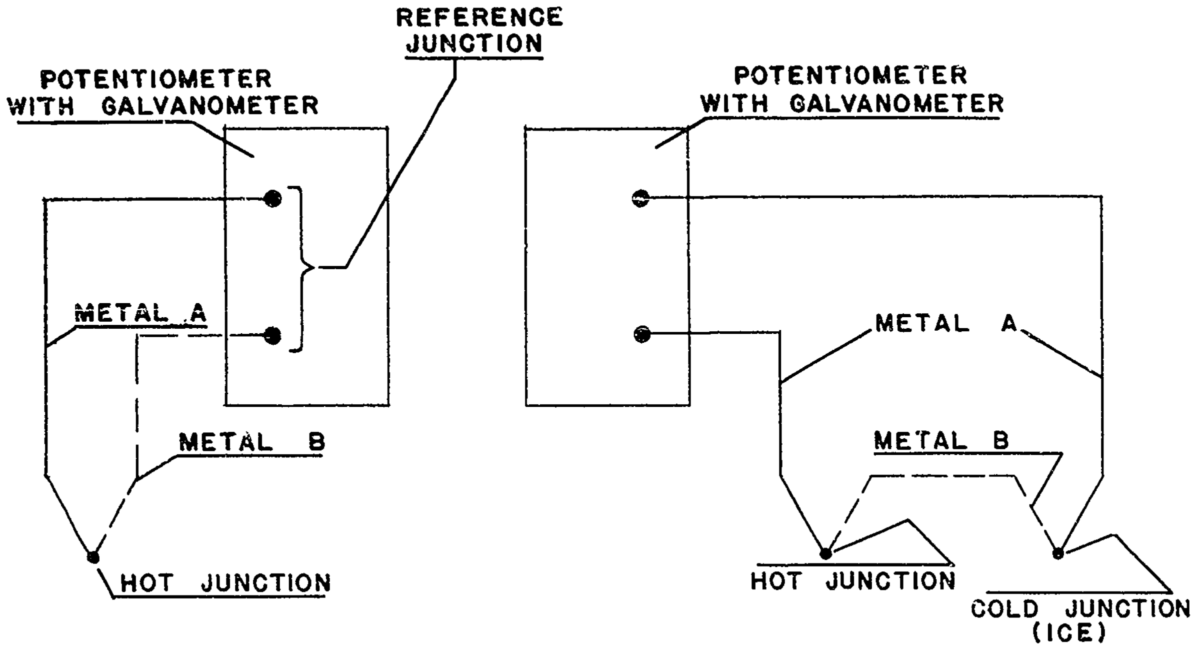


Figure III-2. Thermocouple Circuits

The following discussion is primarily intended to cover outstanding points which are particularly important in the application of thermocouples to temperature measurements in electronic equipment.

The large number of measurements necessary, the range of temperatures and the atmospheric conditions indicate the use of base metal thermocouples. Iron-constantan, copper-constantan, and chromel-alumel couples are suitable. Iron-constantan couples are most useful in the commonly encountered range of temperatures in electronic equipment. For temperature measurement below minus 20°C, copper-constantan couples are preferable. They can also be used within the entire operating temperature range of electronic equipment. Chromel-alumel couples are principally applicable to surface temperature measurement on high-temperature components.

The size of wire to be used must usually be the result of compromise. Ruggedness, ease of handling in installation and ease of forming junctions increase with increasing size of wire. Responsiveness, accuracy in the determination of point temperature, and insensitivity to temperature gradients along the wire increase with decreasing size of wire. The smallest size of standardized wire readily available is 30-gage, or 0.010 inch diameter. It is applicable to most measurements for thermal evaluation, although it may be too large for subminiature components of high surface temperature. For measurements of external air temperature and environmental surface temperatures, larger wires can be used. Plain enameled wire may be used where space is limited. However, a variety of insulations are available which are more serviceable than enamel. Cotton-wrapped enamel wire, or glass-sheathed asbestos-coated wire are advantageous because they resist mechanical breakdown of the insulation better than enameled wire. They are more difficult to accommodate within the equipment and require more elaborate means of making connections from the interior of the unit to the outside. Plain enameled wire can usually be passed between two rubber gaskets from a pressurized unit to the outside.

Couples of wire smaller than 30-gage should be calibrated for high accuracy. Although, with care in selection, wire of at least 0.004-inch diameter can be obtained which would follow standard calibration data with sufficient accuracy for this type of work. The smaller wires are often available only bare or enameled. If used, they should be used only in short pieces adjacent to the junction which are spliced to well-insulated 30-gage lead wires. While fine-wire couples are necessary for the study of subminiature components in the development stage, it is questionable whether the difficulties associated with their use warrants their application in thermal evaluation work. Their use is only recommended under circumstances where the apparent error with 30-gage couples is so great that the danger exists that temperatures exceeding allowable limits may remain undetected. Errors in the measurement of the true component surface temperature of 5°to 10°C are of no great importance for the present purpose, particularly since components are rarely changed between test runs for thermal evaluation. Thus, relative values of sufficient accuracy are obtained.

The required size of the thermocouple junction is dependent upon the application and method of installation. Junctions which are large compared to the wire diameter, can be handled more easily and give more accurate indications if a steep temperature gradient exists along the adjacent wire. Small junctions are more easily installed on surfaces and give a more representative indication of the point temperature if the temperature gradient along the wire is minimized. The shape of the junction is determined by the method used to make it, but can be modified by mechanical deformation so as to suit best the installation conditions. The original junction of small wires should be made by an electrical welding process. Arc, spot, or flash welding can be used. (Arc-welded junctions are best produced by drawing the arc between one electrode formed by the two wires held parallel by twisting together, and a mercury bath forming the other electrode.) The required voltage can be determined by experience; it varies among metals, wire sizes, and size of bead. This method is difficult to use for wires of smaller than 0.004-inch diameter. The bead produced by arc welding is essentially spherical. For installation on surfaces it can be flattened mechanically. Junctions produced by spot-welding the two crossed wires are particularly suitable for thermocouples which are to be installed on surfaces. Also, it may be feasible that the two wires which are to form the thermocouple may be spot-welded individually to the surface whose temperature is to be measured. By this method, an indication of the mean temperature between the points of contact of each wire is obtained. Flash-welding in which each wire acts as an electrode produces a butt-joint between the two dissimilar wires which is most desirable for surface installations. Both spot- and flash-welding are especially suitable for fine wires of 0.004-inch diameter and smaller.

The installation of thermocouples in and near electronic equipment must mainly be made on surfaces, within solid sections and in small or large air spaces. Surface installations are most critical because intimate contact must be assured and no abrupt temperature change should occur along the leads attached to the junction. For the latter purpose, it is essential to provide for contact of the individual electrically insulated leads with the surface for a distance from the junction equal to 50 to 100 wire diameters. For small wires, as used in the applications under consideration, use of a small quantity of heat-resistant adhesive cement is necessary to avoid contact of bare wires with each other at any location other than the thermocouple junction, or contact of both wires individually with a common metal surface, such as the chassis or the case.

Intimate contact of the junction with the surface whose temperature is to be measured is best obtained by either cementing the couple to the surface, or by introducing the junction into a small recess and peening the material around it so as to fasten the couple securely to the surface. Cemented couples are universally applicable regardless of the material of the surface. Their accuracy in indicating true surface temperature is fair. They rarely yield exactly reproducible indications for two components of the same type. However, they are satisfactory for evaluation purposes. They should be principally used on non-metallic surfaces, or on metallic surfaces which should not be deformed by peening. The junction should be as flat as possible to increase the area of direct contact with the surface.

A satisfactory cement for the application of thermocouple junctions to component, chassis and case surfaces is Insa-Lute No. 1 (Reference 4) which is heat-resistant and oil-proof, but soluble in water. The latter characteristic is advantageous because it facilitates removal of the couple after use. The adhesive properties of the cement to clean glass and plastics are excellent. If used on metal, removal of paint is necessary. Before application of the cement, it is best to tape the thermocouple leads to the surface so that the junction and the short length of insulated leads adjacent to the junction are in good contact with the surface. The smallest quantity of cement sufficient to attach the junction permanently should be used. It should usually cover an area of less than 1/8-inch diameter. It must be allowed to harden completely either by air-drying or baking. An excessively large quantity of cement results in a high temperature reading since the cement acts as an insulating barrier to the flow of heat by radiation and convection from the surface covered by the cement. On glass tubes a large cemented area may cause cracking of the glass envelope when the tube is operated, because the coefficients of thermal expansion of glass and cement are slightly different.

A peened-in couple on a metallic surface gives an accurate temperature indication because the junction is fully immersed in the material and practically no temperature gradient exists within the depth of the couple's penetration. In order to make this method of installation feasible, it must be permissible to drill a small hole in the surface. The bead of the junction should fit the diameter of the hole closely. Then, only slight deformation of the surrounding metal by means of three or four indentations made with a small punch should be sufficient to fasten the couple securely. Care must be taken to avoid breakage or weakening of the lead wires when the junction is peened in.

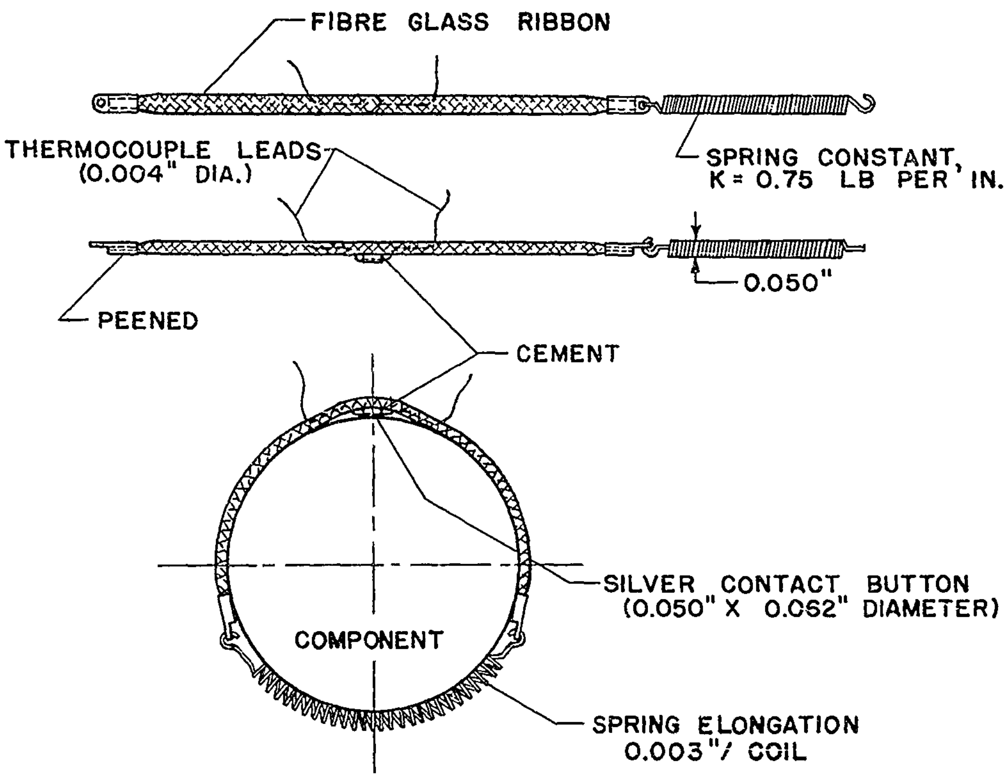


Figure III-3. Removable Thermocouple Element for Surface Temperature Measurement

A method for measuring point surface temperature with better reproducibility, when embedding of the thermocouple is not possible, consists in the application of a removable thermal element of the type shown in Figure III-3. It is important that the thermocouple wires do not exceed the size indicated and that the silver contact button is of the dimensions shown. Unless the curvature of the surface corresponds to a radius less than 1/8 inch, the button's contact surface may be flat, as shown. The contact pressure is of principal importance. The springs used for attachment should be wound as indicated with a spring constant of 0.756 pound per inch and should be elongated 0.003 inch per turn of coil to produce the proper contact pressure. Using these precautions, the device will give an indication of the true surface temperature within less than 2°C over an appreciable range of heat dissipation rates, including conditions of forced convection and liquid cooling.

Another method of measuring surface temperatures by means of thermocouples has been used by Sylvania Electric Products, Inc. on subminiature tubes. It differs from those mentioned above, which principally refer to hot-spot temperature measurements, in that it produces an indication of the mean surface temperature. In the described application, a tightly-fitting spring-clip is placed around the component, such as a tube, with the two thermocouple lead wires spot-welded individually to the clip on opposite sides. Reproducible temperature measurements among several components of the same type have been reported when this method of thermocouple installation is used.

By attaching individual couples by means of a controlled quantity of cement at the hot spot, as outlined above, precise reproducibility among different components of the same type is not obtainable. However, for evaluation purposes this method appears to be adequate and simple. On the other hand, if many similar temperature measurements are expected to be made on certain types of components in various units, the initial investment in procuring a sufficient number of clips with thermocouples may be warranted since the labor of repeatedly cementing couples would be eliminated.

The installation of thermocouples in air is considerably simpler than on surfaces. It is important that the junction and leads are properly supported. For that purpose, a small porcelain tube, preferably provided with two axial holes and 1 to 2 millimeters in diameter is most suitable. This insulator is also capable of supporting a small radiation shield which, if required, would have a diameter of about 1/4 inch and be about 1/2 inch long. Depending on the size of the space, a shield of smaller diameter can be used. However, care must be taken that the shield is sufficiently large and is so supported that air can circulate through it freely. This general method of supporting thermocouples in air is particularly satisfactory inside of equipment and flow passages and is applicable to stationary and moveable probes. Two types of shielded probes are shown in Figure III-4. Considerable simplification is possible, at the sacrifice of some accuracy in locating the couple,

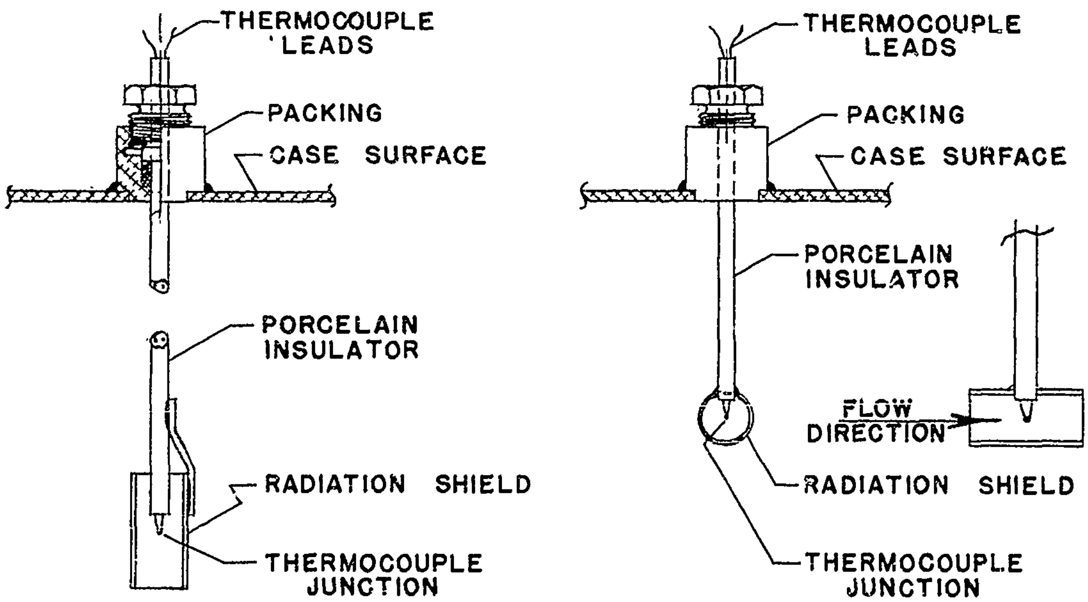


Figure III-4. Shielded Thermocouple Probes for Measurement of Internal Air Temperature if electrically well-insulated leads are used. The leads may be attached on the nearest available surface, while the end of the leads having the junction is merely bent out a distance of 1/4 to 1/2 inch, so as to be immersed in the surrounding air. This method is also quite satisfactory in the measurement of environmental air temperatures, providing radiation shielding is not necessary. However, should shielding be required, installation methods similar to those inside the equipment must be used for locating thermocouples to measure environmental air temperature.

When internal temperature measurements are required within a metallic solid section at a depth of more than 1/4 inch, it is usually necessary to insulate the thermocouple leads by means of a small porcelain tube, as described above. The tube is inserted in a hole of slightly larger diameter and cemented in place. The use of a porcelain tube is unnecessary and a smaller hole may be placed in the metallic body if a cement of good electrical insulating properties is used for installation of the thermocouple. Most dental cements are suitable for this purpose because they are also stable under high temperature conditions, and do not shrink. Internal thermocouple installation in plastics or other insulators is considerably facilitated because no insulating materials other than that applied to the wires for the purpose of insulating them from each other are necessary.

The problem of bringing thermocouple leads from within the case of a closed electronic unit to the outside deserves some attention. The application of pressure-tight connectors is possible, but rather complicated, particularly where a unit may have a large number of internal temperature measuring stations. It should also be observed that the connector jacks may have to be of the same materials as the thermocouple wires where a temperature difference of 20° to 30°C exists between the interior of the unit and its environment. There should be practically no temperature difference between the two points of attachment of the thermocouple wire to the dissimilar material of the connector inside and outside the unit, since such difference in temperature would result in a net thermoelectric effect which would produce an e.m.f. This contact e.m.f. would add to or subtract from the e.m.f. available for the primary temperature indication, depending on the nature of the thermocouple wire and the metal of the connector, and would, therefore, cause an erroneous determination of the internal temperature. The existence of a temperature difference is determined by the difference between internal and external temperatures, the internal length of the individual jack, and the magnitude of the cross-sectional area of the metallic part of the connector. If the latter is great enough, corresponding to individual jacks of about 1/16-inch diameter, 1/4 inch or shorter internally, temperature differences of 30° to 40°C between the internal and the external atmosphere of units can be accommodated without the use of connectors made of the same materials as the thermocouple wires.

A substitute method for bringing out a large number of thermocouple leads from closed units is by means of a flanged opening covered with a plate having a circumference great enough that all wires can be placed alongside each other radially outward and spaced about 1/8 inch apart. This method is shown in Figure III-5. The wires are placed in the arc segments between the hold-down bolts and between two flat rubber ring-gaskets, about 1/2 inch wide. Tightening of the bolts seals the flange surfaces and the wires between the two gaskets. The same principle, as shown in Figure III-5, can also be used for a smaller number of wires using the simplified installation shown in

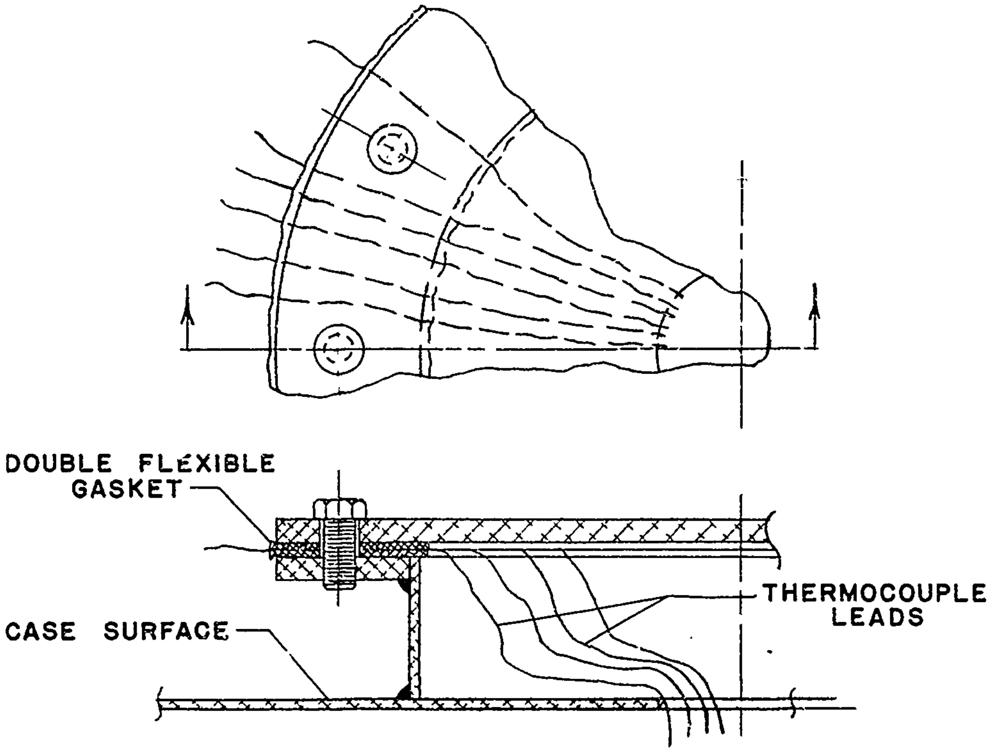


Figure III-5. Method for Connecting to Thermocouples in Pressurized Units

Figure III-6. Satisfactory sealing of 30-gage enameled wire between gaskets 1/16 inch thick can be obtained to hold air pressure differentials of 100 pounds per square inch.

6. Instruments for Thermocouples

There are a number of standard methods by which the thermal e.m.f. of thermocouples can be measured and interpreted in terms of a value on a temperature scale. The most convenient of all, for purposes of temperature exploration in electronic equipment, is the potentiometer method. A variety of instruments operating on this principle are commercially available. They differ in sensitivity, ease of operation, and presentation of data. Precision instruments are not required for this work since an accuracy of 1° to 2°C in the obtained indication is adequate. This is obtainable with all industrial instruments. In addition, such instruments, calibrated in degrees centigrade, are cold-junction compensated. In other words, thermocouple leads from one junction alone if connected to such instruments will cause a true temperature indication. In contrast, precision instruments are not compensated. They require that a reference junction of known temperature, such as

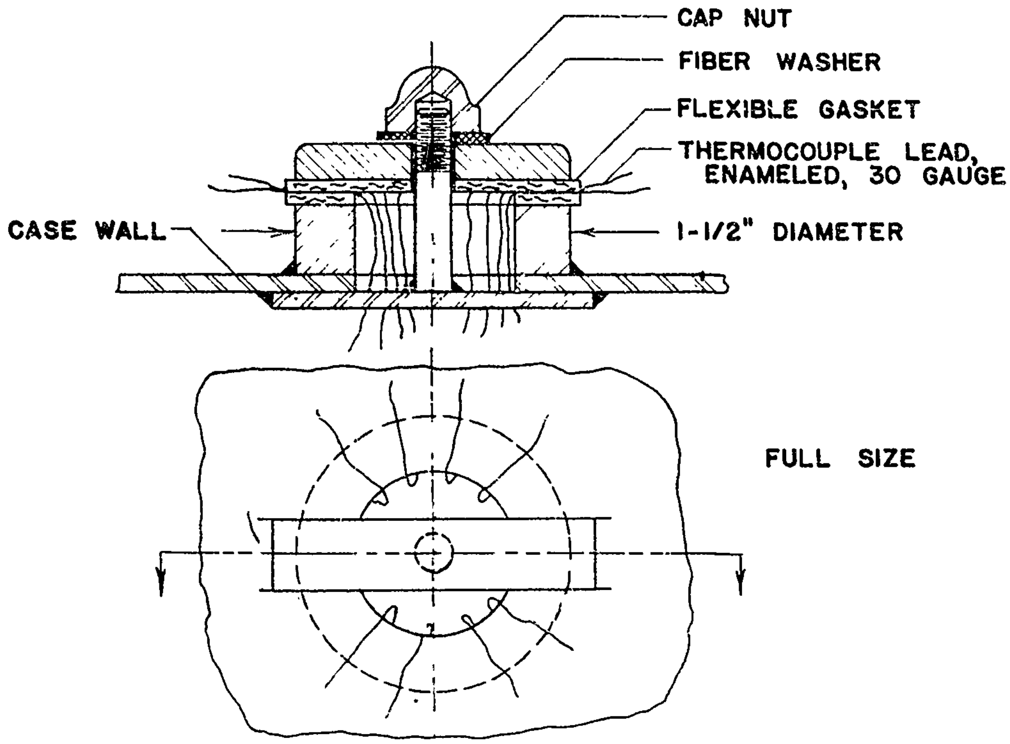


Figure III-6. Simplified Method for Thermocouple Connection in Pressurized Units

is obtained by submersion in ice, is incorporated in the thermocouple circuit. This method is more accurate, but involves some additional complication and work which is not warranted.

For the exploration of small units where the measurement of only 12 to 24 different temperatures may be necessary, a small portable potentiometer with manual standardization and balancing at each temperature is adequate. One type of this instrument is shown in Figure III-7. It has to be used in conjunction with a rotary selector switch to which all thermocouple leads are connected. The switch is connected to the instrument by a single set of leads. By its use, each thermocouple can be successively connected to the instrument and its temperature indication determined by manually balancing the thermocouple e.m.f. against a standard e.m.f. supplied by a battery across a variable resistor. The latter e.m.f. is standardized frequently by comparison with a standard cell contained in the instrument.

The use of a portable potentiometer may be somewhat slow and fatigueing when a large number of points are to be read in each set of data. For this purpose an indicating potentiometer is more suitable. This type of instrument is electrically operated and balances automatically to give a temperature indication. Standardization is usually manual. Both mechanical and electronic types of indicating potentiometers are commercially available. The latter is preferable because it balances more rapidly so that a complete

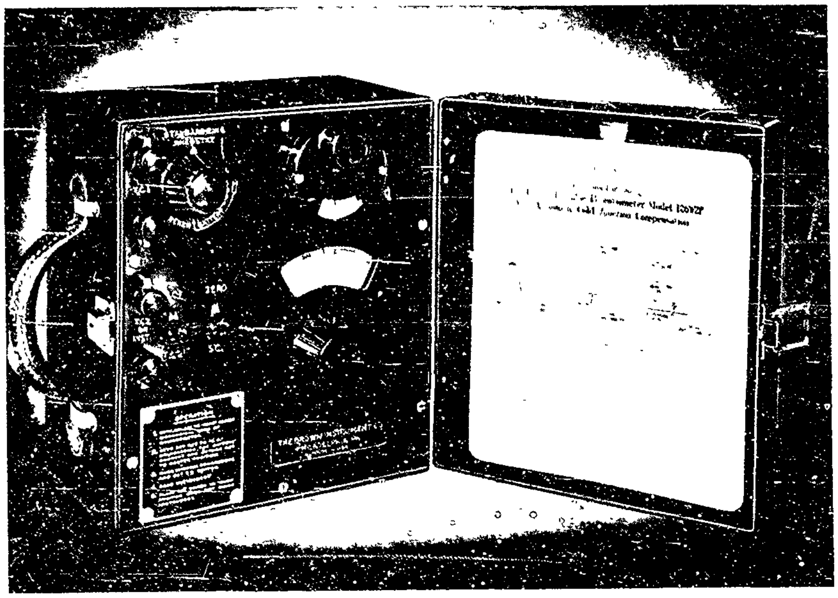


Figure III-7. Portable Indicating Potentiometer
(Brown Instrument Company)

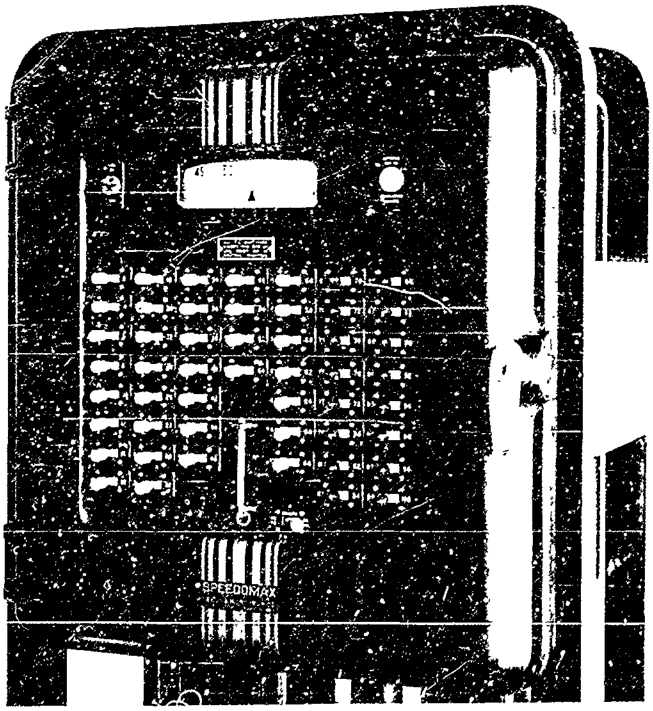


Figure III-8. High Speed Indicating Potentiometer with 100
Integral Selector Switches (Leeds and Northrup Company)

set of measurements can be obtained much more quickly. Separate rotary selector switches can be used with such instruments. However, some instruments are equipped with switches for up to 126 points. This is particularly valuable in connection with electronic instruments because the reading of temperatures is thus speeded up to the greatest extent. An instrument of this type is shown in Figure III-8.

If temperature-time relationships are to be established, the instruments described above are somewhat inconvenient because in conjunction with obtaining temperature readings with them, time data have to be obtained from a separate clock. Recording instruments which are basically the same as the indicating instruments produce a temperature-time record by printing the indication on a chart moving at a constant time-calibrated speed. They require no attendance because they are also standardized automatically. They contain an automatically operated switching mechanism which replaces the manually operated selector switches used with indicating instruments. Standard instruments are built to record up to 16 different temperatures per minute. A typical strip-chart recorder is shown in Figure III-9. The number of temperature readings to be recorded can be increased to 160 points in about 6 minutes by combining a standard recorder with an automatic switching unit which can be used so that either all 160 points are recorded in succession or any group of 16 points, or any single point may be selected. One industrial version of the instrument is shown in Figure III-10. Such relatively expensive instrumentation is not particularly warranted in laboratory test work on electronic units when operation under steady-state conditions is to be investigated. However, it is almost mandatory where transient operating conditions are being studied. In flight testing, recording instruments are essential if determination of transient conditions is of importance or where room for an operator is not available.

7. Checking for Operation within Specified Maximum Temperatures. Lacquers

In the study of an electronic unit for evaluation purposes, with the aim of producing data for the determination of performance under other operating conditions, and for possible modifications, the use of thermocouples for temperature measurement is necessary so as to ascertain successively the effects of different operating conditions without disassembly of the unit. However, there are circumstances under which only compliance with certain temperature limitations must be ascertained. Under such circumstances the probable locations of critical-temperature components or other surfaces are known. A convenient means of ascertaining whether limiting temperatures are reached is by use of lacquers, each rated at a specific melting point. They are commercially available as Tempilaq (Reference 5). A small deposit of such a material with a melting point equal to the limiting temperature on the particular component, case, chassis, or internal equipment atmosphere will indicate, when inspected after operation, whether the limiting temperature was reached. This is convenient in single tests in flight, or when instrumentation is difficult to apply. However, a principal disadvantage of the method is that it will yield no data on the quality of cooling design because it can only show whether the operating temperatures were below or in excess of the limiting values. In the latter case, subsequent measurements with thermocouples are usually necessary to ascertain the extent to which a limiting temperature is exceeded and to furnish the necessary data for redesign.

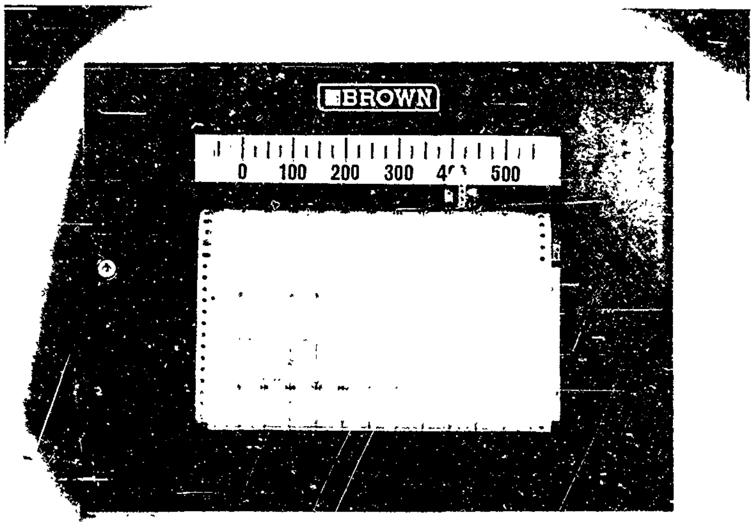


Figure III-9. Automatic Strip Chart Recorder
(Brown Instrument Company)

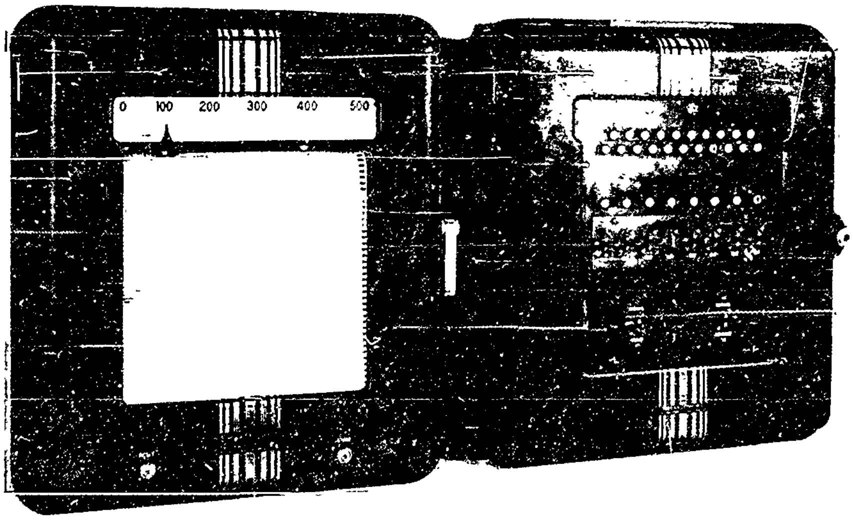


Figure III-10. High-Speed Multi-Bank Recorder for
160 Thermocouples (Leeds and Northrup Company)

Pressure Measurements

1. Operational Pressure Level

Thermal evaluation of electronic equipment under any test condition necessitates knowledge of environmental and internal air pressures associated with the operation of the equipment. Regardless of the type of equipment and the test conditions, the environmental pressure, normally represented by the barometric pressure, must be measured to provide a datum for all other pressures to be determined. The environmental pressure is also needed to define fully the physical properties of the atmosphere in which the equipment is operating. Temperature measurements on equipment cases and heat exchange surfaces can only be properly interpreted in correlating them with atmospheric pressures. Similarly, component temperatures of open units require correlation with atmospheric pressure, particularly when free convection is the principal mode of heat transfer utilized.

The internal pressure level of closed units must be defined to describe the heat transfer conditions surrounding components, regardless of the state of motion or temperature of the internal atmosphere of the equipment. Measurement of internal pressure is made in reference to the atmospheric pressure of the test environment. Therefore, if the pressure-measuring element is surrounded by the test atmosphere and open thereto, a differential pressure is determined which, when added to the barometric pressure, gives the value of the absolute pressure within the closed equipment.

If the only purpose of connecting a primary pressure-measuring device to the equipment case is to determine the internal pressure level, the design of the connection itself is not critical. However, it is always desirable to install the pressure tap in such a way that pressure indications are not affected by rates and direction of internal air motion. The location in the case should be so chosen that the air flow at that particular point is parallel to the wall of the case and is not deviated by upstream or downstream projections in the immediate vicinity. Slight inclination of the air flow direction to the wall can be tolerated if the tap opening in the case surface is small, preferably having a diameter not greater than about 1/16 inch. Good methods for installation of single taps in the wall of a case or a ducted passage are shown in Figure III-11. It is also necessary to prevent projecting burrs by filing the opening flush with the inner wall. This precaution is not effective nor sufficient when high internal velocity exists, should the tap be installed at a location where the velocity is directed normally to the wall surface. Such conditions may result from baffles, turns in the air flow passage, and components in the immediate vicinity of the tap. The error thus produced in the value of the absolute operating pressure level in bench tests may be as high as 1/4 inch mercury which would usually be less than one per cent of the total value. The magnitude of the error would be similar under other test conditions.

2. Static Pressure Differentials

Measurement of static pressure differentials is essential for thermal evaluation of all electronic equipment cooled by internal or external forced air convection. The purpose may be the determination of pressure drop across internal passages containing components and internal heat exchange surfaces, to indicate, together with the rate of air flow, the minimum required

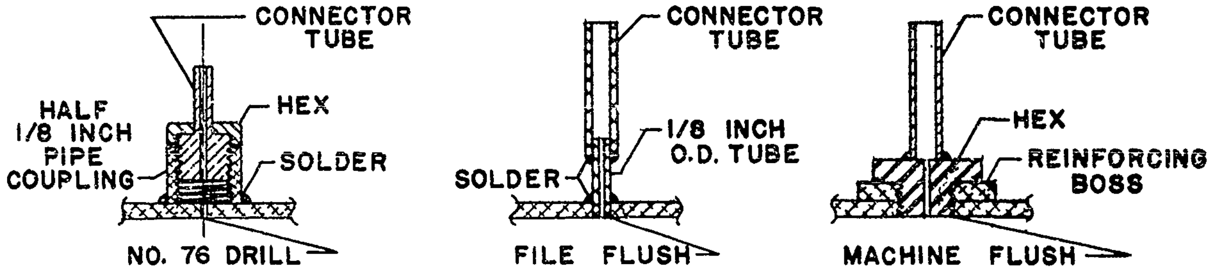


Figure III-11. Methods for the Installation of Pressure Taps in Case Walls and Ducts

total internal cooling power. Or, the purpose may be to find the pressure drop across an external heat exchanger to permit calculation of the external cooling power, or establish the requirements for the selection of a blower. Static pressure differentials may also be measu d in the study of effectiveness of cooling design. For that purpose, sections in the internal passages of the equipment should be compared for the respective ratios of required internal power to heat dissipation in each.

The general equations for air flow through a system with heat transfer are given and discussed in Appendix II. It is apparent from the form of the equations that the exact interpretation of measured static pressure differentials is possible only if the flow rate, cross-section of the flow area at both measuring stations and the mean air temperature at both stations are known. In the evaluation of total internal system resistance, air inlet and exit temperatures at the circulating blower are essentially the same. However, it is unlikely that the duct passages for static pressure measurements closest to inlet and exit of the blower would have the same cross-sectional areas. Their dimensions must be ascertained to evaluate the total pressure produced by the blower.

The flow cross-section of a heat exchanger passage is usually constant, but temperature change of the air occurs. Therefore, measurement must be made of the temperatures at inlet and exit which may in many instances require evaluation of air flow rate at entrance and exit. Ducted passages with components installed in them have usually irregular flow cross-sections. If they contain heat-dissipating components, the air temperature at inlet and exit of such passages would not be the same. However, for many equipments the temperature rise may be rather small. Therefore, for qualitative measurements and for purpose of evaluation, the temperature rise of the air may be neglected. In installing the pressure taps at such stations, it is best to attempt to choose sections of equal area. Since this is not always possible, care must be taken in the interpretation of the data to correct for the inequality of the flow areas at the pressure stations. It is also desirable to use for reference stations such locations where the vicinity of components would not irregularly affect the pressure distribution along the walls of the passage. In any case, it would be necessary to obtain the static pressure distribution along the wall surface at any reference section at more than one location. Therefore, pressure taps such as shown in Figure III-11 must be installed in the same plane normal to the principal direction of air flow at several points on the periphery of the ducted passage. They may be interconnected so as to yield one single static pressure measurement, or the pressures may be measured individually and would then be averaged to determine the representative static pressure for the particular station.

For the exploration of the static pressure within an equipment, the measurement of individual static pressures at various stations is not satisfactory. The principal disadvantage of using this method is that extremely accurate individual measurements must be made to insure good accuracy for the pressure gradients determined from them. Since the latter are relatively small, it is preferable to make measurements of differential pressures. Therefore, individual pressure taps at each station should be inter-connected into a common header, generally referred to as a piezometer ring. Thus, by connecting the pressure-measuring device differentially between two piezometer rings, an accurate measurement of the difference in static pressure between two stations can be obtained readily. Furthermore, it is also possible to determine the actual static pressure at any section if the static pressure has been measured at one section and the differential between the latter and the former is known.

3. Instrumentation

For the determination of the operational pressure level of electronic equipment the measurement of atmospheric pressure is a principal requirement. Instruments used for that purpose fall into the general classification of barometric devices. In the laboratory it may be a mercury column barometer, when high accuracy is required, or an aneroid barometer which usually would be satisfactory. In general, an accuracy to 0.01 inch mercury or 0.25 millimeter mercury is sufficient for general bench test work. Where altitude conditions are simulated by means of an altitude chamber, the pressure-measuring device which would determine the chamber's atmospheric pressure is usually incorporated in the design of the chamber. Such devices normally are calibrated in terms of feet of altitude and permit the experimenter to adjust the operating conditions directly from the reading.

For pressure explorations within equipment cases, instruments are required which are capable of measuring relatively small pressures. They are generally classified as manometers and use fluids of desirable specific gravity to balance the pressures to be measured. The pressure reading is expressed in terms of the length of the fluid column, which may be in inches or millimeters, as desired. Commonly used manometers fluids are mercury, water, light mineral oil, and alcohol. In most types of manometers these fluids may be used interchangeably since their scales are calibrated in inches of fluid, in general. However, there are some types which require the use of a specific fluid since the scale is calibrated in terms of another fluid and therefore, the divisions of scale length are changed by the ratio of specific gravity of the scale-calibration fluid to the manometer fluid. For example, the scale of a manometer may indicate a pressure reading in inches of water but may use a light mineral oil as gage fluid. If the specific gravity of the oil is 0.84, the length of the vertical scale, indicated to correspond to one inch of water is 1/0.84 = 1.19 inches.

When water is used as a manometer fluid, care must be taken that capillary effects are eliminated. This is aided by the addition of a wetting agent such as Aerosol. In general, the use of commercial-type instruments is recommended for test apparatus since in their design the necessary precautions have been taken to eliminate capillary effects of prohibitive magnitude which may result from improper choice of tubing size and non-uniformity in the bore of the tubing. Simple manometers can be made up in the laboratory, if desired. However, factors which may affect the accuracy must not be overlooked. Manometric relationships are discussed in Appendix III.

The U-tube manometer is the simplest device of this type. One of its commercial versions is shown in Figure III-12. It has a tube large enough so that various fluids can be utilized with it. In addition, it usually has two blow-out wells, one at the top of each tube inlet, which prevent the overflow of the manometer fluid into the lines of flexible tubing connecting the manometer to the equipment, should the pressure range of the instrument be exceeded. The U-tube manometer can only be employed to measure pressures sufficiently large so that its maximum accuracy of about 0.05 inch does not represent an appreciable percentage of the total pressure. Its use lacks some convenience in that it is necessary to read both legs of the tube, rather than one, in spite of the fact that usually at zero pressure differential the two fluid columns are to be leveled at the zero-mark of the manometer's scale, and the rise and depression of the fluid in both legs would be expected to be equal. This precaution is necessary because the glass tubing may not be of entirely uniform cross-section. Like all other manometric devices, it can be connected either differentially, or with one leg to the source of pressure and the other vented to the atmosphere. In the latter case, the pressure so measured would be referred to as gage pressure.

A more convenient device is the well-type manometer. A commercial device of this type is shown in Figure III-13. It only requires the reading of the fluid level in a single tube on a calibrated scale which may take into account the difference in cross-sectional area between the tube and the well. This is necessary because, while initially the fluid level in the tube and the well are equal when the fluid level is at zero, the level in the well is depressed slightly below the zero-mark as the fluid rises in the tube. This must be compensated for in the calibration of the scale. The accuracy of the well-type manometer is about the same as that of the U-tube.

A different type of the well-type manometer is the inclined-tube manometer, such as shown in Figure III-14. This instrument is usually made in ranges in order of 1/2 to 3 inches of water and can be read to an accuracy of 0.01 inch or less. By use of the inclined tube the differential of the fluid column is magnified. The scale of the manometer is usually calibrated directly in inches of manometer fluid or in inches of a standard fluid. Thus, the manometer may use a light oil as a fluid, though the scale has been calibrated in inches of water, by taking into account the specific gravity of the oil. Because the magnification factor of the scale is only a function of the inclination of the tube, it is necessary that the inclination of the tube be accurately reproduced under all installation conditions. This is done by means of a spirit level incorporated in the instrument.

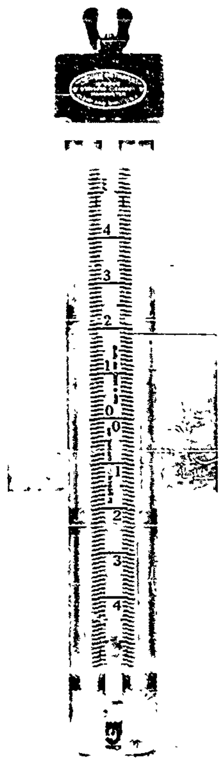


Figure III-12. U-Tube Manometer (Meriam Instrument Company)

Figure III-13. Well-Type Single Tube Manometer (Meriam Instrument Company)

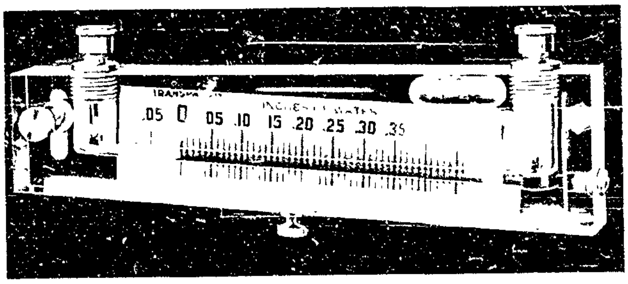


Figure III-14. Inclined-Tube Manometer
(F. W. Dwyer Manufacturing Company)

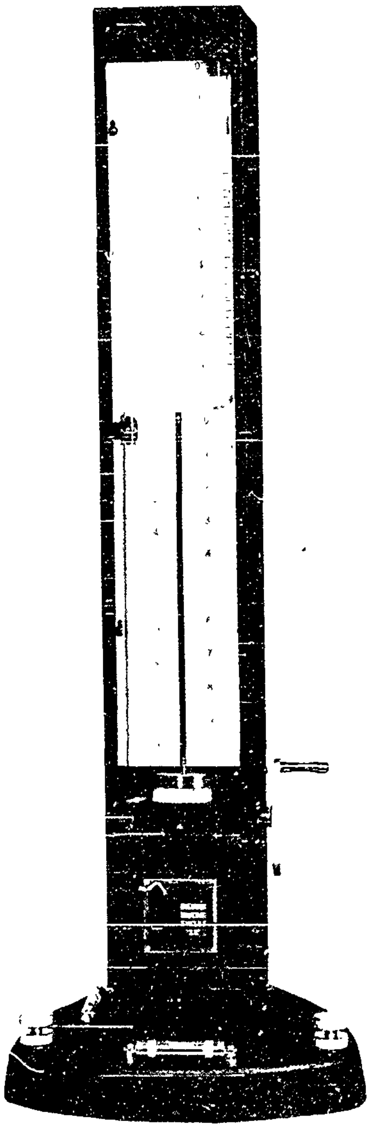


Figure III-15. Micro-Manometer
(Meriam Instrument Company)

For the measurement of large or small differential pressures to an accuracy of 0.001 inch, instruments classified as micro-manometers are commercially available. One type of this instrument is shown in Figure III-15. It combines the advantages of the well-type single-tube manometer with that of the inclined-tube manometer. When the manometer registers zero pressure, the fluid column must be located at an etched mark on the inclined central portion of the tube. If a pressure is applied to the top of the well, the fluid column moves up above the etched mark. The well of the manometer, now still behind the zero-mark, can be lowered to bring the top of the fluid column back to the etched mark. A pointer serves to indicate the motion of the well downward on the scale. The exact position of the pointer can be determined from the markings on the crank disc on the lower right which, by means of a vernier scale, permits reading of the manometer to 0.001 inch water. Similar devices are available which represent modifications of the various types. Some micro-manometers are designed to overcome in part the disadvantage that time-consuming balancing is required for each reading. They use a reversible electric-motor drive for leveling of the well and a revolution counter, geared to the vertical calibrated screw carrying the well, to indicate the pressure.

Air Flow Measurements

For the complete thermal evaluation of units cooled internally and/or externally by forced air convection, measurement of the air flow quantity is necessary. The information thus obtained serves as basis for evaluation of the heat exchange effectiveness of the system as well as for the determination of the flow and the power requirements under different operational conditions. In addition, as pointed out in the discussion of air temperature measurements on page 21, the information is necessary for heat balance calculations which result in the determination of heat quantities otherwise unaccounted for. General equations for air flow through a system with heat dissipation are given and discussed in Appendix II.

1. External Cooling Air Flow

Primary flow-metering elements such as venturi meters, nozzles, and orifices cause pressure losses by virtue of their own characteristics and because they must be installed in duct runs of appreciable length which also offer resistance to air flow. Blowers incorporated in electronic equipment for conveyance of cooling air usually are selected to produce only sufficient static pressure to overcome the resistance of the unit at required air flow rates. Adding to the unit's resistance that of the metering system, would in most instances reduce the blower capacity excessively and would not give correct indication of blower performance in conjunction with the electronic unit. Therefore, measurement of external cooling air flow through a unit open to the atmosphere, or over the heat exchange surfaces of a closed unit is best accomplished by means of auxiliary equipment. The principal task of this equipment is to permit air flow measurement without affecting the performance of the blower installed in the electronic unit.

The basic features of such an auxiliary equipment are shown in Figure III-16. The apparatus consists of a constant-speed centrifugal blower and a duct containing a metering section, a flow straightener, a throttling device,

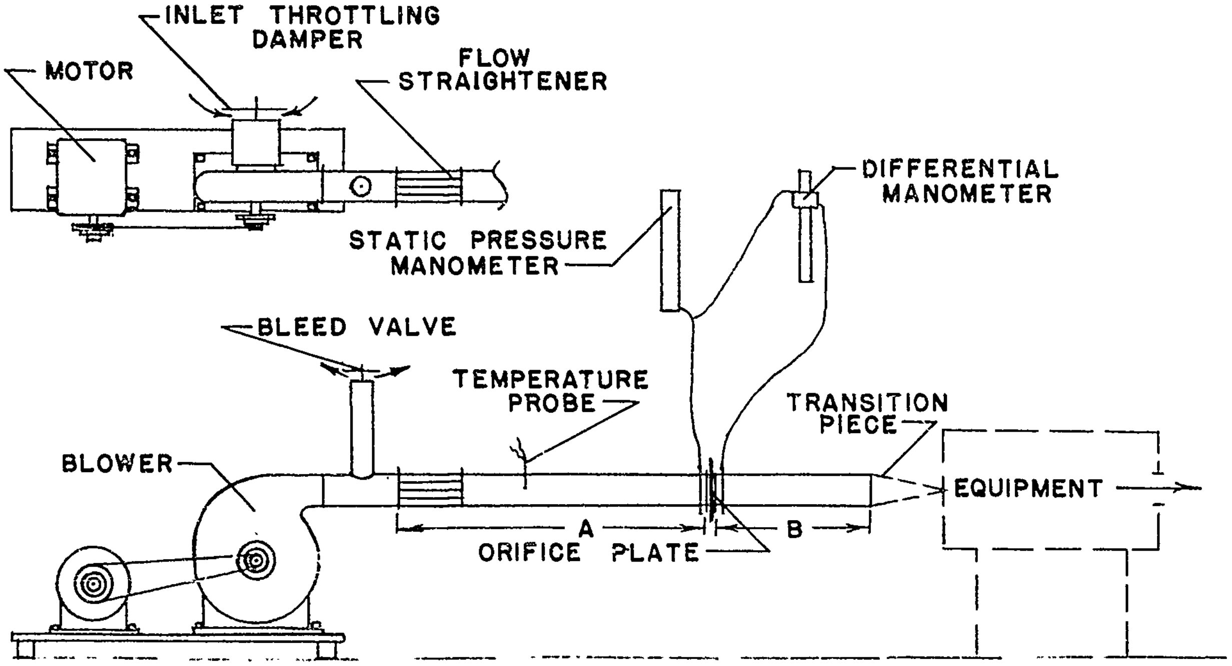
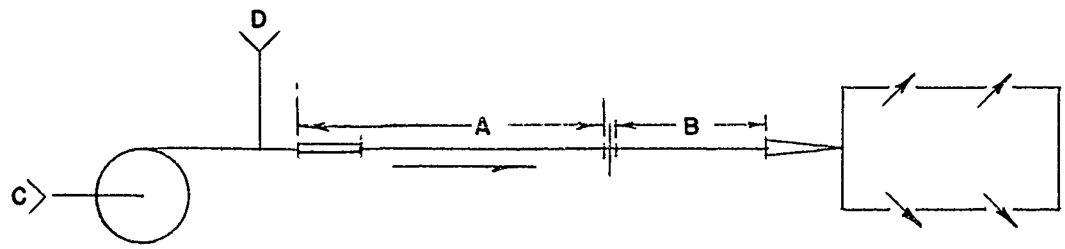


Figure III-16. Auxiliary Air Flow Test Apparatus



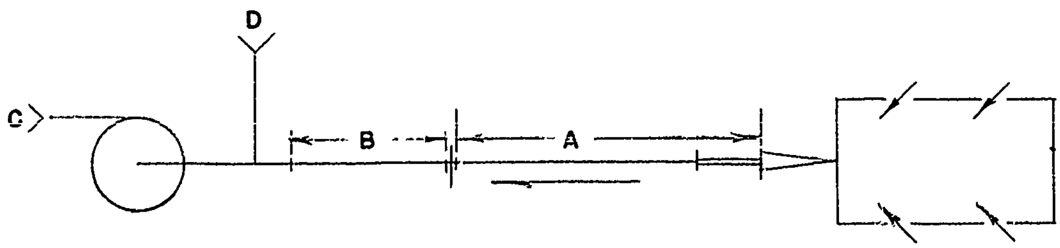


Figure III-17. Basic Schemes for Application
of Auxiliary Air Flow Apparatus

and a bleed valve. The metering section has two manometers, one for measurement of static pressure, the other for measurement of differential pressure, and provisions for the installation of a calibrated orifice plate of suitable size, depending on the range of air flow required. Orifice plates are recommended for metering purposes because they can be reproduced by simple machining processes and need not be calibrated against a standard device for the accuracy required in thermal evaluation studies. Instead, their performance can be predicted from their dimensional characteristics. The flow straightener is required to insure accuracy in meter readings by eliminating spiraling air flow which is a characteristic of single-inlet centrifugal blowers. The throttling device would best be an iris-type damper, but could also take the form of a blast-gate. Similarly, the bleed valve could either be an adjustable plate damper at the end of the bleeder pipe, or a blast-gate. Throttle and bleed valves serve to adjust the air flow and pressure level in the duct to the requirements of the electronic unit to be tested.

The structural details of one embodiment of such a flow apparatus are described in detail in Appendix IV. The specifications are complete enough to permit any organization with only modest shop facilities to assemble such an apparatus with little expense. Also, dimensions for manufacture of four orifices to cover an air flow range from about 0.01 to 0.3 pounds per second are given, together with flow charts from which air flow quantities can be read directly if the barometric pressure, the static gage pressure upstream from the orifice, the differential pressure, and the air temperature at the orifice have been determined. Measurements of these quantities and suitable instrumentation are described in sections beginning on page 21. It should be noted that usually the use of a single thermocouple probe in the duct ahead of the orifice is sufficient for determination of the air temperature.

An auxiliary air flow apparatus of the described type should be part of the permanent facilities of an electronic equipment development laboratory, since it is not only useful in thermal evaluation work of existing equipments, but can be utilized to determine required blower characteristics for cooling of new equipments under development.

For evaluation purposes, the apparatus can be installed in several ways. The two basic schemes are shown in Figure III-17. When the air intake of the electronic unit consists of a single opening to which a duct can be connected, by means of a transition piece from the circular cross-section of the duct to the shape of the opening, the apparatus should be installed as shown in Figure III-17a. Since the auxiliary blower is a constant-speed device, the bleed and throttle valves must be so adjusted that the static air pressure at some reference location within the unit's air flow passage is the same as measured when the auxiliary system is not attached to the unit and only the unit's blower is used. If this reference pressure is reproduced, the unit's blower is operating under the same conditions. Then, the auxiliary blower only serves to overcome the resistance of the auxiliary system, including that of the metering section. The type of operation of the unit's blower, whether as a discharge or a suction device, and the manner of discharging the cooling air to the atmosphere, whether through a single opening or through multiple openings, have no influence on the installation and operation of the auxiliary system.

When the air discharge from the electronic unit consists of a single opening, the apparatus may be converted readily as shown in Figure III-17b so as to be attached to that opening. In that case, the auxiliary blower operates as a suction device. The positions of the bleed and throttling valves are interchanged relative to the auxiliary blower and the duct with straightener and metering section is reversed. The operation of the auxiliary blower is controlled like in the previous arrangement by the adjustment of the two valves so as to reproduce a given reference pressure within the electronic unit. While volumetric capacity of the blower is constant, weight flow capacity is reduced since it handles heated air discharged from the unit. Therefore, if a unit happens to have inlet and outlet openings both suitable for connection of the auxiliary system, it is preferable to connect to the inlet opening as described in Figure III-17a.

2. Internal Circulating Air Flow in Closed Units

The problem of quantitatively measuring the rate of internal air circulation in closed units is considerably more complicated than that of measuring external air flow. Yet, internal air flow measurements are of great value since they make it possible to separate, by calculation, the heat transfer by forced convection between components and the case, or other heat exchange surfaces, from the heat transfer by conduction and radiation. This permits combined evaluation of the extent of utilization of all three modes of heat transfer.

In closed units which have internal air circulation by forced convection, but in random fashion, even measurement of temperature change of the air passing over the inner case surfaces is not feasible because no well-defined flow passages exist. Therefore, for such units, an attempt to measure air circulation quantitatively is of little use since it would not be possible to calculate the heat transfer by forced convection separately. However, in closed units with ducted passages and integrated heat exchanger, the temperature rise of the air passing internally through the unit can be determined by measurement of exit and inlet temperature at the heat exchanger. Knowledge of the flow rate would then permit calculation of the total internal heat dissipation by forced convection. Internal flow measurement by standard means would be most difficult and unreliable, because in the congested conditions which would exist no suitable locations would be available for installation of flow measuring devices. Exploratory devices such as impact tubes, as discussed on page 45 would not be satisfactory for quantitative determinations and should not be used for this purpose. The use of the auxiliary measuring apparatus, as discussed in the previous section, would not be practical for internal flow measurements because it could not be installed within the internal configuration of most electronic units.

It is possible to measure internal flow rates indirectly by means of measuring the static pressure differential between inlet and exit of the blower, as well as the air's absolute pressure and temperature at the blower's inlet, when the electronic unit is operating under the desired conditions. The methods for making these measurements are discussed in previous sections. The rotational operating speed of the blower must also be known. The data so collected can be used to calculate the air flow rate from the characteristics of the blower, if the latter are known. The methods of calculation are given in Appendix V, concerned with blower test methods. However, where the evaluation of a completed equipment, as furnished, must be undertaken, it is possible that the blower characteristics are not known. In that case, calibration of the blower can be undertaken without too much complication if the blower can be removed readily from the unit. The metering section and throttle of the flow apparatus described in Appendix IV can also be used for the purpose of blower calibration. The procedures for the blower calibration and the interpretation of the data for determination of internal air flow rates are described in Appendix V.

Velocity Measurements

In the thermal evaluation of electronic equipment with internal forced air circulation, occasions arise for exploration of the velocity distribution at various sections. Therefore, a requirement exists for the measurement of local velocities. One case like this is mentioned on pages 21 and 22 in the discussion of air temperature measurement in a passage of non-uniform velocity distribution. The velocity measurements needed for such purposes are usually only of qualitative nature and need not yield absolute values of velocity. It is sufficient to compare values measured at various points in the same section so as to determine their relative magnitudes. Their absolute value can also be readily determined if the total quantity of air flow is known. Then, the sum of the products of each local velocity function times the area for which it is representative, divided by the total flow volume, determines a proportionality factor which when applied to each individual velocity function will give the true velocity at any particular point of measurement. This procedure is illustrated in Example A-II-3 in Appendix II.

Exploration of local velocity distributions is of principal importance in equipments which have relatively high internal air velocities. Those using low air velocities of less than 15 feet per second will exhibit less drastic effects on component temperatures by change of velocity distributions and, therefore, might not warrant study from that viewpoint. The instrumentation required for the quantitative exploration of low internal air velocities is very complicated and costly and is considered to be unwarranted for the purpose of thermal evaluation of electronic equipment.

A qualitative indication of the distribution of air flow and local velocities within a unit can be obtained by various flow visualization techniques. For that purpose, it is necessary to replace the case of the unit with one of equal configuration, but constructed of a transparent material. The flow pattern can then be made apparent by injecting into the air stream at the unit's inlet a substance producing a suspension of fine solids. Chemicals such as titanium or silicon tetra-chloride are suitable, but require some experience in their use. Talcum powder or finely divided magnesium silicate may also be used, but may require frequent cleaning of the inspection windows in the case to which the powder would adhere because of electrostatic charge.

The relative magnitudes of local velocities within a unit having a transparent case can also be evaluated conveniently by means of streamers made of fine silk or nylon thread. They are placed in the air flow passages of the unit, each so attached that their inclinations to the principal direction of air flow can be observed. For example, if the flow is predominantly horizontal and across vertical component surfaces, the greatest velocities occur where the streamers assume a nearly horizontal position, with their free ends pointing in the direction of flow. A streamer not displaced from its normally vertical position would indicate a local dead spot in the flow distribution.

Distributions of air velocities of 15 feet per second, or greater, can be determined with a relatively simple device. The total-pressure tube, as shown in Figure III-18 is not only suitable for the determination of the magnitude of air velocities but also their direction. As shown in Figure III-18, the tube should be made of small seamless tubing so as to permit its insertion in locations where little space is available. It has the ability to register a total pressure which is the sum of the static pressure at the particular section plus an impact pressure created by the velocity of the air at the particular location at which the tube is inserted. The tube may be turned about its axis so as to change the relative position of the opening with respect to the direction of air flow. ~~Its proper position is~~ found when a maximum pressure is registered since it can then be assumed that the opening receives direct impingement of air flow. The total gage pressure so produced may be indicated by means of a manometer. Separate measurement of the total pressure and the static pressure at the particular section, is usually not desirable since their difference, representing the velocity pressure is quite small. Hence, a slight error in the measurement of either of the two pressures would give a large percentage error in the determination of the velocity pressure. Consequently, it is desirable to connect the total pressure tube differentially with a piezometer ring indicating the average static pressure at the particular section. Then, the manometer will indicate the velocity pressure only. The latter is usually quite small, less than 0.5 inch water, and requires for measurement either an inclined-tube manometer or a micromanometer. The manometer connections are the same as those shown in Figure A-III-3, page 350.

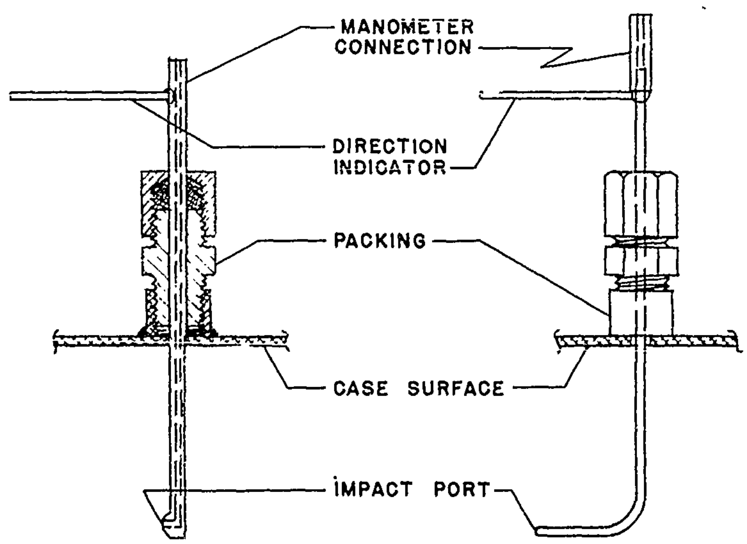


Figure III-18. Total Pressure Tubes

The general air flow equation discussed in Appendix II contains a term for the velocity pressure which is shown to be proportional to the square of the air velocity. Therefore, a differential pressure measurement performed by means of an impact tube will give a pressure indication whose square root will be proportional to the local velocity at the point where the tube is installed. The velocity function thus obtained is referred to above. The absolute value of the local velocity can also be calculated readily from the pressure indication. For that purpose, it is also necessary to know the static pressure at the section so as to be able to calculate the air density. This is shown in Example A-II-1 in Appendix II.

References

1. American Institute of Physics, Temperature, Its Measurement and Control in Science and Industry. Reinhold Publishing Corporation, New York, N. Y., 1941.

2. American Society of Mechanical Engineers, Power Test Codes, Series 1923, Instruments and Apparatus, Part 2, Pressure Measurement. American Society of Mechanical Engineers, New York, N. Y.

3. American Society of Mechanical Engineers, Fluid Meters, Their Theory and Application. Fourth Edition, 1937, American Society of Mechanical Engineers, New York, N. Y.

4. Sauereison Cements Company, Pittsburgh 15, Pa.

5. Tempil Corporation, New York, N. Y.

CHAPTER IV

TEST PROCEDURES FOR EQUIPMENT DESIGNED TO OPERATE IN THE STEADY STATE

In this chapter, suggested procedures for procuring experimental thermal evaluation data are discussed. Bench test procedures applicable to all equipment classifications are given. Altitude-chamber test procedures are described only for equipments which cannot be analyzed on the basis of bench tests, or for which verification of predicted thermal performance is required. Since flight testing is not considered to be a method of obtaining data for preliminary thermal evaluation, but is more applicable to final acceptance testing, detailed descriptions of its procedures are not included.

Bench Tests

In arranging electronic equipment for bench testing care must be taken that the surrounding radiant and convective conditions are well defined in order to make computational analysis of the data and extrapolation to other environmental conditions possible. Analogous to the use of a screen room for electrical measurements, it is necessary that the bench test area be so located or screened to provide a test region in which the air is not disturbed by external influences and where definition of the radiation environment is not complicated by cold outside walls, cold glass surfaces, or hot room heating surfaces. It is well to use screens to protect the electronic units from drafts and air movements caused by building heating or cooling systems and to provide for surrounding surfaces of known emissivities. The screens are best located about five feet from the test stand. This spacing prevents the screens from affecting the free convective heat transfer from the units under test and permits the use of screens of any consistent material and surface condition. With such an arrangement, the radiant heat transfer is not affected by the characteristics of the surrounding receiving surfaces, since sufficient distance is maintained between them and the radiating surfaces of the electronic units.

Several precautions must be observed in mounting electronic units for bench testing in order that the cooling performance for a particular unit is only negligibly affected by adjacent units. Units which are cooled externally by free convection and radiation must be located at least two feet apart and should be mounted on racks, one foot or more above the table top. The units must not be stacked one above the other, since this arrangement would have a sizeable effect on the convective heat transfer. Units employing forced convection with an external exhaust must be so located or baffled that the exhausted air will not affect other units being tested.

If the relative mounting positions have been established by mocking up the component units of an equipment to simulate the positions of the final installation, it is possible to determine the effects of the installation environment on the thermal operation. For evaluation of radiation effects it is essential that the materials composing the compartments of the test mock-up duplicate exactly those of the final installation. The test data obtained under those conditions, when compared to the data obtained in the operation of each individual unit, indicates the magnitude of the environmental effects of the installation.

A considerable number of measurements is required to obtain complete bench test data. Therefore, the best practice is to test at one time only one or two of the individually cased units composing an equipment. In addition, the purpose of the tests must be taken into consideration since it affects the type and quantity of measurements to be taken. If the sole purpose is to obtain data from which the operational thermal conditions of the equipment in a particular aircraft installation are to be calculated, the number of necessary measurements is smaller than when the effectiveness of the thermal design is to be studied. In the former case, when no modifications are anticipated, it is mainly necessary to determine whether all components of the unit would operate without any one of them reaching its limiting temperature when specified operational conditions are maintained. Therefore, in bench tests conducted for that purpose, component temperature measurements may be limited to the most critical components, except when all components are in direct contact with external cooling air ~~supplied by a~~ blower. On the other hand, when the effectiveness or quality of the thermal design is to be determined, the objective would usually be to accumulate sufficient data which would provide the necessary information for modification of the unit, either to make it capable of withstanding more severe operating conditions, or to reduce its size or cooling power requirement. This would usually include a rather complete survey of component temperatures.

Since the data discussed in this chapter are to be used in the evaluation of steady-state operation, it is important that all measurements be made under equilibrium conditions. It is understood, therefore, that all test runs should be of sufficient duration to insure thermal equilibrium. Thus, measurements of the same quantity would give values which are constant with time within the limits of experimental accuracy.

Techniques for measurements discussed in the following sections are described in Chapter III. The measurements are classified in accordance with their purposes, whether for determination of operational thermal conditions, or effectiveness of thermal design. In addition, the measurements are discussed from the standpoint of their applicability to all types of units, related groups of units, and individual units.

1. Measurements Required for all Types of Units for Prediction of Operational Thermal Conditions

There are certain measurements which must be taken during the bench test of any unit to define the test conditions and the overall characteristics of the unit.

The test conditions are defined by the following measurements:

a. Room Air Temperature. The conditions in close proximity to the unit under test are most important. Therefore, the measurements should be made within a few inches of the unit with thermocouples or a thermometer. In any event, care should be taken that the air temperature reading is not affected by radiant heat exchange between the measuring element and the unit, or other surrounding surfaces at appreciably different temperature than the air. Therefore, shielding of the thermometric element so as to permit free convective air circulation through the shield and over the thermometric element is usually necessary.

b. Environmental Surface Temperatures. Surfaces surrounding the test stand are in radiant heat exchange with the outer surfaces of the test unit. Particularly for units which are dependent on external heat dissipation by free convection and radiation the definition of the external radiation environment is very important. If the test area is screened, as described on page 48, the surrounding surfaces may be assumed to be at room air temperature. However, if for any reason the test area is open and is exposed to adjacent cold walls, glass surfaces, or building heating surfaces, the temperature of such surfaces should be determined by installation of thermocouples on them in accordance with methods described on pages 26 to 28 in Chapter III. If such surfaces are at low temperature, the thermocouples may be taped to them. On heated surfaces a heat resisting cement may be used. On metallic surfaces, the thermocouples may be installed by peening of the metal around the thermocouple's junction.

In addition to measuring the temperatures of the surrounding surfaces, their surface characteristics in respect to material and finish should be identified as completely as possible. This furnishes the basis for the selection of representative values of thermal emissivities which must be known for calculation of radiant heat exchange.

c. Barometric Pressure. The third variable defining the test environment is the barometric pressure. Its value has considerable bearing on the convective conditions and should be measured, not necessarily near the test stand but somewhere in the laboratory where the test is being performed.

The overall characteristics of any unit can be defined by the following measurements:

d. Heat Dissipation of Entire Unit. Knowledge of the total heat dissipation is necessary for supplementation and verification of heat transfer calculations made in the analysis of the test data. Measurement of total heat dissipation is of electrical nature and should be made in accordance with standard methods which are not considered within the scope of this manual. The measurement of heat dissipation of individual components is necessary only in closed vented units if only data for the determination of operational thermal conditions are required.

e. Outer Case Temperature. In all types of units, the outer case serves as a heat exchange surface which may vary greatly in its importance. Nevertheless, it is necessary that the surface temperature of the case, represented by a single temperature, or by the combination of several measurements, be determined. The temperature of all case surfaces should be measured at points about six inches apart with thermocouples cemented to, soldered to, or peened into the metal of the case. It may also be important to measure case and chassis temperatures when high-temperature components are installed in the vicinity, and when it is indicated that considerable heat transfer may occur locally by conduction and/or radiation to the case. The case surface must also be described with respect to its external finish so that proper known values of emissivity can be selected in the calculation of radiant heat exchange between the case and its environment.

For the sole purpose of establishing sufficient test data for the calculation of operational thermal conditions under aircraft installation conditions and to determine the unit's suitability for a specific application defined by the compartment temperature, installation environment, and operational characteristics of the aircraft, all types of units require the following limited measurements:

f. Component Surface Temperatures. From inspection of the unit and from knowledge of characteristics of individual components, certain components can be selected which are most likely to reach their limiting temperature under unfavorable conditions for heat dissipation. Temperature measurements must be made principally on two types of components. They are (1) high-output components singly, or surrounded by other high-temperature components, or located in a congested environment where access of the circulating internal cooling atmosphere appears to be inhibited, or individually shielded; (2) low-output or non-heat-dissipating components which are sensitive to high temperatures and appear to be exposed to unfavorable environmental conditions, resulting either from stratification of high-temperature air around the component, or heat transfer by radiation and/or conduction to the component from surrounding high-temperature components.

Component surface temperatures should be measured by use of cemented thermocouples or other techniques, as discussed on pages 26 to 28. On high-temperature components the thermocouple should be located at the anticipated hot spot. On temperature-sensitive components the thermocouple should be installed on the surface most exposed to radiant and conductive heat transfer from high-temperature components.

2. Measurements Required for All Types of Units for Evaluation of Effectiveness of Thermal Design

Considerably more extensive measurements are required when a unit is to be studied from the standpoint of quality of cooling design and for possible modifications to reduce its size and/or cooling power requirements. For such purposes, the measurements necessary for determination of operational thermal conditions, as outlined above, must also be made. In addition, the following measurements should be made for all types of units:

a. Internal Air Temperatures. For evaluation purposes, the determination of internal temperature distribution is of considerable importance since it gives an indication of required spatial modifications. Measurements should be made with shielded thermocouples located throughout the unit, in voids between components and near the surfaces of the case. Particular care should be taken in locating thermocouples where stratification of high-temperature air is indicated and where lack of air distribution is suspected because of the interference of other components and baffles. There is no fixed specification in reference to spacing of internal temperature measurement stations, but they should not be separated more than six inches. For small units they should be located considerably closer. Where the internal air circulation pattern is known, particularly in units employing internal forced convection, air temperatures should be measured at regular distances along the air flow path. In most locations, the thermocouple probe used for air temperature measurement must be shielded from radiation of surrounding components. Only when the environment consists principally of non-heat-dissipating components may a bare thermocouple be used.

b. Component Surface Temperatures. If possible, the surface temperature of most components should be measured if a true determination of the quality of the design is to be made. It is equally important to determine the surface temperatures of components exposed to favorable cooling conditions, as well as of components exposed to unfavorable cooling conditions. This is desirable since important clues for the spatial re-arrangement of components and the re-distribution of air flow can be gained in this manner. This applies equally to high- and low-temperature components since the tests are designed to disclose information which will indicate the changes necessary to bring the operating temperatures of all components as close as possible to the allowable temperature limit of each component.

c. Internal Blower Input. For all units utilizing blowers internally for spot cooling or general air circulation, it is desirable to determine the electrical input to the blower motor in order to provide data for the evaluation of the blower's effectiveness. A measure of the blower's effectiveness is furnished by comparison of the added external heat dissipation requirement of the entire unit, resulting from the losses of the motor, with the reduction of component temperatures resulting from the acceleration of internal air flow by means of the blower.

3. Measurements Required for Pressurized and Sealed Units Cooled Externally by Forced Convection

Certain additional measurements, besides those applicable to all units, are required in bench tests of all pressurized and sealed units employing an external cooling air blower. The principal purpose of these measurements is to furnish the data which would give information on the relationships between the performance and characteristics of the heat exchange system and internal and external blower performance.

For the determination of operational thermal conditions the following measurements should be made on these units:

a. Temperature of Heat Exchange Surfaces. Temperature measurements are required on all surfaces which are on opposite sides in contact with internal and external cooling air. As pointed out in Chapter II, the heat exchange surface may be represented by the case proper but may also consist of tubes or plates within a heat exchanger core. It is important to obtain a representative heat exchanger surface temperature usually derived from several temperature measurements, since the component temperatures in such units can be directly correlated with this temperature. Therefore, it is desirable to measure the temperature of the heat exchange surface at a number of points along the cooling air path and in any event at at least three or four locations where it is possible to install thermocouples. However, in compact heat exchanger cores the internal installation of thermocouples would be extremely difficult and, therefore, it may only be possible to measure the surface temperature near the inlet and near the outlet of the heat exchanger. With heat exchanger cores employed in equipments having an integrated or separate heat exchanger, the difficulty in measuring temperatures of heat transfer surfaces is avoided by describing performance in terms of air temperatures on both sides of the heat transfer surfaces. For the installation of thermocouples, methods such as those described on pages 26 to 28 in Chapter III should be used. It is most desirable to peen the thermocouple junctions into the metallic heat exchange surfaces.

b. Cooling Air Flow Rate. In order to make the analysis of any heat exchange system possible, it is necessary to correlate flow rate, pressure drop, and surface temperature. Therefore, in bench tests the determination of cooling air flow rate is necessary. This air flow may be caused by a blower integrated into the design of the unit or located at any other point in the flow system. For the purpose of air flow measurement, an apparatus consisting of a blower and a metering section, mentioned on page 41 and described in Appendix IV, should be used. It incorporates a convenient source of air supply which is satisfactory for determination of the characteristics of the heat exchange system under bench test conditions, since blowers integrated into the design of units often are not capable of operating continuously under bench test conditions. They may be designed for only intermittent operation at low altitude since their motor may be designed to carry the cooling blower's load only at high altitude. Therefore, if no speed control is used, the weight flow capacity of the blower would be excessive at low altitude and would overload the motor. If the blower motor is capable of operating continuously under bench test conditions, it is desirable to use the auxiliary air flow apparatus only in such a way that it principally serves to overcome the resistance of the metering section while the cooling blower of the unit is used to overcome the resistance of the heat exchange system. Operating in this manner, not only air flow measurements, but also data on the blower characteristics are obtained. Variation of the air flow rate by means of the auxiliary apparatus is possible and necessary for certain tests, as discussed in subsequent sections of this chapter in reference to particular types of units.

c. Heat Exchanger Pressure Drop. An additional requirement for the determination of heat exchange system characteristics is the measurement of the pressure drop of the external cooling air passing over the heat exchange surfaces. Measurement of an overall pressure drop is usually sufficient. It can be made quite simply by installing two pressure taps, one at the inlet and the other at the outlet of the heat exchange system, taking the necessary precautions as discussed and described on pages 35 and 36 of Chapter III. A differential manometer connected between the two pressure taps would then simply indicate the pressure drop at any flow rate of the cooling air through the heat exchanger. Locations of pressure taps should be accurately recorded to allow adjustment of the measured pressure drop for the differences in flow area at the measurement stations.

d. Cooling Air Temperatures. Measurement of the cooling air temperature at the inlet and exit of the heat exchanger is required for definition of the heat exchanger characteristics as well as for checking the total heat dissipation of the unit calculated from electrical measurements. By comparing the heat dissipation of the unit determined from electrical measurements to the heat dissipation calculated from the measurement of the air flow quantity and the temperature rise of the cooling air passing through the heat exchanger, an indication is usually obtainable of the accuracy of air temperature measurements which are most likely to be less precise than the other measurements. In some instances where the cross-section of the air flow passage is most irregular at inlet and outlet and where the air flow distribution is not uniform it is difficult to obtain a precise measurement of the temperature of the air stream at entrance and exit of the heat exchanger. The methods to be utilized to get an indication of the representative air temperature by means of multiple measurements and sub-division of the cross-sectional flow area are indicated on page 21 in Chapter III and should be used wherever possible. For certain types of equipments, particularly for those in which the cooling air flows through parallel heat exchange passages in which the air temperature rises are unequal, use of an insulated mixing chamber at the exit of the equipment may prove necessary when fairly accurate bulk temperature measurements are desired. The data on temperature rise of the cooling air combined with the information on the exchanger surface temperature, heat exchanger pressure drop, and total cooling air flow rates serve to define the characteristics of the external heat exchange system.

e. Blower Tests. When the operating characteristics of the external cooling air blower and its driving device employed with any forced-air-cooled unit are not known, the interpretation of the bench test data for determination of performance under operational conditions is not possible unless a blower test is performed. For that purpose the blower which may be integrated into the design of the unit may have to be removed and tested separately as outlined in Appendix V. However, if the installed blower is accessible, so that it is possible to make the necessary measurements, the blower test may be performed on the unit. The air flow measuring apparatus described in Appendix IV can be used for the purpose of flow measurement. Other measurements to be made in conjunction with the blower tests are atmospheric air pressure, static pressure at the discharge of the blower, and blower speed. It is also desirable to ascertain the power requirements of the cooling air blower for certain loads although this information is of little significance in determining the operational thermal conditions of a unit. However, for determination of the quality of the cooling design of the entire system, knowledge of the power requirements of the cooling air blower is essential.

For the determination of the effectiveness of the thermal design and to obtain data for possible modifications, the following measurements should be made on pressurized and sealed units cooled by forced convection:

f. Internal Air Pressure. Units equipped with mechanical means of pressurization are influenced in their performance by the adequacy of the pressurization device, since the internal pressure level has an effect on the convective heat transfer process between the components and the internal atmosphere of the unit. Therefore, measurement of the internal pressure should be made during bench tests. For the same reason, measurements of the internal pressure of sealed units should be made since a characteristic pressure rise of the internal atmosphere would occur as the internal temperature of such units increases. The maintenance of constant pressure at equilibrium temperature would also indicate the quality of the methods utilized for the purpose of sealing the unit. The pressure measurements can be made simply by means of a manometer, as described on page 35 in Chapter III.

g. Internal Cooling Air Flow Rate. For evaluation purposes, not only determination of the power input to the internal cooling air blower is desirable, as mentioned on page 52, but it would also serve well to be informed on the rate of internal air circulation. Measurements for this purpose are rather difficult to accomplish since in equipment containing internal air cooling blowers the conditions are such that the installation of an air metering device is hardly possible. Therefore, it may ordinarily be necessary to perform a blower test on the internal circulating device according to the methods indicated in Appendix V. For the purposes of the test, the blower would have to be removed from the unit. Another alternative is that from known operating characteristics of the blower, the measured static pressure rise of the air by the blower, the air temperature and static pressure at the blower intake, and the measured rotative speed of the blower, the internal air circulation rate can be calculated. For internally ducted units only, where the total internal air circulation passes through the blower and the internal heat exchanger, the air flow rate can be calculated from the measured temperature drop of the internal cooling air across the heat exchanger. By dividing the product of this temperature drop and the specific heat of air into the total heat dissipation of the unit, determined by electrical measurements and corrected for external heat loss or gain of the case, the approximate weight flow rate of the internal cooling air would be obtained.

h. Internal Pressure Gradient and Velocity Distribution. The effectiveness of the cooling design of closed units cooled externally by forced convection and ducted internally requires the internal exploration of the unit for pressure drop across various sections to determine whether undue power requirements result by reason of the cooling air flow through sections where little or no heat dissipation may take place. The measurements of static pressure required in such evaluation tests are described on pages 35 to 37 in Chapter III. In addition, it is also desirable for the study of local cooling conditions to investigate velocity distributions at various sections so as to ascertain what possible modifications could be made in the arrangement of components. The techniques outlined on pages 45 to 47 in Chapter III may be utilized, depending on the design of the unit. When relatively high internal circulation rates are used, rather accurate quantitative measurements are feasible. Otherwise, qualitative methods of flow visualization, such as mentioned on page 47, are applicable. They require the construction of a transparent case which in dimensions and other details affecting the flow pattern is equivalent to the actual metal case of the unit.

4. Tests of Pressurized or Sealed Units with Case Cooled by Free Convection and Radiation

a. Measurements. All necessary measurements to be made on such units are covered in the preceding Sections 1 and 2 of this chapter. The requirement for measurement of internal pressure level, mentioned in preceding Section 3 with reference to units cooled externally by forced convection, also is applicable to this type of unit.

b. Test Runs for Determination of Operational Thermal Conditions

At least four test runs should be made in which the unit should be so operated that the surface temperature of the case is different in each run. The first run should be made under known conditions of room temperature and pressure and with a well-defined radiation environment, as previously discussed. In this run the standard case and its finish to be used in the actual aircraft installation should be used. The three additional runs should be made with increased case temperature so that at least four points are available from which curves of component temperatures versus case temperature may be plotted.

The case temperature may be increased by insulating the case with thin evenly distributed layers of insulating material. The type of material is unimportant, other than that it should be capable of withstanding the maximum anticipated temperature. For insulating many units, cotton fabrics or corrugated cardboard may be used without difficulty, although for high-temperature units woven glass fabric or asbestos paper is preferable.

It is most desirable that several runs are made, each with a different thickness of insulation and that data for each run be recorded under equilibrium conditions. In one run the temperature of the case should be increased to a point where one or more components, either heat-producing or non-heat-producing, would approach their respective critical or limiting temperatures. This would define the entire range of possible application of the unit in terms of its case temperature. However, choosing the correct thickness of insulation to cover this range by preferably no more than four runs may give some difficulty. It is simplest to use heavy insulation, immediately after the first run without insulation, and thus to attempt to approach limiting conditions. The necessary reductions of insulation thickness to obtain the two intermediate conditions may be estimated on basis of the observed increase in case temperature due to application of insulation. This increase in case temperature should be approximately proportional to the insulation's thickness. Thus, for example, if 1 inch of insulation is found to raise the case temperature, say, 40°C, it is necessary to reduce the thickness to 1/2 inch if the case temperature should be increased but 20°C.

Surrounding the unit with an environment of controlled temperature would best accomplish the desired aim of producing data on variation of component temperatures with case temperature. A special heated test chamber with temperature control could be utilized, but would add substantially to required costs of instrumentation. Submersion of the unit in an agitated liquid bath of controlled temperature would be best for small units with a volume in the order of one cubic foot, but would involve more elaborate construction for units of greater volums. A method similar to that of wrapping the unit in fabric insulation, but less cumbersome, would consist in employing an electrically heated blanket around the case. The operation of the heating elements in the blanket can be controlled and the case temperature can be maintained at the desired value by adjusting the level of heat input to the blanket. A potentiometer controller, actuated by a thermocouple imbedded in the wall of the case, may be used as an on-off control for the heating elements, thus automatically maintaining the required case temperature.

c. Test Runs for Determination of Effectiveness of Thermal Design

If data also have to be gathered for the determination of operational thermal conditions, no additional test runs are required for the determination of the effectiveness of thermal design. It is only necessary to make additional more complete measurements, as outlined in the preceding general sections on measurements applicable to all types of equipment. Should it only be desired to study the temperature distribution within the unit, a single test run would be adequate to furnish sufficient data and to indicate what modifications may be necessary in the arrangement and methods of installation of individual components. Such a test run should preferably be made under equilibrium conditions which would produce limiting temperatures of one or several components. The methods for accomplishing this are described in the preceding paragraph on test runs for the determination of operational thermal conditions. Thus, one of the test runs applicable to the determination of operational thermal conditions can be also utilized for the determination of the effectiveness of thermal design. The case temperature obtained in this run, when interpreted in terms of the cooling conditions expected in the installation compartment, would indicate whether the external methods of heat dissipation from the unit, i.e. free convection and radiation, are adequate to dispose of all the generated energy. In other words, the heat dissipation from the case, calculated on basis of the maximum allowable case temperature and known convective and radiant heat transfer coefficients being determined by environmental conditions, must equal the difference between the electrical energy input and output of the unit. A conclusion can also be reached on the required modification, if necessary, to maintain the case temperature within the determined limit.

5. Tests of Pressurized and Sealed Units with Case Cooled by Forced Convection

a. Measurements. All necessary measurements to be made on such units are covered in the preceding Sections 1, 2, and 3.

b. Test Runs for Determination of Operational Thermal Conditions

The purpose of the test runs is to obtain data which would correlate the surface temperature of individual components with the surface temperature of the case. In such units the surface temperature of the case is not only a function of the environmental conditions but is principally determined by the rate of air flow. Therefore, the test runs must define the flow and heat transfer characteristics of the cooling air passage and the external loss characteristics of the entire unit. Using the air flow measuring apparatus described in Appendix IV, the rate of air flow cannot only be measured but may also be controlled independently of the natural performance of the unit's external blower. The cooling air blower may be operated continuously for all test runs and would require the assistance of the auxiliary blower of the air flow apparatus only when its motor would be incapable of carrying the entire load.

The test runs will give only part of the data required for the determination of operational thermal conditions. For that purpose, the characteristics of the blower and the motor must be known to permit calculation of the available air flow under any specified conditions. This procedure is illustrated in Example V-3 on page 127. For experimental determination of blower characteristics, test methods described in Appendix V may be used unless manufacturers' data are available. Similarly the speed-torque curve for the driving motor of the blower would best be obtained from a manufacturer's curve.

In the first test run the limiting flow rate should be determined for which one or several components would reach their maximum allowable temperatures. This determination is based on a trial-and-error adjustment which involves the successive reduction of the rate of air flow. It is also likely that under bench test conditions the rate of heat loss from the outer surface of the flow passage by radiation and free convection would be sizeable. Therefore, the minimum flow rate so determined would not be entirely representative, unless the case is insulated. The insulation should be made up of 3 inches of glass wool or felt, covered externally with aluminum foil. This would reduce the heat loss from the surface of the case to about 1 watt per square foot for each 25°C of temperature difference. On basis of the required air flow rate in the limiting test run, the desired flow rates for additional test runs can be up to three-times greater than the basic rate. At least three, but preferably five runs should be made with the increased rates of air flow.

In summary, to produce part of the data for the determination of operational thermal conditions, several test runs must be made with cooling air flow rates varying between a minimum quantity corresponding to limiting temperatures of one or several components, to a maximum quantity at least three-times as great as the minimum quantity. In such test runs the measurements pointed out in previous sections to be applicable to all units, and more particularly those applicable to pressurized units cooled by forced convection, must be made to define the heat transfer characteristics of the unit. The use of the auxiliary air flow apparatus for the supply of air as well as for a means of measuring the air flow is necessary.

c. Test Runs for Determination of Effectiveness of Thermal Design

The test runs which are required to produce data necessary for the calculation of operational thermal conditions may also be used for the determination of the effectiveness of the thermal design. For that purpose, a single test run, preferably the one at which limiting component surface temperatures are reached, would be sufficient. However, more complete instrumentation and additional measurements, as outlined in the preceding Sections 1 and 2, are necessary. Extensive exploration of internal surface temperatures and air flow velocities would give data on the quality of the internal design of the unit which consists principally in the arrangement of components, the internal air distribution system, and the means employed for the transfer of heat from the components to the heat exchanger. The results of this single test run would be sufficient to give an indication of the modifications necessary in the arrangement of components and other contributing factors to improve the effectiveness of the design.

6. Tests of Pressurized and Sealed Units with Case-Envelope Heat Exchanger Cooled by Forced Convection

Measurements and test runs to be performed on such units for the determination of operational thermal conditions as well as for the determination of effectiveness of cooling design are the same as those for pressurized and sealed units with plain case cooled by forced convection. Because of more effective forced convective cooling, units with case-envelope heat exchangers need not necessarily be tested with external insulation since heat losses from the external surface of the heat exchanger would represent a relatively small quantity of the total heat dissipation. However, in making measurements of the temperature of heat exchanger surfaces it is necessary to make a more complete exploration than where the plain case is the only heat transfer surface. Similarly, in the study of the internal heat transfer pattern, desirable for the evaluation of the effectiveness of the thermal design, a more complete exploration of internal air distribution patterns, pressure drops, and individual component temperatures is desirable because units with external heat exchangers would have, as a rule, rather systematic internal air flow patterns which deserve considerable care in their design in order to insure best power economy. The operating characteristics, consisting of power requirements, discharge quantity, as well as static pressure production, of the internal as well as the external cooling blowers, should also be determined. As mentioned in the preceding section on units with a plain case cooled by forced convection, the heat exchanger's flow characteristics should be determined by the use of the auxiliary air flow apparatus and its measuring section. If the blower characteristics must be determined experimentally because performance data are not available, a standard test as described in Appendix V must be performed on the blower, disconnected and removed from the electronic equipment.

7. Tests of Pressurized and Sealed Units with Integrated or Separate Heat Exchanger Cooled by Forced Convection

The principal measurements and test runs required for this type of unit are the same as for the other pressurized and sealed units cooled by forced convection described in the preceding Sections 5 and 6. If the same type of test runs are performed, data are produced which would correlate the component surface temperatures with the average internal air temperature, and the external cooling air pressure drop with the weight flow rate through the heat exchanger. Measurement of heat exchanger surface temperatures is not required as shown in Example VI-7, page 186. In these test runs with variable external air rate heat dissipation other than that caused by forced convective heat transfer in the integrated or separate heat exchanger may be eliminated by insulating the external surfaces of the equipment case. The cooling data so obtained would minimize the effects of radiation and conduction from the components to the case. This would result in greater air flow requirements than would be necessary if the environmental conditions were such that some heat loss would be possible from the surface of the heat exchanger case under normal operating conditions. However, it is also permissible to make the test runs with bare case, but then case temperature measurements must be made to provide the data for calculation of the external heat dissipation. The heat dissipation so calculated may be checked by heat balance calculations in which the difference between the total heat dissipation determined from electrical measurements and the heat dissipation accounted for by the air flow quantity and temperature rise of the external cooling air passing through the heat exchanger is determined.

8. Tests of Vented Units with Closed Case Cooled by Free Convection and Radiation

The only measurements to be performed on such units are those which are applicable to all types of units as discussed and described in Sections 1 and 2. Calculations, based on bench test data, for determination of operational thermal conditions have only limited accuracy when applied to this type of unit. Similarly, it is difficult to determine the effectiveness of the thermal design accurately because variation of the case temperature alone under bench test conditions would not be sufficient to give information on corresponding variations in component temperatures under various altitude conditions. Any given surface temperature variation of the case would correspond to a certain operating altitude and operational environment for which the internal atmosphere of the equipment would be different than that available under bench test conditions. Consequently, the mechanism of heat transfer from the components to the case would also be altered and, therefore, the temperature difference measured between components and the case under bench test conditions would be materially different, even though the case temperature could be identical. Therefore, bench test data of such units must be supplemented with altitude-chamber test data if a complete thermal evaluation study is to be made.

Bench test data such as may be used for the determination of operational thermal conditions and the effectiveness of the thermal design may be obtained in a single test run. In this test run which could be performed for both purposes, depending on the extent of the measurements being made, the unit should be operated until equilibrium is reached under standard bench test conditions and in the form in which it is intended to be installed in the aircraft. No additional test runs under modified conditions, such as could be obtained by covering the case with insulation, are required since the data so obtained would be only of limited significance. They would give some information on the operating characteristics of such components which would be dependent largely on conductive heat transfer to the case and would, therefore, disclose the contribution which other modes of heat transfer such as free convection and radiation would make to the total heat dissipation from such components. In general, however, it should be emphasized that the single test run, indicated above, would be sufficient to provide data for the calculation of temperatures of the most critical components under operational conditions and to give information on temperature distribution within the unit which would be indicative of the quality of thermal design.

9. Tests of Vented Units with Open Case and Through-Flow of Atmospheric Air by Natural Convection

Bench tests of such units yield insufficient data for the determination of operational thermal conditions by calculation. Altitude-chamber tests are required to produce data for evaluation of operational thermal conditions. However, bench tests are at least partly applicable to the determination of the effectiveness of thermal design of such units, although for this purpose also supplementary information from altitude chamber tests is desirable. The bench test data would not give a reliable indication of the quality of thermal design in terms of operation under high-altitude conditions. Under such conditions, the distribution between the various modes of heat transfer contributing to the heat dissipation from individual components would be altered. On the other hand, in a single bench test in which most of the component temperatures are measured, a temperature distribution pattern within the unit can be determined which would readily disclose most of the existing unfavorable conditions. The type of measurements to be made would be essentially the same as those for all other types of electronic units. Component temperatures are of principal importance, while case temperatures and air temperatures are of interest for reference purposes but would not be necessarily indicative of the quality of the unit's thermal design. In some instances, internal measurements of air temperature distribution would disclose regions of stratification which are the result of unsatisfactory arrangements of air circulation passages. Since the test procedure is quite simple, requiring no control equipment and a limited number of measurements only, repeated measurements may readily be made. Successive tests should be performed under equilibrium conditions. They should serve to supplement data previously obtained with progressively more detailed information. The type of required information would be established by cumulative analyses of available data.

10. Tests of Vented Units with Open Case and Forced Through-Flow of Atmospheric Air

a. Measurements

In addition to the measurements required for all electronic units, several measurements must be made which are similar to those required for pressurized units being cooled by forced air flow. Like the pressurized unit, the vented unit with forced air flow over the component surfaces can be defined as a heat exchanger on the basis of the surface temperatures of components, the overall pressure drop of the cooling air, the total rate of air flow, and the temperature rise of the cooling air passing through the unit.

For the purpose of the determination of operational thermal conditions the following additional measurements must be made:

(1) Cooling Air Flow Rate. Measurements have to be made by application of the air flow measuring apparatus as described in Appendix IV. The details as discussed on page 53 with reference to the pressurized equipment cooled by forced air flow are also applicable to this type of equipment.

(2) Cooling Air Pressure Drop. In order to be able to define the flow characteristics of the electronic unit in which the components are in direct contact with the cooling air, not only the cooling air flow passing through it has to be measured but also the pressure drop. An overall measurement from the inlet to the outlet of the unit is sufficient. Actually this can be obtained by a single measurement of static pressure at the inlet or the exit of the cooling air blower, depending whether the blower is a suction or a discharge device, inducing or forcing flow through the unit, respectively. Details indicating the technique of making the static pressure measurements are discussed on pages 35 to 41 in Chapter III.

(3) Cooling Air Temperature. Measurements of the inlet and exit temperature of the cooling air must be made in accordance with the requirements discussed for pressurized units on page 53.

(4) Component Temperatures. For this particular type of unit, the complete exploration of component temperatures, for other units usually only required for the determination of the effectiveness of the thermal design, is of considerable importance for the purpose of the determination of operational thermal conditions. Only if a sufficient number of component temperatures are known can the heat exchange characteristics of the unit be defined similarly to those of a surface-type heat exchanger of tubular or plate construction. It is necessary to measure the temperatures of as many components as possible which would be expected to operate near their respective limiting temperature under thermally severe operational conditions.

For the determination of the effectiveness of thermal design and to obtain data for possible modifications, the measurements to be made on vented units with forced through-flow of atmospheric air are the same as those required for the exploration of the internal characteristics of pressurized units with internal blowers, as discussed on pages 54 and 55. In addition to the exploration of component temperatures, the following measurements are of particular significance:

(5) Air Flow Distribution. When several parallel flow passages exist, measurements should be made of flow distribution. These data combined with knowledge of component temperatures would be indicative of possible energy savings and reduction of blower size to be effected by redesign of the flow passages. The techniques to be utilized are dependent on the design of the unit. In general, the pressure drop across a given passage, when the unit is operating normally with flow through all passages, could be used as reference to reproduce the same flow when all other passages are blocked. Then, measurement of partial flow is possible by means of throttling the blower, utilizing methods indicated in Chapter III.

(6) Pressure Gradients and Velocity Distribution. The evaluation of the effectiveness of air distribution within a passage requires knowledge of the pressure drop across various portions of the passage. Comparison of the cooling energy, determined on basis of such pressure drops and the flow rate in the passage, with corresponding heat dissipation would reveal the adequacy of the component spacings used. The applicable techniques are outlined on pages 35 to 37 in Chapter III.

b. Test Runs for Determination of Operational Thermal Conditions

Like for pressurized units cooled by forced convection the test runs are designed to produce data for the determination of the heat transfer and flow characteristics of the unit. The same general methods as described for pressurized units on page 51 are also applicable to vented units with forced through-flow of atmospheric air. An attempt should be made to make the basic test run with such a rate of cooling air flow that the limiting operating temperature on one or several components is obtained. However, since the case for all test runs should remain uninsulated, it is very likely that if the attempt is made to produce such cooling conditions in bench tests, a major portion of the heat transfer from components would be by free convection which could not be correlated with the operation of the unit at high altitude where forced convective cooling produced by the blower would predominate. Therefore, care must be taken in the interpretation of the data of this basic test run. They may have to be disregarded if found not to correlate satisfactorily with data obtained at higher flow rates. These latter test runs should be made then at increased rates of air flow up to the natural discharge capacity of the blower under bench test conditions. It is possible that the blower motor would have inadequate power to carry the load imposed by the blower under bench test conditions operating continuously. In that case, the auxiliary apparatus must be relied upon to supply the cooling air, although the integrated blower of the unit may still operate but would then not have to provide for the total pressure drop which the cooling air would encounter passing through the unit. However, the air flow quantity would be the same. At least five runs should be made with different rates of air flow. It is desirable to make at least three runs at higher flow rates where, under bench test conditions, component temperatures may be reduced appreciably below the allowable limit. For the data produced by these runs, the correlation and interpretation of the heat transfer process based on forced convection would be most applicable. At the lower flow rates, where component temperatures would tend to reach the maximum values, the analysis would not be entirely reliable because of the aforementioned considerable free convective cooling effect to be expected.

c. Test Runs for the Determination of Effectiveness of Thermal Design

These test runs would not differ from those to be made for the determination of operational thermal conditions, other than by the type of measurements to be made, particularly in respect to air flow distribution pressure gradients and internal velocity distributions. Since in the test runs for the determination of operational thermal conditions a considerable number of component temperatures should be measured, few supplementary component temperature measurements would have to be made for the purpose of determining the effectiveness of the thermal design. Additional information would consist in determination of the power requirements of the blower. Consequently, for the sole purpose of determining the effectiveness of thermal design, only a single test run is required under standard bench test conditions with the blower operating at its normal discharge rate. From such data sufficient information would be obtainable to indicate possible modifications in the arrangement of the components and cooling air passages.

Altitude Chamber Tests

Procedures used in altitude chamber testing are little more complicated than those for bench testing. In fact, as will be apparent from the following subject matter, altitude chamber tests are simpler in some instances. However, it is suggested that they not be looked upon as the principal experimental method for thermal evaluation. They require a sizable investment in test equipment, and complications arise frequently in the operation of this equipment. Installing an electronic unit in an altitude chamber and instrumenting it for suitable temperature measurements is a time-consuming task. Also, it should be understood that environmental conditions produced in an altitude chamber fall far short of simulating actual operational conditions encountered in aircraft installations. Altitude chambers of limited size do not permit testing of an entire equipment made up of several units, for evaluating the thermal interaction of these units. In particular, the state of air motion in most altitude chambers makes it impossible to correlate test conditions with those in aircraft compartments. The latter must always be defined in terms of four variables, i.e. the ambient pressure, the ambient air temperature, the rate of air motion, and the temperature and surface characteristics of the compartment walls. As a rule, altitude chambers presently used can only simulate the first two of the four variables.

In view of the above limitations of altitude chambers of current design, it is recommended that their use in the thermal evaluation of electronic units designed to operate in the steady state be restricted to certain types of units and evaluation problems. Recommended tests would provide data for the prediction of operational thermal conditions only, since the effectiveness of thermal design can usually be evaluated on the basis of bench test data. Vented units are the exception, because for them the distribution between free convection and radiation from basic components affects the effectiveness of thermal design under high-altitude operating conditions. This distribution cannot be predicted from bench test data. Thus, while required for vented units, complete temperature explorations for the purpose of evaluating effectiveness of thermal design would, in general, be considered incidental to altitude chamber tests.

Three basic applications of altitude chamber tests for thermal evaluation of electronic units apply to (1) vented units which cannot be evaluated on the basis of bench test data, (2) units equipped with blower-motor units of unknown characteristics and for which operational conditions need be predicted only for a limited range of environmental conditions, and (3) verification of analytical results obtained from the interpretation of bench test data, providing environmental conditions can be controlled to correspond to those assumed in the analysis.

The first type of application is principally concerned with vented units which are either closed and cooled externally by free convection and radiation, or are open, and cooled by natural through-flow of compartment air. While for the former type of unit no special precautions are necessary in the performance of altitude chamber tests, the latter type is very sensitive to environmental conditions, particularly in respect to air motion, and therefore, in order to obtain data which are suitable for interpretation, extensive precautions must be taken to eliminate the effects of the chamber's internal air circulation pattern. Details of the required test apparatus are further discussed on page 68.

In the second type of application to electronic units equipped with blower motor whose characteristics are not known, simple test procedures may be utilized to correlate directly component temperatures, or, by means of heat balance, the available air flow rate, with environmental conditions as defined by air pressure and temperature. In such tests all effects of the altitude chamber design must be eliminated by thermal insulation of the test unit. This is further discussed on page 66. It is also possible to obtain, by means of altitude chamber tests, additional information on units which are subject to appreciable heat losses from their external surfaces by other modes than the forced-convective cooling provided by their blower-motor units. This information can be utilized to predict more accurately operational thermal conditions in an environment of known characteristics. For all pressurized units the necessary test procedures are quite simple. For vented units utilizing forced through-flow of compartment air, more extensive test apparatus is required, as discussed on page 70.

It should be mentioned here that for pressurized units cooled by free convection and radiation, altitude chamber tests are entirely insignificant since they could not simulate actual installation conditions unless extensive auxiliary apparatus within the altitude chamber is used to provide essentially a mock-up of the installation compartment. Otherwise, it is more reliable to calculate case surface temperatures on such units from analytical considerations and to correlate component temperatures with the case temperatures so determined, based on values of temperature rise measured in bench tests.

Details on required measurements and test procedures to be used for various types of units in altitude chambers are discussed in the following sections to the extent in which they differ from bench test practice. The general considerations outlined on page 49 for bench tests in reference to types and quantity of temperature measurements, duration of test runs, and techniques of measurements also apply to altitude chamber tests.

1. Measurements Required for All Types of Units

The following measurements are required for all altitude chamber tests to define the test conditions and the overall characteristics of the unit. The latter are determined as in bench tests, discussed on pages 50 and 51, except that component temperature measurements are not necessary if bench test data are available. Component temperature measurements would be made on vented units or on pressurized units if running of additional bench tests is not desired. The test conditions in altitude chambers are defined by the following measurements:

a. Air Temperature. Measurement of the air temperature in an altitude chamber is usually practical by means of the temperature-measuring device installed in the chamber for control purposes. However, frequent calibration of such devices which are commonly expansion-type elements, is recommended. An independent measurement for calibration can be obtained at one location by means of an unshielded probe, since the inherent turbulence of air inside the chamber causes thorough mixing.

b. Environmental Surface Temperatures. The measurement of altitude chamber wall temperatures is usually not necessary, since they are essentially equal to the air temperature. The analytical procedures outlined in Chapter VII do not require definition of the radiation environment created by the walls of the chamber.

c. Chamber Pressure. Pressure-measuring devices usually permanently installed in altitude chambers, are of sufficient accuracy to give indications of corresponding altitude conditions whenever individual tests are performed for specified conditions. However, if test data are desired for several altitudes and should subsequently be correlated, or should serve to verify calculations made on basis of bench test data, accurate pressure measurements are desirable. The sensitivity of altitude gages commonly installed on chambers may not be good enough for such purposes and their calibration may not be entirely reliable. Under such circumstances, the use of a mercury-column manometer for measurement of the gage pressure and a sensitive barometer for the determination of the atmospheric pressure are recommended to provide information on the absolute chamber pressure with an accuracy of at least 0.05-inch mercury.

2. Tests of Pressurized and Sealed Units Cooled by Forced Convection

The objectives, measurements, and procedures discussed in the following apply to units with a plain case surrounded with a circumferential baffle to provide an air flow passage, as well as to units with case-envelope type heat exchangers. Altitude chamber tests on such units may be desirable if it is necessary to determine the performance of the unit for a specified combination of air pressure and temperature and the characteristics of the blower-motor unit are not known. Under such circumstances either one of two objectives may be fulfilled. First, it may be desired to determine merely the air flow rate of the blower in combination with the unit for the given pressure and temperature conditions; then measurements of surface and component temperatures would not be necessary since bench test data could be used to calculate them on the basis of the determined air flow rate. Second, it may be desired to determine the highest possible case temperature for the given air pressure and temperature if no external losses from the unit occur; then measurement of air flow would be unnecessary since component temperatures can be calculated from the determined case temperature on basis of temperature rise data obtained in bench tests. In addition to blower performance, it may also be desired to determine the heat transfer characteristics of the external surfaces and cooling passages of the unit so as to provide more concise data for the calculation of external losses than would be available from bench test data. For that purpose, more extensive measurements and test runs would be necessary, as outlined below, but component temperatures would not have to be determined.

a. Measurements. In addition to the measurements required for the determination of overall characteristics, as pointed out on pages 50 and 51, certain other measurements are necessary. In particular, if it is desired to determine the air flow rate of the blower, accurate measurements of heat dissipation and of the mean air temperature entering and leaving the cooling passage of the unit must be obtained. The positioning and methods of shielding thermocouples to be used for such purposes are discussed in Chapter III, pages 26 to 28. Also, if it is desired to obtain data on the heat transfer characteristics of the external passages and surfaces, thermocouples must be installed at regular intervals on the external surfaces of the cooling air passage.

b. Test Runs

For all test runs, the unit must be mounted in the chamber in such a manner that the inlet and outlet of the cooling passage are not affected by the turbulent conditions which may exist in the chamber. Particular care must be exercised that these openings are not so located in reference to the air circulation pattern within the chamber that a positive or negative pressure be produced on the air intake or outlet of the unit. Obviously, this would result in incorrect determination of the air flow rate and the flow-producing capabilities of the blower. Therefore, particularly if the chamber is small as compared to the size of the unit, the installation of deflection baffles or of suitable screens is recommended. Quantitative measurements to determine such effects are complicated and laborious. The use of flow visualization techniques, as outlined on page 46, is suggested. Indications should be obtained when the chamber is closed and its air circulation system is in operation, while the blower of the electronic unit is inoperative.

For the purpose of determining the air flow rate obtainable under specified conditions of ambient pressure and temperature, it is necessary to make one test run in the altitude chamber for each combination of pressure and temperature until thermal equilibrium conditions are obtained. Since it is desired to determine the air flow rate on basis of the known total heat dissipation of the unit and the temperature rise of the air passing through the unit, it is required that the unit be insulated by several layers of felt, at least 2 inches thick overall, so that practically all the heat dissipation would occur to the cooling air. The same test procedure, also with one run each for specified conditions of pressure and temperature, would be necessary to determine a direct correlation of measured case surface temperature with ambient pressure and temperature conditions without calculation of the air flow rate. The resulting measurements of case surface temperature would be the extreme values to be expected under actual installation conditions, since the external insulation prevents any other heat dissipation to the test environment. The other probable extreme would be established by one test run each for given conditions of pressure and temperature with a bare external surface of the unit. It is likely that this condition would represent the greatest probable heat loss since the rate of air motion in the altitude chamber would usually be greater than in an aircraft compartment. Such a second series of test runs is required when data are desired on the heat transfer characteristics of the external cooling passages of the unit and their effects on heat dissipation other than to the flow of cooling air supplied by the unit's blower.

3. Tests of Pressurized and Sealed Units with Integral or Separate Heat Exchanger

The objectives of performing altitude chamber tests on this type of unit are usually the same as for the other pressurized units, as described in the preceding section.

a. Measurements. In addition to the measurements made on the other pressurized units, it is of particular importance for units with integral or separate heat exchangers to obtain temperature data on the air entering and leaving the section within the unit where the electronic components are installed. Also, to provide data for determination of the external heat dissipation characteristics of the unit the surface temperatures of the case must be measured by means of embedded thermocouples for all test conditions. It may also be desirable in some instances to measure temperatures of the outer surface of insulation applied to the case in the various tests, as outlined below. Such measurements are useful to provide data for comparison of heat flow through the insulation calculated by measured temperature drop across the insulation and estimated thermal conductance, with heat flow determined from heat balance calculations.

b. Test Runs. The same precautions as outlined in the preceding section must also be observed for this type of unit. For each ambient condition specified, the number of required test runs would vary, depending on the purpose for which the data are intended. To determine the available air flow rate alone, one test run for each condition would be sufficient, providing the external case of the unit is heavily insulated so as to eliminate external heat loss. Such a run would also correlate directly the mean internal air temperature with external conditions and would permit, with the aid of additional bench test data, determination of component temperatures for the worst possible conditions which would occur when there is no heat dissipation from the case. Additional test runs are required to provide data on the heat-dissipating capacity of the unit's case. For that purpose, for each combination of environmental pressure and temperature at least two additional test runs are desired with reduced thicknesses of insulation and one additional test run with bare case. Data so established for each environmental operating condition would correlate the heat dissipation from the case with the case surface temperature.

4. Tests of Vented Units with Closed Case

The objectives in testing such units in altitude chambers are usually to provide data for the prediction of component temperatures for specified environmental altitude, as well as for the evaluation of the effectiveness of thermal design with particular reference to altitude operation. Therefore, the scope of the tests is considerably greater than for other types of units and of more general significance.

a. Measurements. In altitude chamber tests of closed vented units, temperature measurements to be made must be considerably more extensive than for other units for which the purposes of the tests are more limited. A complete exploration of surface temperatures on components, chassis, and case is required and should be performed by means of thermocouples installed in accordance with the methods outlined in Chapter III.

b. Test Runs

If the case of the unit is so designed that small pressure differences between opposite surfaces would not cause air flow through the unit, no particular precautions are necessary in installing the unit in an altitude chamber. However, if the possibility exists that the air circulation within the chamber may affect the convection conditions within the unit's case, care must be exercised that suitable deflection baffles are installed so that the circulating blower of the chamber does not create pressure differentials between the various surfaces of the case.

The test runs must be designed to determine a correlation of component temperatures with case temperature for each environmental pressure.

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The test runs must be designed to determine a correlation of component temperatures with case temperature for each environmental pressure.

Therefore, it is necessary to perform at each chamber pressure several test runs, each at a different chamber temperature. The lowest air temperature to be used at any chamber pressure would be that predicted for the installation compartment. By using higher air temperatures within the chamber, the air circulation effect of the chamber which tends to produce an increased heat transfer coefficient at the case surface is counteracted so that for a given ambient pressure case surface temperatures are produced which would be encountered if the unit were installed in an environment of less turbulent air motion but lower temperature.

5. Tests of Vented Units with Open Case and Through-Flow of Atmospheric Air by Natural Convection

Altitude chamber tests of this type of unit are required to give a direct indication of the operational thermal performance under installation conditions. It is not practical to apply analytical methods to the interpretation of test data or their correction for environmental conditions different than those for which the tests are performed. Therefore, accurate prediction of operational thermal performance on the basis of altitude chamber tests is feasible only if the environmental conditions of the intended installation can be reproduced. In some instances it is feasible to simulate, in addition to pressure and temperature conditions, also the location of a unit relative to others in an aircraft compartment. However, the state of air motion in such compartments is extremely difficult to predict. Frequently, the exact installation conditions of a unit in reference to other equipment items are not known and the study of the unit must be based merely on pressure and temperature specifications. Under such circumstances, the effect of adjacent units which may greatly reduce radiant heat transfer from the external surfaces of the unit to be tested cannot be taken into consideration. However, every effort must be made to produce in other respects the most severe environmental conditions which would be expected, although considerations of surrounding surfaces at considerably higher temperatures than the unit would not be included. Otherwise, an environment must be created within the altitude chamber which would insure, as nearly as possible, free convective conditions around the unit and a relatively poor radiation environment. The latter not being due to high surface temperatures but principally to a highly reflective environment. In accordance with the above requirements, a principal task in the performance of tests on such a unit is the elimination of effects of air circulation within the altitude chamber on the unit's mechanism of heat dissipation.

To provide the necessary test environment within the altitude chamber, the construction of a double-box enclosure is necessary. The outer box would be similar in size to the inside of the altitude chamber but sufficient space would remain all around it for circulation of air within the chamber. This box would be closed but vented so that its internal pressure would be that of the chamber. The air flow within the chamber would not enter this box but would pass over its outer surfaces and thus remove heat from it. Within this outer box another enclosure would be installed with clearances from 3 to 5 inches between all adjacent walls. The inner en-enclosure must have louvered openings in the top and in the upper third of all vertical sides. Other openings should be placed around the bottom perimeter of the inner enclosure to allow air circulation without direct impingement of the air on the surfaces of the unit being tested. A unit being tested in a given enclosure should have dimensions not exceeding one-third of the corresponding dimensions of the inner enclosure.

The surfaces of the outer box should be painted inside and outside with a flat finish, preferably black, to promote radiant heat transfer. The external surfaces of the inner enclosure should also be painted. The internal surfaces of the inner enclosure should either be painted or should be of polished aluminum. The polished surfaces would provide a less favorable environment and would give higher test temperatures. If reflective inner surfaces are used, polished radiation shields should be placed outside all louvered openings of the inner enclosure so as to avoid heat dissipation through the louvers by direct radiation to the internal surfaces of the outer box. The use of both types of internal surfaces of the inner enclosure in the test of a given unit may be desirable to ascertain the effects of changes in the radiant heat dissipation from the case of the unit. It may not be practical to modify the surface characteristics of a given inner enclosure. Instead, it may be more desirable to construct two identical inner enclosures, one with painted, the other with polished internal surfaces.

a. Measurements

Measurements to be made on this type of unit are the same as those required for vented units with closed case since the tests to be performed in the altitude chamber provide the sole means for prediction of thermal operating conditions and for determination of the unit's effectiveness of thermal design. In addition, other measurements of temperature and pressure are required to define better the environment of the unit during the tests.

It is of importance to measure the air temperature within the inner test enclosure by means of several shielded thermocouple probes located at various levels and several inches removed from the unit. The mean of the air temperature so determined is the control temperature describing the unit's environment. This temperature would differ from the air temperature as given by the altitude chamber's permanent measuring device since a temperature difference must exist to transmit the heat generated by the unit through the external surfaces of the outer box. The surface temperatures of the inner enclosure which should be determined by means of several thermocouples embedded in all surfaces would describe the radiation environment of the test unit.

b. Test Runs. One test run is sufficient for each specified combination of pressure and temperature which must be obtained within the inner enclosure. Considerable time may be required for equilibrium conditions to be established because of the thermal barriers existing between the inner enclosure and the chamber. For each operating condition, component temperatures should be observed with the inner enclosure having a reflective surface. A duplication of each test run would be necessary if it were desired to ascertain the effects of improving the radiant heat dissipation of the unit's case. This can be done, as mentioned above, by replacing the inner test enclosure with one having painted internal surfaces.

6. Tests of Vented Units with Open Case and Forced Through-Flow of Atmospheric Air

Units of this type should be tested in altitude chambers for the same reason as other blower-cooled units. The objective may be either to determine the flow rate delivered by the blower under given conditions of pressure and temperature, without knowledge of the blower characteristics, or the objective may be to correlate component temperatures directly with environmental conditions defined by air pressure and temperature. In the first instance, it is necessary to prevent heat losses from the external surfaces of the unit since the air flow rate must be determined by heat balance. In the second instance, it is also desirable to eliminate as completely as possible the effects created by the air circulation pattern within the altitude chamber. In any event, the same precautions in placing the unit in the chamber must be observed as for other forced-air cooled units. Even if the influence of the air velocity within the chamber on the discharge characteristics of the unit's blower are eliminated, cooling of the external surfaces of the unit by the air circulation within the chamber would give slightly lower component temperatures. However, the error so introduced would depend entirely on the type of unit and its design characteristics. For example, in a unit of high heat concentration requiring a sizable through-flow of air created by the blower, component temperatures would be affected only slightly by air circulation within the chamber. On the other hand, in a unit having small heat concentration and, consequently, requiring a small through-flow of air, component temperatures may be affected appreciably by air circulation within the chamber. The air circulation may boost the flow through the unit appreciably or may reduce the case temperature to an extent that sizable heat dissipation from components to the case would occur.

Frequently, it is not feasible to insulate the case of this type of unit to prevent external heat loss since the air inlet and outlet openings might be so arranged that the pressure drop characteristics of the flow passage would change if insulation were applied externally. The only type of forced through-flow unit suitable for application of insulation to the case is one which has essentially a single inlet and outlet opening. Therefore, if the altitude chamber test is intended to provide for a reliable determination of the air flow rate from heat balance calculations, to be used subsequently with bench test data for the prediction of component temperatures, or if a unit has a relatively small rate of forced through-flow and has several inlet or outlet openings, an auxiliary test box should be used within the chamber. A schematic diagram of this test enclosure containing a unit with an induced-flow blower is shown in Figure IV-1. The unit to be tested is installed within the box and the inlet or the outlet, whichever has a single opening, is connected to the surface of the box, as shown in Figure IV-1. On the opposite side of the box another opening is provided to which an auxiliary blower, preferably of the axial-flow type, is attached. This blower would be built into a duct equipped with a regulating damper. For tests of a unit having induced flow and a single discharge, and thus exhausting air from the test box, the auxiliary blower would be so installed that it would discharge into the test box. By means of the control damper regulating the flow from the auxiliary blower, the pressure within the box would be held equal to that in the surrounding space of the altitude chamber. For tests of units having forced flow and discharging air into the test box through multiple openings, the auxiliary blower and its duct would be reversed so that it would exhaust from the test box and would maintain within the box the same pressure as in the surrounding space of the altitude chamber. Thus, the blower of the unit would operate essentially under the same conditions as in an installation compartment with a pressure equal to that within the altitude chamber. The inner surfaces of the box should be lined with highly reflective material such as aluminum foil. The outer surfaces require a thick layer of insulation covered externally with a reflective material so that heat loss from the box is practically eliminated. When the auxiliary blower discharges into the box, it is of particular importance that screens be installed 6 to 10 inches from the discharge opening so that the flow is diffused and does not impinge directly on the surfaces of the test unit. A deflection baffle may also fulfill this requirement. In general, the dimensions of the box in all directions should be approximately three-times the dimensions of the unit to be tested. Shielded thermocouples must be installed at the inlet and outlet of the box so as to provide for an accurate measurement of the temperature rise of the cooling air passing through the unit.

a. Measurements. In addition to the general measurements required to determine the operating characteristics of the unit, it is important to

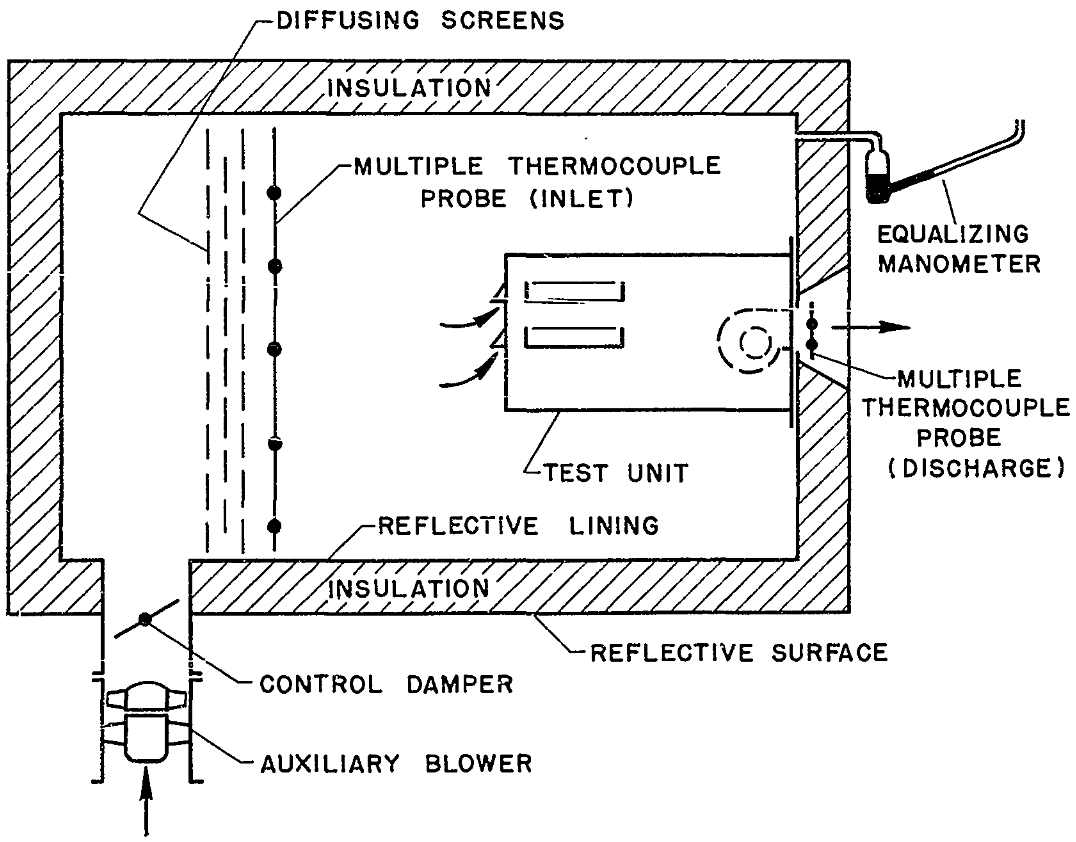

Figure IV-1. Schematic of Enclosure for Altitude Chamber Tests of Vented Units with Forced Through-Flow of Air and Multiple Openings

measure, as mentioned above, inlet and outlet temperatures of the air temperatures of the air traversing the test box, the inside surface temperatures of the test box, and the pressure within the test box in reference to the pressure within the altitude chamber. For pressure measurement, a manometer should be installed within the altitude chamber so as to be visible through the inspection window. It should be connected to a manifold with lines to several static pressure taps in the wall of the test box, all located in one plane around the box' periphery. These measurements of temperatures and pressure apply specifically to tests which are made for the purpose of determining the air flow rate through the unit. Component temperatures would be calculated on basis of the correlation between air flow rate and temperature rise established in bench tests. Therefore, it is not necessary to measure component temperatures in these tests.

In tests of vented units with forced through-flow having the primary purpose of correlating component temperatures directly with pressure and temperature of the cooling air supplied to the blower, measurements of component temperatures in the usual manner are necessary. Whether such tests would be performed using the test box described above would depend on the magnitude of the air circulation through the unit in comparison with the external air circulation pattern within the altitude chamber. The box would, however, only be used for units which cannot be insulated externally because of multiple openings in several sides of the case and the indefinite effect the insulation configuration would have on their flow pattern.

Measurements to be made on units with inlet and outlet at opposite ends, and all others which may also be insulated, are fewer because no pressure balance need be made. However, the air temperatures at inlet and outlet must be measured as accurately as possible. This represents a problem for units with multiple outlets because it is difficult to obtain a true mixed temperature. For such units, flow determinations by heat balance are not reliable.

b. Test Runs. One test run must be performed for each combination of ambient pressure and temperature for which either component temperatures, or the air flow rate of the blower are to be determined. When the test box illustrated in Figure IV-1 is used, the necessary adjustments in the control damper of the auxiliary blower must be made so that pressure equilibrium exists between the test box and its environment within the chamber. Since data are desired at thermal equilibrium, each test could be terminated when the observed temperature rise of the air passing through the test box becomes constant. However, it may be desirable to check attainment of equilibrium by the measurement of at least one or two component temperatures. In other tests, with or without the test box, in which all component temperatures are measured, equilibrium is ascertained on basis of component temperatures rather than air temperatures.

For units restricted to test in the box, data for the determination of external heat transfer characteristics cannot be obtained. Others that may be insulated furnish such data if at each environmental condition two test runs are made. One with heavy insulation with as little heat loss as possible, the other with bare case.

CHAPTER V

BLOWERS FOR AIR-COOLED EQUIPMENT

Design, modification and experimental evaluation of electronic equipment, air-cooled by forced convection, requires familiarity with the various types of applicable blowers, a thorough understanding of their operating characteristics, and the ability to determine their performance under specified operational conditions.

Cooling Blower Applications

Blowers used as source of air motion for the cooling of electronic equipment fall into two general categories. They are (1) internal devices in closed equipments, producing circulating air flow over the components, and (2) devices supplying external cooling air for the dissipation of heat by forced convection from the ultimate heat transfer surfaces of the equipments. Internal devices of the first category are employed to establish uniform thermal conditions within the equipment. They aid in the transfer of heat from components to the equipment's case surfaces, or to other heat exchange surfaces which are utilized for external heat dissipation. However, whether the external heat dissipation takes place by free convection, forced convection, radiation, or a combination of these modes, the selection of the internal blower is influenced only indirectly. The principal requirement the blower must fulfill is that the air distribution within the equipment is suitable for heat dissipation from individual components. The devices of the second category are used with open or closed equipments. They may produce cooling air flow directly over the surfaces of components, or they may supply cooling air to external heat dissipating surfaces, such as the case surface proper or extended surfaces forming a case-envelope or separate heat exchanger.

The internal and external phases of heat dissipation are in many instances divorced with regard to blower selection and application. In choosing a blower for internal air circulation in pressurized or closed vented equipment, the selection is principally affected by the pressure level within the equipment and the air flow and distribution requirements derived from a knowledge of component hot spots, their severity and their locations. With pressurized equipments, the basic problem of proper cooling under variable operational conditions rests primarily upon the ability to provide adequate heat dissipation from the case surface. The requirements imposed on internal blowers of such equipments are constant for all operational conditions. Therefore, such internal blower-motor units may be selected to operate at constant speed, which would result in the same air circulation rate under all operational conditions since the internal pressure level of such equipments would remain essentially fixed. With closed vented equipments, internal blower-motor units having no means of control may be used, providing that at all operational conditions the external heat dissipation from the equipment case is sufficient to prevent overheating of components contained within the equipment. However, since the pressure level within the equipment is reduced as the equivalent altitude of operation increases, the weight flow capacity of the internal blower and its ability to improve the heat distribution within the equipment are diminished. Under equipment operating conditions where components tend to reach their temperature limits, the problem of inadequate air circulation within the equipment, resulting from a reduced pressure level assumes importance since, even without improving the external heat dissipating characteristics, reductions in component temperature may be produced by increasing the flow rate of the internal blower. In such a situation a requirement may exist for control of the internal blower to provide an increased calculation rate. However, for the general application of internal blowers to aid in the cooling of closed vented equipments, provisions for control are not of importance.

Among blowers of the second category, used to supply air for direct dissipation of heat either by flow over components or over the surface of the equipment, the demands placed on their operational characteristics are considerably more severe. In many instances unattainable variations in air flow requirements may result if the components are to be sufficiently cooled over a wide range of operational pressure levels. As is shown in the sections of Chapter VI, dealing with units cooled by forced convection, careful evaluation of the equipment's heat transfer and pressure drop characteristics is required as basis of blower evaluation or selection. Knowledge of the variation of system pressure drop and air quantity, required to provide adequate cooling in the prescribed range of operational conditions, is essential to determine the adequacy of a blower which is to serve as air source for ultimate or external heat dissipation. The methods available for defining system characteristics for different classes of equipment based upon bench tests, are outlined in Chapter VI. Once the equipment's characteristics are established, the cooling performance of a blower and drive unit of known characteristics may be predicted, when operated with the equipment under specified conditions of air temperature and pressure. If a blower is to be selected, its choice should be made on basis of the calculated equipment cooling air requirements at the maximum altitude of operation. The characteristics of its drive unit to be selected would depend on the degree of temperature control desired at other altitudes to prevent over- or undercooling. The control requirements are severe, particularly if the desired range of operational altitudes is appreciable. The selection of a blower-motor unit that passes the proper quantity of air over or through the equipment for all conditions of operation, without introducing excessively complicated controls, is a difficult problem to solve.

Evaluation or selection of an external cooling air blower must be made on basis of the required air flow volume and head production. For operation at maximum altitudes above 40,000 feet these two requirements would usually both correspond to the highest altitude. However, if high air temperatures must be considered at low altitude, the maximum required air flow volume may occur at that point. As a rule, the requirements at maximum altitude would govern and adequate cooling would then be available at lower altitudes. Without control, over-cooling of the equipment would result, attended by excessive power requirements. Therefore, if practically feasible, provisions for blower control to maintain the equipment at or near its maximum permissible operating temperature are desirable. However, in selection and in evaluation with a view towards modification, a choice must be made from knowledge of the system characteristics for all conditions of operation, not only to define the requirements at the proposed maximum altitude of operation, but also to allow necessary compromise that invariably must accompany acceptance or selection of a blower-motor unit for cooling.

In a subsequent section of this chapter, beginning page principles for the control of cooling blowers are described to familiarize the electronic designer with various possibilities. Some automatic control systems can be assembled from commercially available parts. Others contain elements which could be perfected with a relatively small development effort.

Types of Blowers

The two basic types of blowers suitable for the supply of external cooling air to airborne electronic equipment are (1) centrifugal blowers and (2) axial-flow blowers. These types are also applicable as internal circulating devices. Since most internal circuits have low pressure drop, the axial-flow blower is often preferred for such purposes, for reasons apparent from the subsequent description of its characteristics.

1. Centrifugal Blowers

The centrifugal blower consists of an impeller rotating within a scroll-type housing and depends upon the action of centrifugal force and conversion of kinetic energy for the production of air pressure. Air enters the blower in an axial direction through a single or a double inlet, i.e., at one or both sides in the center of the impeller, is turned, and flows radially outward, discharging into a scroll. The scroll is eccentrically mounted around the periphery of the impeller and serves the purpose of collecting the air and allowing its discharge from the blower through a single outlet. A typical single-inlet centrifugal blower is shown in Figure V-1. This unit is of the type generally employed as a device for distributing and circulating air inside of equipments and throughout compartments. By the nature of its construction, impeller tip speeds are limited to relatively low values so that only small pressures can be generated with it. However, it has appreciable volumetric air capacity per unit of space occupied. Thus, whenever flow resistance is low and the air density is at least comparable to that at sea-level, this type of blower is readily adaptable to equipment requiring considerable internal air circulation or agitation. For a given impeller speed and diameter, a double-inlet blower of the same type has greater air capacity and roughly the same pressure-producing ability as the single-inlet unit. Figure V-2 illustrates the type of centrifugal blower capable of producing appreciable air-pressure rise. In comparison with the blower of Figure V-1, the main difference lies in the construction of the impeller, which is such that the very much higher peripheral speeds necessary for greater pressure generation are permissible. For equipments cooled by forced convection and operating at high altitude, the required pressure-

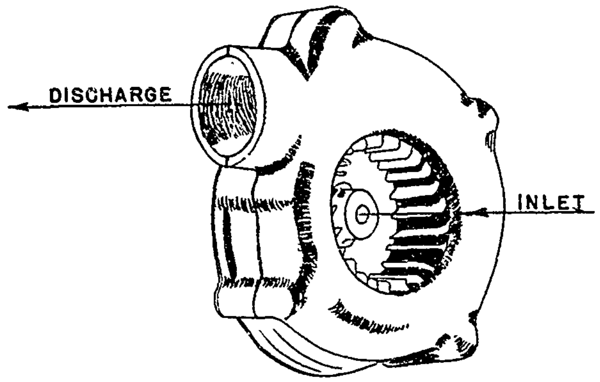

Figure V-1. Centrifugal Blower with Low Discharge Pressure

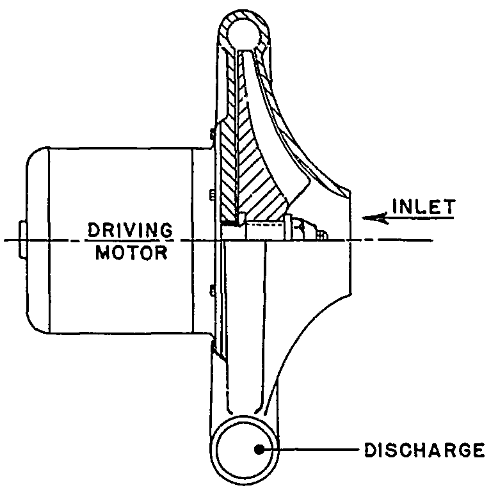

Figure V-2. Centrifugal Blower with High Discharge Pressure

producing ability of the blower is always quite large. Thus, when a centrifugal unit is to be employed at high altitude, it should be of the type shown in Figure V-2.

Centrifugal blowers may be grouped, according to the shape of the vanes of the rotating impeller, into three general classes. They are (1) forward-curved vanes, (2) radial vanes, and (3) backward-curved vanes. Forward-curved vanes curve away from the radial direction, in the impeller's direction of rotation. Backward-curved vanes curve opposite to the direction of rotation. The performance of centrifugal blowers may be classified in accordance with these general directions of vane curvature. For a given rotational speed and impeller diameter, the pressure-producing ability of the forward-curved vane impeller is the greatest, that of the radial-vane impeller intermediate, and that of the backward-curved vane impeller the smallest. Hence, from the standpoint of utilizing the smallest possible blower for a given required air pressure the forward-curved vane unit is superior. However, forward-curved vane blowers have relatively poor stability of operation and allow considerably less flexibility in control designed to avoid overloading the drive unit and pulsating discharge. Backward-curved vane impellers are best suited to attain these objectives, although radial-vane units have also proven satisfactory. From the viewpoint of mechanical strength of the impeller, the radial-vane unit is far superior to the two other types. Mechanical strength is particularly important for units handling atmospheric air at high altitude since they must operate at relatively high impeller tip speed even if the required pressure production is nominal. For a given discharge velocity and static pressure, the pressure rise in the diffuser or discharge scroll must be greatest for forward-curved vane units which have the highest resulting air velocity at the impeller periphery. This static pressure rise resulting from the reduction of air velocity is intermediate for radial-vane and smallest for backward-curved vane units. Considering all the advantages and disadvantages of the various types, the radial and forward curved vane impellers are best suited for centrifugal blowers for cooling of electronic equipment. The backward-curved vane type should only be employed for conservative designs in which size and pressure production are not critical. The radial-vane type should be employed when appreciable pressure production is required from a reasonably small unit having characteristics which make fairly good control possible. In most applications the forward-curved vane type would prove best because of its greater pressure producing ability for a given speed and size. However, proper care must have been taken that the possibility of over-loading the drive motor, due to reduction in flow resistance, is avoided.

2. Axial-Flow Blowers

The axial-flow blower depends entirely upon the conversion of kinetic energy for the production of air pressure, since air flows through it parallel to the axis of rotation of the impeller or propeller. The purpose of the rotating propeller is to increase the kinetic energy of the air stream for subsequent conversion into static pressure. Units of this type capable of producing appreciable pressure, must have several rotating elements and are called multi-stage blowers. However, the general designation of axial-flow "blower" or "fan" usually infers a single-stage unit. Two general types of single-stage axial-flow blowers, each with its drive motor, are illustrated in Figure V-3. Unit (a) illustrates the type of axial-flow or propeller fan employed as a circulating device. Unit (b) is of a type of greater pressure-producing ability and is applicable to certain equipments operating

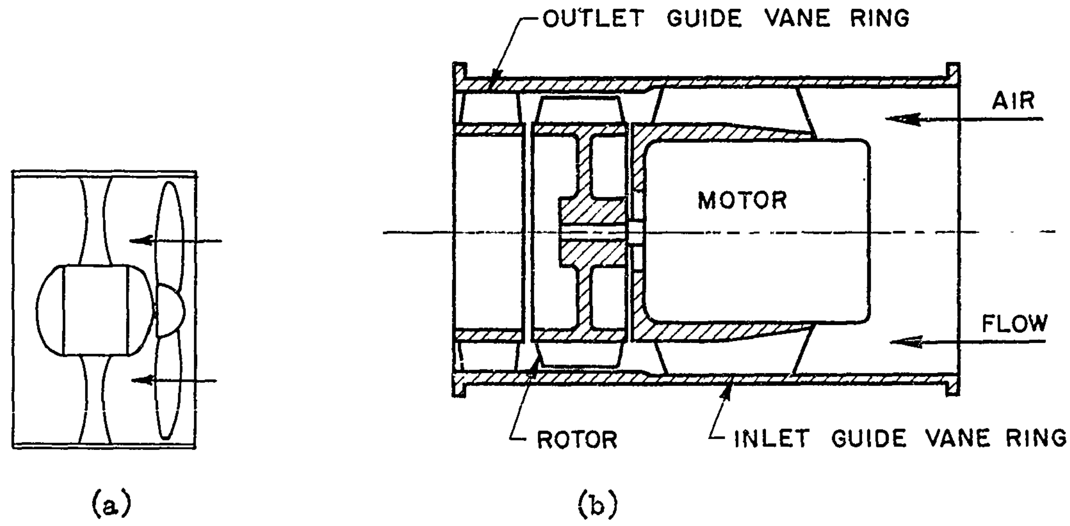

Figure V-3. Axial-Flow Blowers

in an atmosphere at a pressure corresponding to relatively low altitude, and cooled by high-volume forced air convection. For operation at high altitude a single-stage unit of this type is usually not capable of producing the required pressure for forced convective cooling. If an axial-flow blower is to be employed under such conditions it would generally have to be a multistage unit.

In comparison with centrifugal blowers, axial-flow blowers possess the definite advantages of higher operating efficiency and greater volumetric capacity for comparable external diameters. They have the disadvantages of inferior control characteristics and must assume complicated multi-stage configurations in order to be capable of appreciable air pressure production. Single-stage axial-flow blowers are no more, if not possibly less complicated, than centrifugal blowers, but are capable of delivering air at low pressure only. For higher delivery pressure, as required by electronic equipment operating in very low-density air, multi-staging is absolutely necessary and results in the impediment of practical control procedures.

When used for the delivery of large quantities of air to systems having very low flow resistance and relatively constant demands on volumetric and pressure performance, the use of an axial-flow blower as a device for cooling of electronic equipment is practical. Its configuration allows rather simple installation in air ducts without change in flow direction, or in side panels of equipment cases without appreciable external projection. It can be mounted more conveniently than a centrifugal blower for producing induced flow over the equipment's components. This feature is of importance when it is desired to improve internal air distribution without the use of extensive baffling.

Performance of Blowers

The evaluation or selection of a blower for a specific application may be made experimentally. Operation of an electronic unit with any blower under specified conditions of air supply and environment will disclose whether the blower is suitable to maintain component temperatures within prescribed limits. If similar tests are performed under all probable operating conditions the necessary information is obtained without formal knowledge of the blower characteristics and the heat transfer characteristics of the equipment. This method of attack, as applied to blower selection, is one of trial and error. It is most direct where a blower's adequacy is to be ascertained under operational conditions of increased altitude and/or air temperature. However, extensive test facilities are required to reproduce probable operating conditions.

Instead of experimental evaluation or selection, methods of analysis may be used which permit determination of equipment thermal conditions based on known heat transfer and pressure drop characteristics of the equipment and performance characteristics of the blower obtained at some reference condition of air supply. Performance data of blowers are generally available from their manufacturers on basis of test data obtained by standard procedures. These procedures and their modifications, suitable to determine the characteristics of a blower, as furnished, are given in Appendix V.

1. Catalog Data

The performance data most commonly given by manufacturers of blowers, use speed of rotation in revolutions per minute, volumetric discharge in cubic feet per minute, static pressure in inches of water, and horsepower required to drive the blower, all at a standard air density, to show characteristics. The static pressure is actually not the entire pressure produced by the blower since the kinetic energy due to the air's discharge velocity can also be converted into a pressure, called the velocity pressure, as discussed on page 335, Appendix II. The total or dynamic pressure, being the sum of the static and velocity pressure, is also given at times. However, particularly for blowers which at ground level produce static pressures of more than 10 inches of water, the velocity pressure represents a small percentage of the total pressure. Therefore the static pressure is the principal quantity indicating the ability of the blower to deliver air flow through an equipment.

The air density used as standard in the representation of blower performance data is always calculated at sea level pressure of 29.92 inches of mercury, but the reference temperature may differ. The practice for blowers principally designed for ground application has been to use a reference temperature of 70°F which gives a reference density of 0.075 pound per cubic foot. Data for aircraft blowers are sometimes given for the above reference density, but are also frequently given for 0.0765 pound per cubic foot corresponding to a sea level air temperature of 59°F (15°C) in accordance with the N.A.C.A. standard atmosphere, as given in Table A-I-2 in Appendix I.

Performance data released by manufacturers are found in graphical and/or tabular form. The most general graphical presentation of blower characteristics shows the variation of static pressure and horsepower as a function of discharge volume at constant speed. Often the efficiency, based on static or total pressure, and the total pressure are also included. Typical plots for a forward-curved vane and a backward-curved vane centrifugal blower are shown in Figure V-4. The performance of axial-flow blowers is shown in the same manner, but may give the total pressure more frequently since, particularly for single-stage units, the ratio of velocity pressure to static pressure is appreciably greater than for centrifugal blowers.

Another method of showing blower characteristics is by means of multi-rating tables which are also given for constant air density. Table V-1 contains an example of a multi-rating table for a blower designed for ground-level application in duct systems. For such application outlet velocity and tip speed of the impeller are of importance because the audible noise level must be held as low as practical. Therefore, these values are given in the table together with the other variables previously mentioned. An example of a multi-rating table for an aircraft blower is given in Table V-2 which gives basically the same type of data as Table V-1, but arranged in different manner. Discharge velocities and tip speeds can be determined from Table V-2 only by calculation.

Data as shown in Table V-2 are also given in condensed graphical form by some manufacturers to facilitate interpolation. A plot of this type

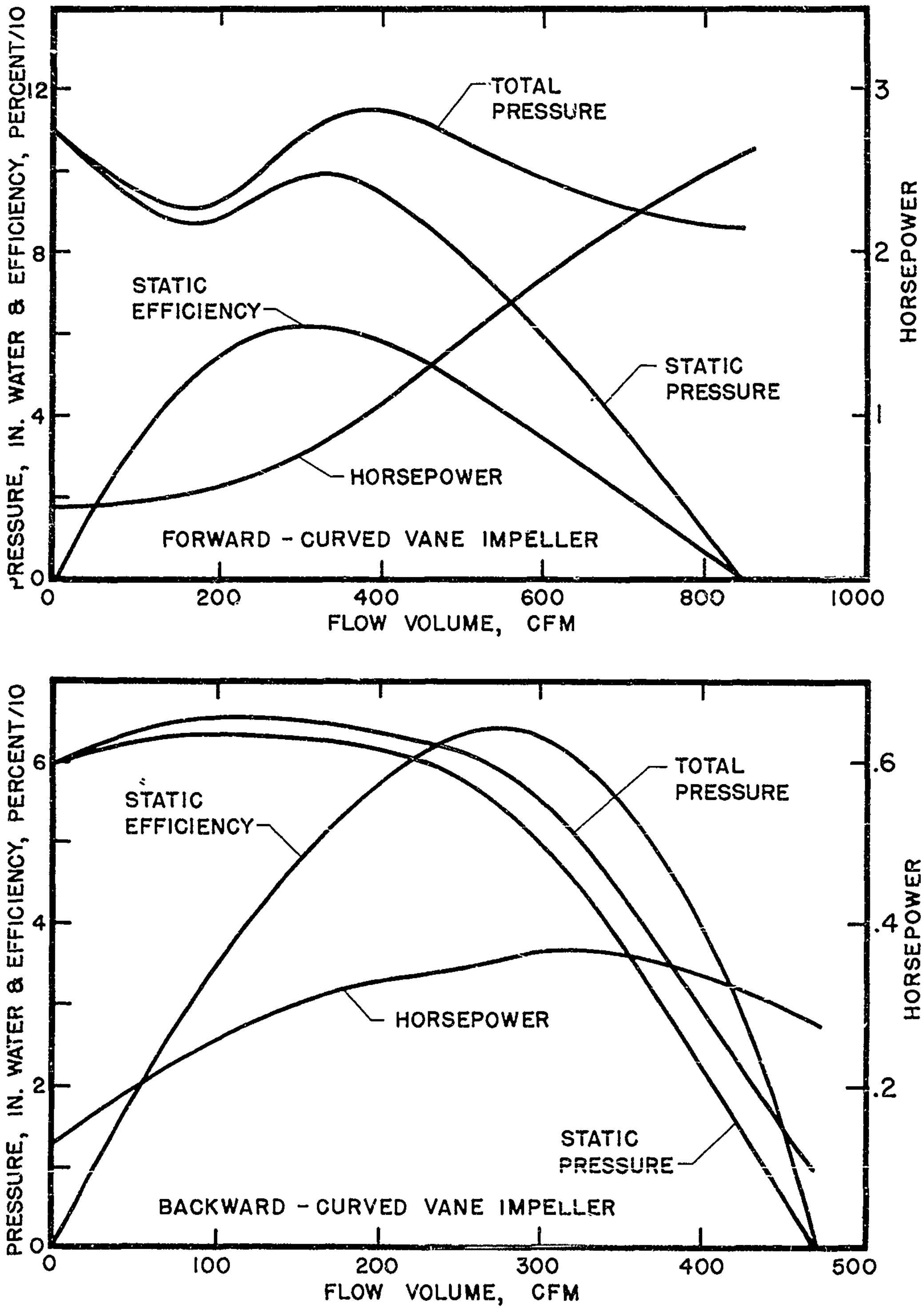

Figure V-4. Typical Performance Characteristics of Centrifugal Blowers at Constant Speed

Table V-1. Multi-Rating Table for Ventilating Blower

Static pressure in. water		1/8			3/8			5/8		
cfm	Outlet velocity ft/min	Tip speed ft/min	rpm	hp	Tip speed ft/min	rpm	hp	Tip speed ft/min	rpm	hp
1745	800	1090	278	0.08						
2618	1200	1322	337	0.17	1800	459	0.30	2205	562	0.45
3490	1600	1598	407	0.36	1995	509	0.51	2350	600	0.69
4360	2000				2240	572	0.83	2550	650	1.03
5235	2400				2510	640	1.31	2780	709	1.5(

Table V-2. Multi-Rating Table for Aircraft Blower

2 in. water Static pressure			4 in. water Static pressure			6 in. water Static pressure			8 in. water Static pressure		
cfm	rpm	hp	cfm	rpm	hp	cfm	rpm	hp	cfm	rpm	hp
500	2920	0.06	142	4120	0.16	174	5050	0.29	200	5840	0.44
142	3150	0.09	201	4440	0.25	246	5475	0.46	284	6300	0.72
184	3460	0.13	261	4880	0.38	320	5990	0.69	368	6920	1.06
243	3950	0.23	343	5580	0.64	422	6880	1.16	486	7900	1.81

as shown in Figure V-5 for a centrifugal blower permits the determination of a blower's discharge and power requirement for any speed of rotation and equipment resistance at the air density for which the rating plot is made.

Compact blower-motor units for aircraft application with d.c. drive motors operating at constant motor voltage may be subject to some speed variation with change in system resistance. Characteristic curves for such units are given for constant density and would contain also an indication of the speed variation, the motor current, and the set efficiency, the latter being defined by the ratio of the energy imparted to the air, enabling it to overcome flow resistance, to the electrical input. A typical plot of this type for a single-stage axial-flow blower-motor unit is shown in Figure V-6.

The application of the catalog data described in the preceding paragraphs for the evaluation and selection of blowers at other air densities, but at the same speeds, is simple since it involves only correcting pressure rise and power in direct proportion to the air density, while discharge volumes remain the same. Corrections for speed variation and modifications of blower dimensions can also be made and are treated more completely in a subsequent section beginning on page 90, dealing with the general laws of blower performance.

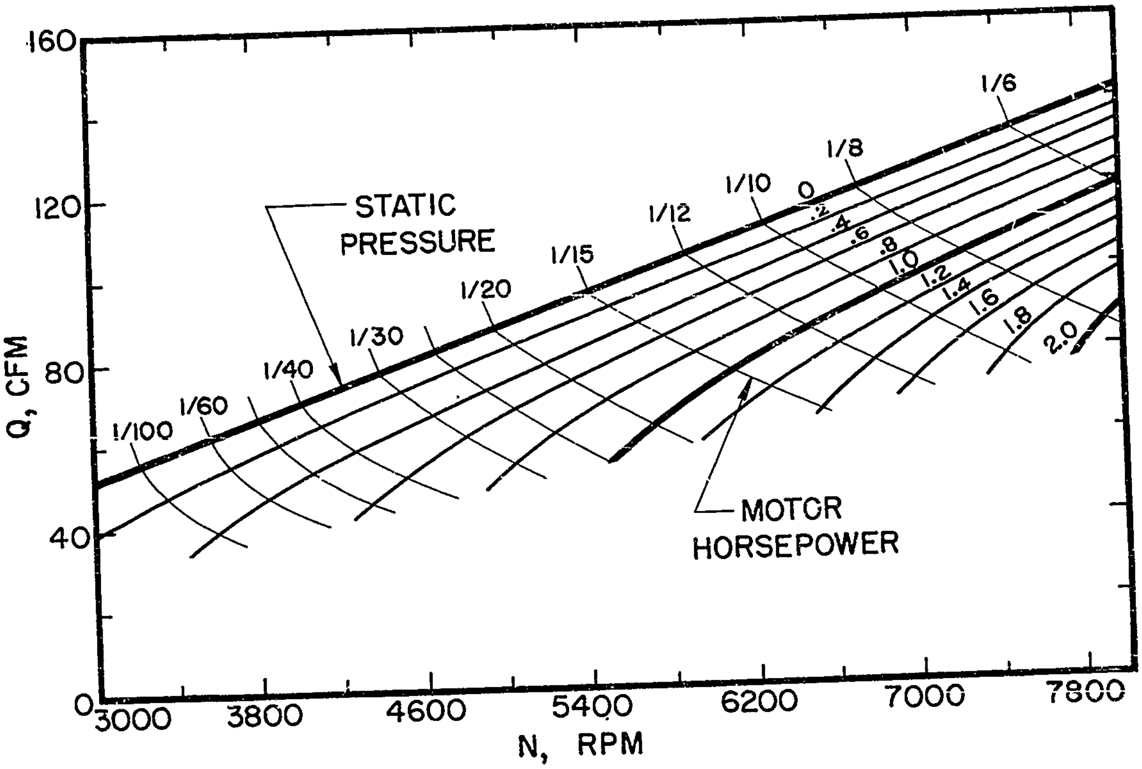

Figure V-5. Manufacturer's Plot of Centrifugal Blower Performance Characteristics

2. Control Characteristics

The performance curves for various types of blower as shown in Figures V-4 and -5 indicate various general and comparative characteristics. It is apparent that for centrifugal and axial-flow blowers alike the maximum air flow capacity at any particular speed is obtained when no static pressure is produced. Under this condition, the power requirement of forward-curved vane centrifugal and of axial-flow blowers is greatest. Consequently, it is theoretically possible that when such blowers are powered for a given equipment resistance and the flow is diverted, say through a large break in the duct, the drive motor may be overloaded. The backward-curved vane centrifugal blower cannot overload its drive motor since at wide-open discharge its power requirement is not greatest.

As the air handling capacity of the blower, usually defined in terms of the per cent of maximum air capacity, is reduced, the static air-pressure production increases and reaches a maximum at some point intermediate between zero air capacity and maximum air capacity. The point of maxi-

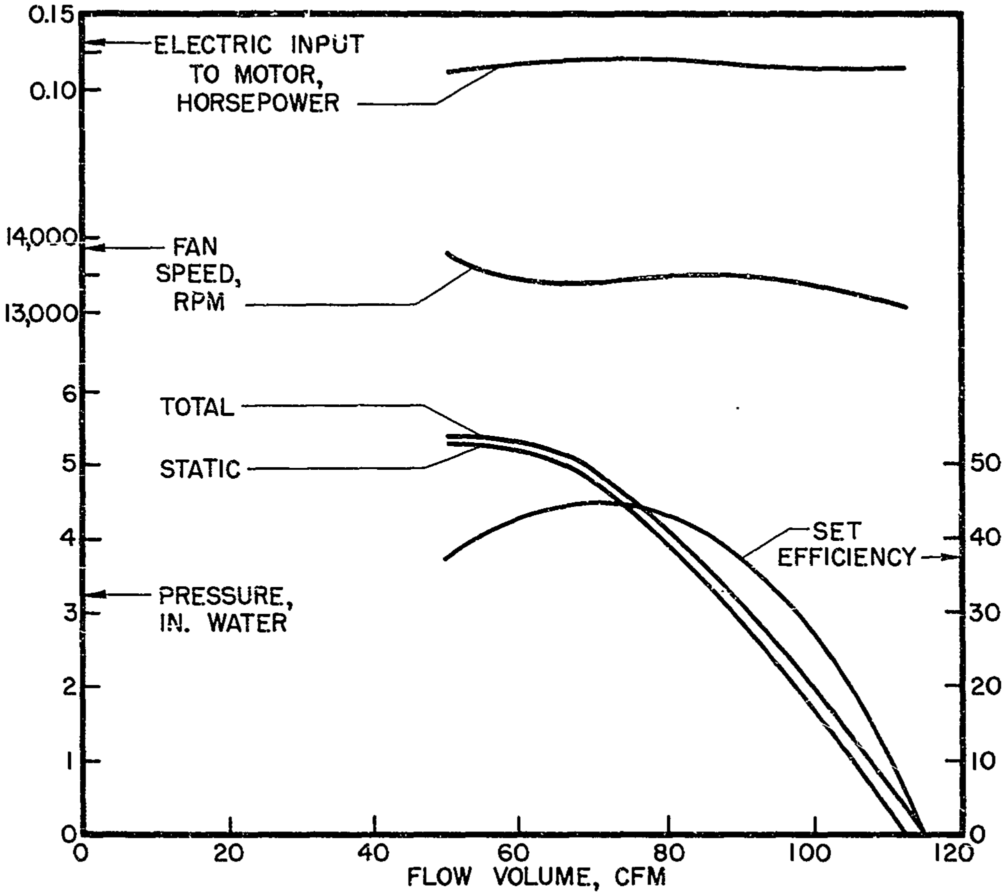

Figure V-6. Manufacturer's Plot of Axial-Flow Blower-Motor Unit Performance Characteristics

mum static pressure development, at other than zero capacity, is significant in terms of possible blower performance in that almost all blowers, whether centrifugal or axial-flow, have unstable performance when operating at lower air capacity. The instability of operation evidences itself in the form of pulsating or surging flow through the system which is usually considered to be an undesirable operating condition. For the ordinary type of centrifugal blower, the peak pressure point for finite flow volume generally occurs at a flow volume of about 25 to 40 per cent of the maximum air handling capacity. With axial-flow blowers the point lies between 50 and 70 per cent of the maximum air capacity. For blowers handling very low density air, and operating to produce a high head, this point shifts to about 50 to 60 per cent of maximum air capacity for centrifugal units and 70 to 85 per cent for axial-flow units. Since operation at lower air capacity is not considered desirable, it is apparent that for any given rotational speed and size of a blower, the permissible operating range of a centrifugal blower is appreciably greater than that of an axial-flow blower. Herein lies the basic reason why the centrifugal blower is superior to the axial-flow blower from the standpoint of control.

Possible methods for controlling blower operation along the pressure-volume characteristic line, within the permissible range of operation, are discussed beginning on page 118.

3. Data for Performance Analysis

a. Generalized Performance Plots. Density Ratio

For performance analysis at variable altitude, it is most convenient that the blower characteristics, as given by manufacturers or as obtained from tests, be generalized. In particular, plots of static pressure, horsepower, and efficiency, as shown in Figure V-4 for constant rotational speed and air density, are suitable for this procedure. The generalization of this plot does not consist in its modification, but rather in its reinterpretation. Based on the fact that pressure and power are proportional to the air density, the values shown in a plot such as Figure V-4 may be labeled as shown in Figure V-7. The values in the latter figure are corrected values which means that they have been divided by the air density ratio σ_i, defined as the ratio of the density at the blower inlet, for which the performance curves are given, to a reference density. If the reference density on which the density ratios are based is the same as that for which the performance curves are given, σ_i is unity and the curves are not changed at all. However, if, for example, a reference density ρ_r = 0.0765 pound per cubic foot is chosen to express the densities under altitude conditions, such as shown by the plot in Figure V-8 (in back pocket), performance curves given for a

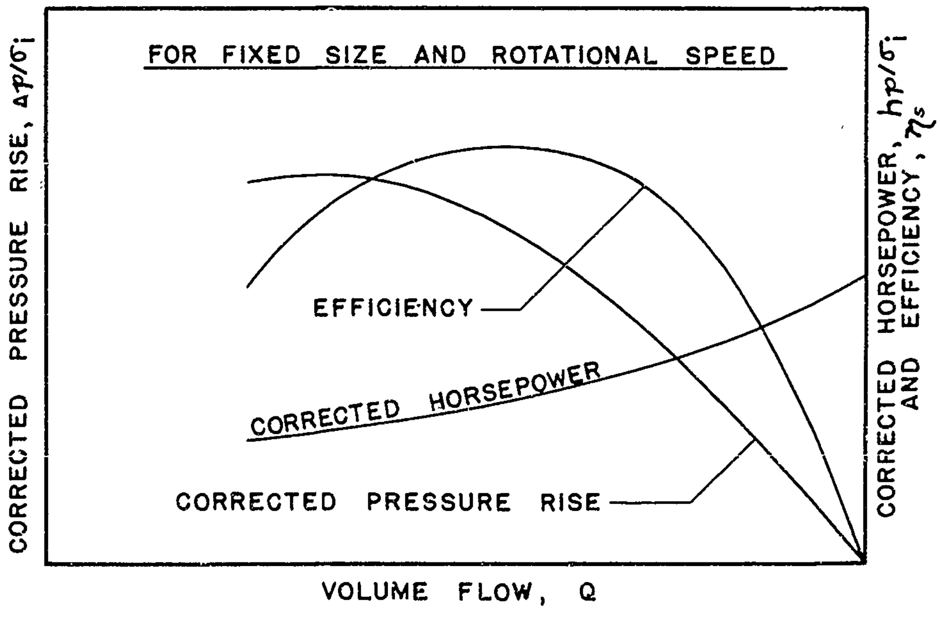

Figure V-7. Generalized Characteristic Curves for a Blower standard density of 0.075 pound per cubic foot must be corrected by $\sigma_i = (0.075/0.0765) = 0.98$. This value may be found in Figure V-8 for the condition of 29.92 inches of mercury pressure and 21°C which corresponds to the density of 0.075 pound per cubic foot. The reference density of 0.0765 pound per cubic foot is used throughout this manual since it corresponds to sea level conditions of the N.A.C.A. Standard atmosphere which is basic for aircraft work. Thus, the corrected pressures and horsepowers in a plot such as in Figure V-7 would actually be 2 per cent greater than in the original performance data. This new corrected plot may now be used with the density ratio chart in Figure V-8 to determine the pressure-producing capacity and power requirement of the blower under all operational conditions identified by air temperature and pressure, providing the speed and, consequently, the discharge volume and efficiency are fixed. If, for example, the corrected static pressure rise corresponding to a certain volume rate of flow is read from the blower characteristics as 2 inches of water, at an air pressure of 4 inches of mercury and an air temperature of 0°C, where, according to Figure V-8, $\sigma_i = 0.141$, the actual static pressure rise would be

$$\Delta p = \sigma_i(\Delta p/\sigma_i) = 0.141(2) = 0.282 \text{ inches of water}$$

The horsepower is found in the same manner. The efficiency remains the same. The values of $(\Delta p/\sigma_i)$ and (hp/σ_i) are quantities determined directly from the blower characteristics and are the pressure drop and horsepower, respectively, at the reference density ρ_r, but are applicable universally by means of multiplying with the actual value of σ_i, as indicated above.

A chart of the type shown in Figure V-7 is descriptive of the performance of a centrifugal or axial-flow blower for one speed of rotation and a single size of the blower, usually defined by the impeller dimensions. The performance so presented may be considered as the reference performance or characteristic curves of the prototype blower, and may be used for the determination of blower performance not only at other air densities but also at other rotational speeds and impeller dimensions. For such purpose, values taken from corrected performance plots, such as Figure V-7, must be modified in accordance with the laws of blower performance which are given on pages 90 to 92.

b. Weight Flow Rate

As stated above, the volume flow indicated by the abscissa of the performance plot in Figure V-7 is independent of the density of the air being handled by the blower. However, it should be remembered that, for given volume flow, the weight flow handled by the unit varies in direct proportion to the density ratio σ_i. Thus, as the density ratio decreases the weight rate of flow delivered by the blower to the electronic equipment will decrease for any fixed operating point of the blower.

Weight flow data are usually required to evaluate cooling performance of a blower with any specific equipment. Therefore, the conversion from volume flow, as given by the performance curves, to weight flow in pounds per second is usually necessary. For this purpose, the equation

$$W = \sigma_i Q/784 \qquad \text{(V-1)}$$

may be used, where σ_i is the density ratio at blower inlet based on the reference density $\rho_r = 0.0765$ pound per cubic foot, as determined from Figure V-8, and Q is the volume flow in cubic feet per minute. Equation (V-1) is shown graphically in Figure V-9 (in back pocket) which may be used instead of the equation. The range of the chart may be extended by use of a common multiplier for abscissa and ordinate.

As further discussed in Chapter VI, page 153, the pressure drop characteristics of most equipments can be generalized in terms of an equation in which the product of the density ratio at the equipment inlet and the actual pressure drop is proportional to an exponential of the weight flow. The constant of proportionality may be assumed to remain unaltered by environmental conditions. This relationship simplifies the evaluation of blower performance over a range of operational conditions, providing the necessary conversion is made from volume flow to weight flow in blower performance, or oppositely, from weight flow to volume flow in equipment characteristics. Either basis may be chosen as long as it is used to describe both blower and system characteristics.

c. Temperature Rise of Air Across Blower

A factor of considerable importance in the performance analysis of blowers supplying forced air flow through or over an electronic equipment, is the temperature rise the air undergoes while passing through the blower. This temperature rise is the result of compression and losses and is directly related to the head of the blower which indicates the magnitude of pressure rise relative to the absolute air pressure. Thus, at high altitude where the head required to produce flow through the equipment may be great, this temperature rise may become particularly significant. It is conceivable that if forced air flow must be provided at an altitude in the order of 60,000 feet to an equipment with high flow resistance, the discharge temperature of the blower may be so high that the air would not be suitable as a cooling medium. The only alternative would be to induce air flow through the equipment by installing the blower at the equipment discharge. This blower would receive air at higher temperature and lower pressure than the forced-flow blower. The blower would be larger and would require more power to drive it.

In evaluating the temperature rise the air undergoes while passing through the blower, it may be assumed that the heat loss from the blower to its environment would be practically negligible. Therefore, all the input energy of the blower would be transferred to the air and would evidence itself in form of air temperature rise. On basis of this assumption the following two equivalent equations for the calculation of the temperature rise of the air Δt_B in °C from inlet to exit of the blower are derived:

$$\Delta t_B = (1280/Q)(hp/\sigma_i), \text{ and} \qquad \text{(V-2)}$$

$$\Delta t_B = (0.202/\eta_s)(\Delta p/\sigma_i), \qquad \text{(V-3)}$$

where hp is the input power to the blower in horsepowers, and η_s is the blower efficiency based on static pressure rise. Strictly speaking the temperature rise calculated from equation (V-2 or -3) accounts for the change in total temperature, as determined when the discharge jet of the blower is caused to stagnate. In heat transfer calculations, the temperature so determined is more significant than the static temperature. However, for usual blower discharge velocities there is little difference between the total and static temperature.

For the sake of convenience in determining air temperature at the blower discharge, Figures V-10 and V-11 are presented as graphical solutions to equations (V-2 and -3), respectively. To use Figure V-10, it is necessary to proceed as shown by the set of lines with arrows, i.e., from the known flow volume Q on the abscissa vertically up to the density ratio line corresponding to σ_i, horizontally to the horsepower line corresponding to the actual power input to the blower, and vertically up to read the temperature rise Δt_B on the upper scale. The type of available blower performance data determines by which chart the temperature rise of the air may be most conveniently evaluated. Equation (V-2) and Figure V-10 are generally applicable when the necessary data are available. However, equation (V-3) and Figure V-11 are strictly applicable only to situations wherein the density of the air remains essentially constant while passing through the blower and the air may be assumed as an incompressible fluid. This relationship may be considered valid so long as the pressure rise across the blower does not exceed about 7 per cent of the absolute pressure of the air at the inlet, although its use for blower pressure rise of as much as 10 to 15 per cent of the absolute pressure at the inlet would not introduce serious error. Under conditions of very high altitude of operation, where the ambient air pressure is quite low, the pressure rise of the air across the blower may well exceed these limitations. Under such circumstances the temperature rise of the air across the blower should, if possible, be calculated from equation (V-2) or from

$$\Delta t_B = (t_i/\eta_s) \left\{ \left[1 + (\Delta p/13.55 p_i) \right]^{0.283} - 1 \right\} \qquad \text{(V-4)}$$

where t_i is the air temperature at the blower inlet in °C, and p_i is the air pressure at blower inlet in inches mercury absolute.

In addition to the required use of equation (V-4) for calculating air temperature rise across the blower when the pressure rise represents an appreciable percentage of the inlet absolute pressure and the use of equation (V-2) is precluded because power and air rate data are unknown, it is also necessary to revise the procedure for the determination of actual static pressure production. As pointed out above, for incompressible flow, which may be assumed to occur in most applications, the method of determining actual air pressure rise involves only multiplication of the corrected blower pressure rise by the actual density ratio σ_i of the air at the blower inlet. However, for compressible flow the actual blower pressure rise curve must be calculated by the relation

$$\Delta p = 13.55\ p_i \left\{ \left[1 + (\sigma_i\ \Delta p_{corr})/(47.6\ p_o) \right]^{3.53} - 1 \right\} \qquad \text{(V-5)}$$

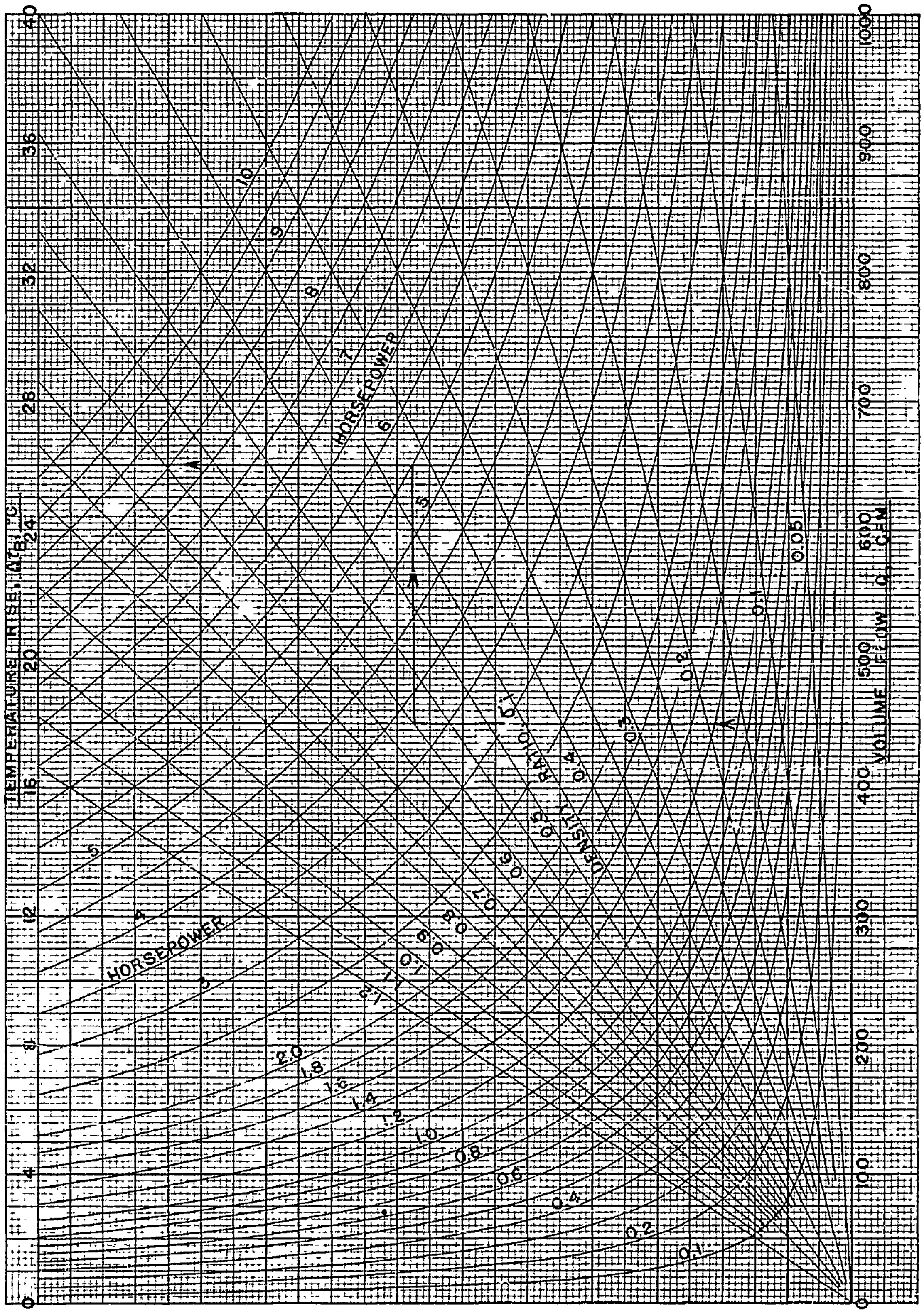

Figure V-10. Temperature Rise of Air Across Blower

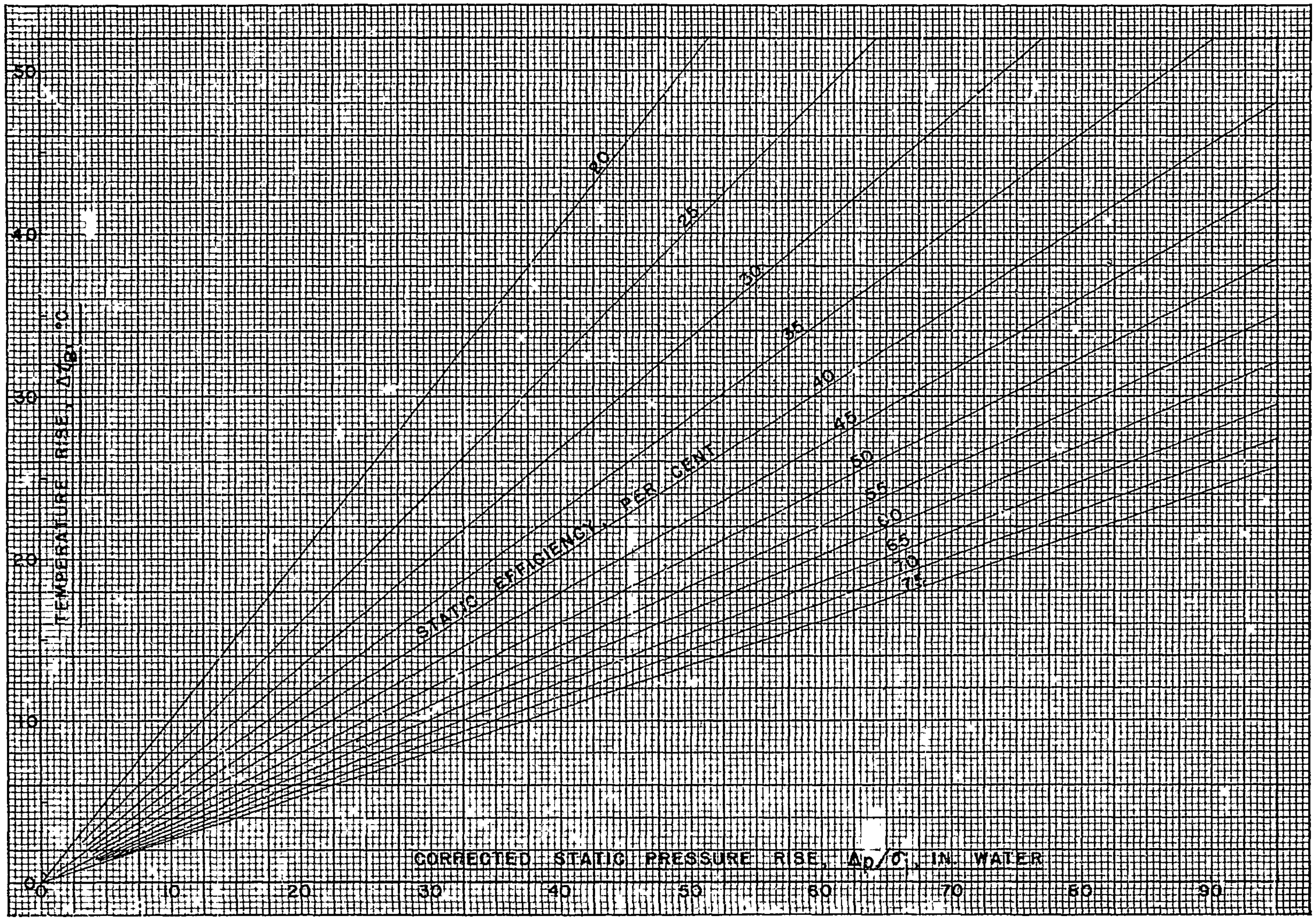

Figure V-11. Temperature Rise of Air Across Blower (for $\Delta p/p_i \leqq 0.15$)

where the value of Δp_{corr} is the static pressure rise developed by the blower for the same flow volume Q under standard reference conditions of operation where $\sigma_i = 1.0$ and $\rho_i = \rho_r = 0.0765$ pound per cubic foot.

4. Laws of Blower Performance

In evaluating any one blower, or selecting a blower from a group of homologous, i.e., geometrically similar blowers, the known laws of blower performance are useful. By their application, the operating characteristics of a combined equipment-blower system may be defined. They make possible the selection of blower-motor units to meet the cooling requirements of an equipment if blower, motor, and equipment characteristics are given. Although the exactness of these laws depends on the assumptions of dynamic and geometric similarity, in the comparison of blower performance they are sufficiently accurate and of great utility for thermal evaluation work.

The blower laws may be stated in form of the following generalized proportionalities:

$$\Delta p \quad \alpha \quad \sigma_i D^2 N^2 \tag{V-6}$$

$$Q \quad \alpha \quad ND^3 \tag{V-7}$$

$$hp \quad \alpha \quad Q\Delta p \quad \alpha \quad \sigma_i N^3 D^5 \tag{V-8}$$

$$W \quad \alpha \quad \sigma_i Q \tag{V-9}$$

where N is the rotational speed of the blower, generally expressed in revolutions per minute, and D is some radial dimension of the blower, almost always taken as impeller diameter in inches. In addition to the above-mentioned general relationships for centrifugal and axial-flow blowers, the following qualitatively accurate rules apply with variation in impeller width B of centrifugal blowers, providing impeller diameter, rotational speed and inlet air density are maintained constant: (1) volume flow rate is proportional to impeller width, (2) horsepower is proportional to impeller width, and (3) static pressure rise is independent of impeller width. The rules are of value when predicting performance of centrifugal blowers on basis of known data for blowers of equal diameter but different impeller width.

The above-stated general laws of blower performance may be broken down into the following specific cases:

a. For constant impeller diameter (blower size), air density, and impeller width, but variable rotational speed:

$$\Delta p \quad \alpha \quad N^2 \tag{V-6a}$$

$$Q \quad \alpha \quad N \tag{V-7a}$$

$$hp \quad \alpha \quad N^3 \tag{V-8a}$$

$$W \quad \alpha \quad N \tag{V-9a}$$

b. For constant rotational speed, air density, and impeller width, but variable impeller diameter:

$$\Delta p \;\alpha\; D^2 \qquad \text{(V-6b)}$$

$$Q \;\alpha\; D^3 \qquad \text{(V-7b)}$$

$$hp \;\alpha\; D^5 \qquad \text{(V-8b)}$$

$$W \;\alpha\; D^3 \qquad \text{(V-9b)}$$

c. For constant rotational speed, impeller diameter, and impeller width, but variable air density:

$$\Delta p \;\alpha\; \sigma_i \qquad \text{(V-6c)}$$

$$Q = \text{constant} \qquad \text{(V-7c)}$$

$$hp \;\alpha\; \sigma_i \qquad \text{(V-8c)}$$

$$W \;\alpha\; \sigma_i \qquad \text{(V-9c)}$$

d. For constant rotational speed, impeller diameter, and air density, but variable impeller width of centrifugal blowers only:

$$\Delta p = \text{constant} \qquad \text{(V-6d)}$$

$$Q \;\alpha\; B \qquad \text{(V-7d)}$$

$$hp \;\alpha\; B \qquad \text{(V-8d)}$$

$$W \;\alpha\; B \qquad \text{(V-9d)}$$

Any of the independent variables described above may be combined to illustrate the effect on blower performance when more than one variable is involved, such as would be the case, for example, when considering variable speed in conjunction with variable impeller width. The use of these laws for the prediction of blower performance at other than reference conditions is illustrated in Example V-1, page 124.

In order to apply these laws to the prediction of performance, it is necessary to have available as reference, the performance characteristics of a blower of known dimensions which is dynamically and geometrically similar to the blower under consideration. With these characteristic curves, given for a known speed and corrected to standard conditions, the performance of any homologous blower of different impeller, diameter operating at any other rotational speed, may readily be evaluated by the above-stated laws of blower performance for any inlet air density and, if a centrifugal blower, for any impeller width.

The application of the blower laws is necessary for many purposes such as: (1) the selection or modification of a blower to provide satisfactory cooling for a given equipment, (2) the evaluation of an installed cooling blower's adequacy based on extrapolation of bench-test performance of electronic equipment to other specified operational conditions, (3) the determination of control requirements for temperature regulation of blower-cooled equipment, and (4) the correlation of blower-performance data obtained from tests. When using these laws for the determination of blower performance with equipment at other than reference conditions, more often than not trial-and-error solutions are required which involve considerable labor and time for computation. By graphical presentation of the blower laws many problems of blower evaluation, selection and design may be performed more rapidly than by computation. The chart in Figure V-12 (in back pocket) is intended to serve this purpose. Its construction and use are discussed in the following two sections.

5. Description of Chart for Blower Performance Analysis and Design

The generalized working chart shown in Figure V-12 is a graphical presentation of the blower laws. It consists of eight quadrants with interrelated curves. The chart is applicable to centrifugal and axial-flow blowers. The individual quadrants and their functions are described as follows.

a. Grid for System Characteristics

In quadrant (1) the characteristic curve for any equipment can be inserted which would show the relationship between standard pressure drop (the product of density ratio at inlet and true pressure drop) and weight flow rate. These system (equipment) resistance characteristics are independent of blower evaluation and design considerations. They are derived from experimental data. Methods of data reduction for blower-cooled electronic units are discussed in Chapter VI and illustrated in Examples VI-5, -6, and -7.

The density-ratio lines for system resistance and flow volume shown in quadrant (1) may be used to translate the entire system resistance characteristics or any point thereon into the relationship of actual pressure drop as function of volume flow at any operating condition. defined by a specified density ratio. Thus, for any flow condition defining a point in quadrant (1) of known weight flow and standard pressure drop at specified density ratio at the equipment inlet, the actual pressure drop (system resistance) is determined by projecting vertically from this point to the appropriate density-ratio line for system resistance (σ at equipment inlet) and then horizontally to the right-hand ordinate of quadrant (1). The volume flow is determined by projecting from the point horizontally to the appropriate density-ratio line for flow volume (σ at equipment inlet) and then vertically downward to the abscissa of quadrant (1).

If in a problem of blower selection or design the equipment's required flow volume and corresponding resistance are given specifically for a certain operating condition, the operating point can be plotted directly in quadrant (1) using the right-hand ordinate and the lower abscissa.

b. Grid for Blower Characteristics. In quadrant (5), a reference blower's characteristic curves of corrected static pressure versus volume flow rate and corrected input horsepower versus volume rate should be plotted. Depending on the problem, the reference blower may be the actual blower whose performance with a given equipment is to be evaluated, or it may be the prototype for evaluation, selection, modification, or design of a similar blower which may be homologous or of different impeller width ratio. When selecting the reference performance of a blower for use in this quadrant, it may occur that the ranges of volume flow and static pressure rise are greater than can be accommodated by the scales of the grid. In that case, it is simplest to reduce the selected performance to the range of the quadrant's scales by reducing the reference speed. According to the blower laws, this reduces the range of the flow volume in direct proportion, the range of the static pressure in proportion to the square, and the horsepower in proportion to the cube of the reduced speed. The same procedure in reverse can be used to obtain the best accuracy in the use of the charts when a small blower is chosen as the reference blower. The other blower laws can also be applied to better match the available characteristic data to the ranges of the grid's scales. Particularly, the laws relating the variation of static pressure and horsepower with that of the impeller diameter are useful in magnifying the performance of a small reference blower.

c. Diameter Ratio Lines for Determination of Flow Volume. Quadrant (2) contains a graphical presentation of the blower law given by equation (V-7b) which indicates that, if all other conditions remain fixed, the air volume delivered is proportional to the cube of the impeller diameter. The diameter ratio, as shown in this quadrant, is defined as the ratio of the diameter of the actual blower being evaluated to the diameter of the reference blower whose characteristics are inserted in quadrant (5) as discussed above.

d. Width-Ratio and Density-Ratio Lines for the Determination of Flow Volume and System Resistance

The width-ratio lines in quadrant (3) are a graphical presentation of the basic assumption, given by equation (V-7d), that the flow volume is proportional to the effective impeller width of a centrifugal blower. The width ratio is defined as the ratio of the impeller width of the actual blower being evaluated to the impeller width of the homologous blower derived from the reference blower by geometrical similarity. If, for example, the impeller of the reference blower has a 10-inch diameter and a 4-inch width and the actual blower has a 5-inch diameter and a 1-inch width, then the actual blower should have an impeller width of 2 inches in order to be homologous. Since, however, the impeller width of the actual blower being considered is only 1 inch, then the width ratio to be used in quadrant (3) is 0.5. In using this chart for axial-flow blowers only geometrically similar blowers can be analyzed and, therefore, only the unity width-ratio line in quadrant (3) may be used.

The constant density ratio or σ_i-lines are based on the proportionality of blower pressure to air density for incompressible flow, as indicated by equation (V-6c). As pointed out previously, this proportionality is valid for almost all operating conditions encountered by cooling blowers of electronic equipment. Should a situation arise wherein the use of compressible-flow relationships are required, these curves are no longer usable for the determination of corrected blower pressure rise. Then, the correct $\Delta p/\sigma_i$ should be calculated by equation (V-5) and introduced on the abscissa of quadrant (3). By use of the σ_i-lines, all system pressure drops are converted to standard conditions and represent the pressure drop in the system when the flow volume at standard conditions is the same as that required for altitude conditions.

e. Speed-Ratio Lines for the Determination of Flow Volume and System Resistance. The flow-volume lines in quadrant (4) represent the blower law of equation (V-7a) which states that the delivered flow volume varies in direct proportion to the blower speed. The system-resistance lines represent equations (V-6a and -6c) combined, stating that the corrected blower pressure rise is proportional to the square of the blower speed. The lines are labeled "system resistance", although they are actually static pressure lines for the blower, because they are used in matching blower pressure and equipment resistance at a given speed ratio. The speed ratio is defined as the ratio of the rotational speed of the actual blower being evaluated to the rotational speed of the reference blower, for which the characteristics are inserted in quadrant (5). Quadrant (4) is particularly useful when determining required variation in blower speed to be produced by a variable-speed control.

f. Diameter-Ratio Lines for the Determination of System Resistance and Power. The diameter-ratio lines for system resistance contained in quadrant (6) state graphically equations (V-6b and -6c) combined and indicate that if everything else remains the same, the corrected static pressure rise is proportional to the square of the impeller diameter. The word "system-resistance" is here used in the same sense as in quadrant (4). The diameter-ratio lines for power state graphically equation (V-8b), i.e., that input horsepower is proportional to the fifth power of the impeller diameter. The diameter-ratio is defined as explained for quadrant (2).

g. Speed-Ratio, Width-Ratio, and Density-Ratio Lines for Determination of Power. By equation (V-8a), the required input power varies with the cube of the rotational speed. The speed-ratio lines contained in quadrant (7) illustrate this variation graphically. The speed-ratio is defined as explained for quadrant (4). The reflection-line shown in quadrant (7) serves merely to project a value obtained on the vertical coordinate axis in quadrant (7) to the horizontal coordinate axis of quadrant (8). Quadrant (8) contains width-ratio and density-ratio lines to correct for these effects on the required power input. These two families of lines graphically illustrate equations (V-8c and -8d), which state that the power is directly proportional to both inlet air density and, for centrifugal blowers only, to the impeller width also. To find the required power input of a blower by use of these quadrants, it is necessary to proceed in the direction indicated by the solid line and arrows.

6. Applications of the Analysis and Design Chart

The working chart in Figure V-12 may be used conveniently to facilitate solution of problems where blower performance in connection with cooling a given equipment must be determined. Such problems may be concerned with evaluation, modification, selection, design, or control. The method of attack for each problem is somewhat different. However, in every case it is necessary that the characteristic curve of the blower to be evaluated, modified, or controlled, or of the reference blower on which the selection or the design is to be based, be plotted in quadrant (5). The resistance characteristics of the equipment to be cooled must be defined generally in quadrant (1) for problems of evaluation or in terms of specific operating points for other types of problems.

In some problems it is convenient or necessary to modify, for best accuracy, the known characteristics of the actual blower or of the reference blower by applying the blower laws, in order that the scales in quadrant (5) be fully utilized or may accommodate the characteristics. In particular, the laws of speed effect, equations (V-6a to -9a), and of width effect, equation (V-6d to -9d), should be used. For that purpose, the width relationships apply to any type blower. Thus the modified blower becomes the reference blower so that in order to analyze the actual blower, speed ratios and/or width ratios to be used must represent the ratios of the actual blower to the fictitious reference blower. Therefore, in problems discussed hereafter in which it is specified that the width ratio must be unity, because it is assumed that the characteristics of the actual blower or a true homologous blower are given in quadrant (5), the width ratio may be other than unity if a fictitious reference blower is used.

a. Evaluation of Available Flow Rates

The chart can be used to determine the flow rate produced through an equipment of known characteristics by a blower operating at predetermined speed. If the speed is a function of the load, a direct determination of the flow rate by means of the chart is not feasible. However, preliminary data towards the solution of this problem can be obtained with the chart. The evaluation of equipment-blower-motor combinations is discussed in detail in the subsequent section, beginning page 105. A method for flow determination by calculation, without use of the chart in Figure V-12, is given there. That method can be used as an alternate to the procedure described in the following.

The performance of a given blower may be known by its own pressure-volume flow characteristic, or by the characteristic of a homologous blower. Also, the numerical values of the characteristic curves may be either at the specified operating speed or at another reference speed. Whichever curves are available for reference, they are plotted in quadrant (5). The pressure curve must be translated to quadrant (1) to determine for each environmental condition the available air flow by intersection with the system characteristic.

In general, irrespective of environmental conditions (as defined by the density ratio), the blower's corresponding points of corrected static pressure ($\Delta p/\sigma_i$) and flow volume can be plotted on the abscissa of quadrants (1) and (3), respectively. The subsequent constructions can be performed originating from these two axes.

There are two methods available to obtain corresponding points of corrected static pressure and flow volume to be plotted on the abscissa of quadrants (1) and (3) respectively. Values to be plotted can be determined directly from the available characteristic performance of the blower when the diameter and speed of the reference blower are those of the actual blower. Also, corresponding values can be ascertained by simple calculations using equations (V-6 and -7) when the speed and diameter of the reference blower are not those of the actual blower. These calculation procedures constituting the first method would usually be more convenient than the second method described in the following, in which quadrants (2), (3), (4), and (6) are used. The second method is presented to indicate the solution to the general problem by using the chart exclusively.

The process using the second method can be explained best with reference to Figure V-13 which shows the necessary construction lines. The complexity of the example is increased in that a homologous blower of greater diameter but lower than specified speed is used as reference blower, rather than the actual blower. Thus, the diameter ratio of the actual blower is 0.9 and its speed ratio is 3. The width ratio is 1.0 since the blowers are homologous. To find for a point on the reference curve of quadrant (1) the corresponding pressure on the abscissa of quadrant (3), the construction shown for point A in Figure V-13 must be used. From A in quadrant (5), a line is projected downward to a_1 on the line of 0.9-diameter ratio for system resistance in quadrant (6), then over to a_2 on the line of 3-speed ratio for system resistance in quadrant (4), and finally up to a_3 representing the desired point on the abscissa of quadrant (3). Similarly, the point representing the actual flow volume corresponding to A is found on the abscissa of quadrant (1). From A a line is projected horizontally to a_4 on the line of 1.0-width ratio for flow volume in quadrant (3), then downward to a_5 on the line of 3-speed ratio for flow volume in quadrant (4), then over to a_6 on the line of 0.9-diameter ratio for flow volume in quadrant (2), and finally up to a_7 representing the desired point on the abscissa of quadrant (1). By the same methods are found points b_3 and b_7, c_3 and c_7, and d_3 and d_7, corresponding to points B, C and D, respectively.

The final steps to translate the points of the reference blower characteristic to a curve in quadrant (1) of actual pressure versus flow volume, based on blower inlet conditions, are performed by using for each point the appropriate density ratio line in quadrant (3). Under identical environmental conditions, the procedure for forced and induced flow through the equipment differ since the inlet conditions to the blower would not be the same. The differences are apparent with further reference to Figure V-13. The constructions are based on an example for which the environment where the cooling air originates is at 16°C and at a pressure of 3 inches mercury. The equipment flow characteristics are represented by the plot of standard pressure drop versus weight flow (points E, F, G, H) in quadrant (1). The heat dissipation of the equipment is such that for a weight flow of 0.1 pound per second the temperature rise of the cooling air is 50°C.

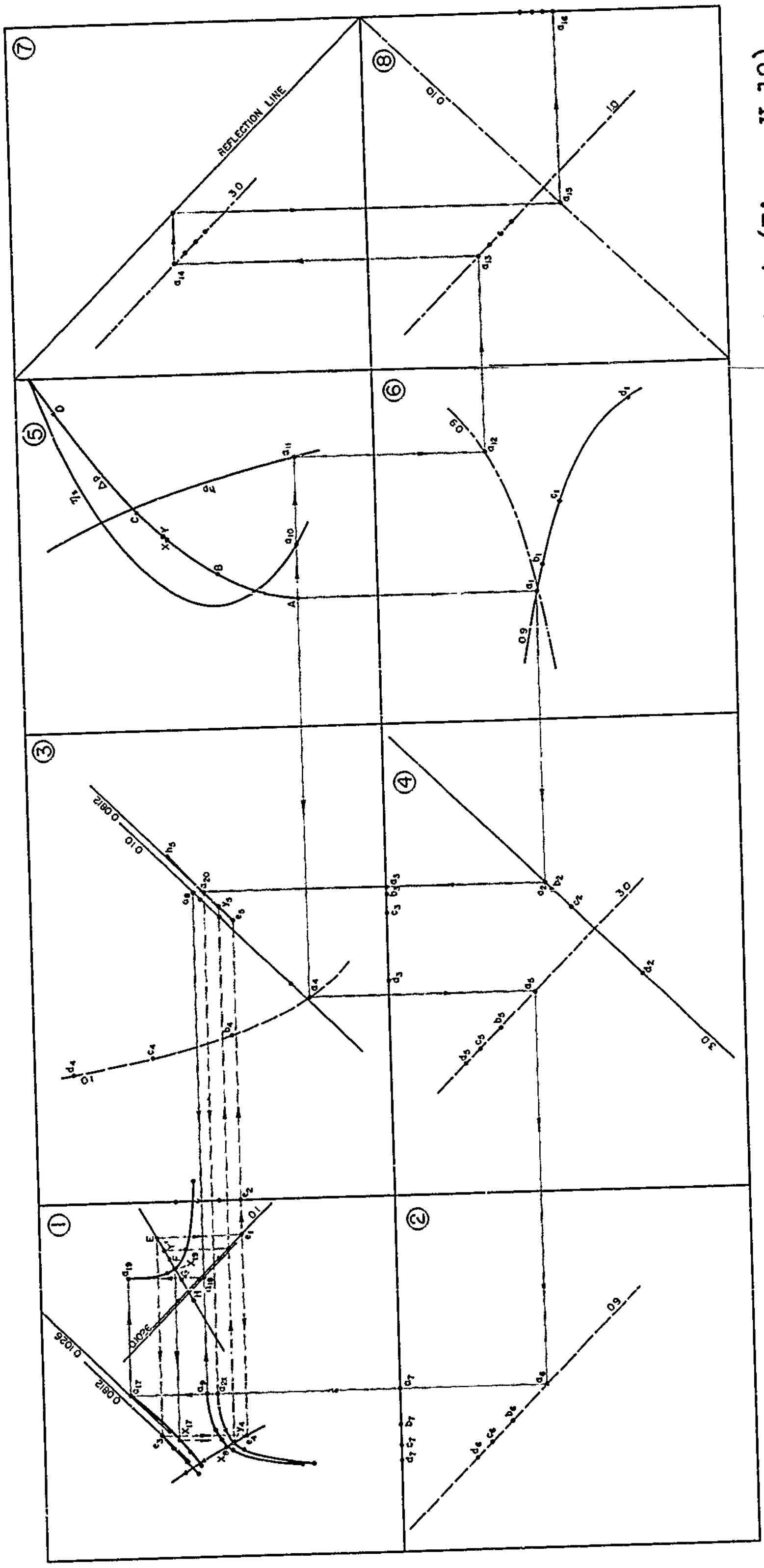

Figure V-13. Evaluation of Available Forced and Induced Flow Rates by Use of Chart (Figure V-12)

With forced flow, i.e., the air pressure at the inlet to the equipment is greater than atmospheric, the blower receives air from the environment. Thus, the density ratio at the blower inlet corresponds to 16°C and 3 inches mercury and is 0.1 (Figure V-8). Using this density-ratio line in quadrant (3), the blower characteristic is constructed in quadrant (1) by point a_9, and others obtained in the same manner, corresponding to B, C and D. For example, a_9 is obtained by projecting upward from a_3 to a_8 on the line of 0.1-density ratio in quadrant (3), then over to quadrant (1) and intersecting with a line projected upward from a_7 on the abscissa of quadrant (1).

In the solution of the forced-flow evaluation problem, the blower characteristic in quadrant (1) must be further translated into a plot of standard pressure rise versus weight flow, based on the discharge conditions of the blower, in order to establish an operating point with known equipment characteristics. To perform this conversion, the density ratio lines in quadrant (1) are used. For conditions of low altitude and for blowers supplying little pressure the density ratio at blower discharge is not different from that at inlet. However, with high-speed machines at high altitude, some difference may result. Within the limitations of incompressible flow relationships, as pointed out on page 87, for which alone the chart and the chosen interpretation of equipment characteristics are applicable, the differences are normally not great because pressure and temperature rise across the blower tend to compensate mutually. In the following, the complete procedure, taking into account the compression effect of the blower, is described.

For example, to find weight flow and corrected pressure corresponding to point A, the temperature rise across the blower can be calculated directly from equation (V-3), by using the value of the static efficiency read for a_{10} in quadrant (1) and the value of the corrected pressure rise read at a_3. If instead of the static efficiency curve the horsepower curve is given, equation (V-2) must be used. To determine the actual blower's brake horsepower corresponding to point A, the following procedure is used. Point a_{11} in quadrant (5), representing the horsepower of the reference blower corresponding to A, is projected downward to a_{12} on the line of 0.9-diameter ratio for power in quadrant (6), then over to a_{13} on the line of 1.0-width ratio (appropriate for a homologous blower), then upward a_{14} on the line of 3-speed ratio, then horizontally to the reflection line, then downward to a_{15} on the line of 0.1-density ratio, and finally over to a_{16} indicating the horsepower required. That value, the density ratio of 0.1, and the flow volume determined by a_7 are used to find Δt_B by equation (V-2). For point A the temperature rise so calculated is 21°C, indicating a blower discharge temperature of 37°C. At a_3 the value of $\Delta p/\sigma_i$ is found, and from it and $\sigma_i = 0.1$ the static discharge pressure, here 4.1 inches water. This pressure converted to inches mercury and added to the environmental pressure gives the absolute inlet pressure to the equipment, here 3.3 inches mercury. Using the equation in Figure V-8, the density ratio at the blower discharge is calculated as $\sigma_d = 0.1026$. This value is used to find the point corresponding to A on the blower's discharge characteristic of standard static pressure ($\sigma_1 \Delta p$) versus weight flow, since the blower's σ_d is the equipment's inlet density ratio σ_1. From a_9 vertical and horizontal projection lines determine a_{17} and a_{18} on 0.1026-density ratio lines for flow volume and system resistance, respectively. The intersection of horizontal and vertical projection lines from a_{17} and a_{18}, respectively, determines the desired characteristic point a_{19}. In similar manner b_{19}, c_{19} and d_{19} are found. The curve drawn through them intersects the system characteristic at x_{19} which represents the operating point of the blower. This point can be projected back to X in quadrant (5) by establishing first its location on the curves through points with subscripts (17) and (9), in quadrant (1), as shown in Figure V-13, and then the lines on which points with subscripts (6), (5) and (4) are located. For the weight flow rate given by x_{19} (0.14 pound per second) and the other values of flow volume, pressure rise and horsepower (0.92 hp), determined as for A, the temperature rise across the blower at the operating point (11.8°C) can also be found.

The preceding procedure may be compared with that for induced flow which is also indicated in Figure V-13, using dashed projection lines. The curve for actual flow volume versus static pressure, at the equipment exhaust, which is equivalent to the blower inlet, must first be determined. The static pressure is negative in reference to the environment. For example, for point E in quadrant (1), the actual system resistance is found by projecting downward to the line of 0.1-density ratio for system resistance, corresponding to σ_1 (conditions at equipment inlet), and over to e_2. Thus the absolute pressure at the equipment outlet is established by deducting the value at e_2, converted to inches mercury, from the environmental pressure (3 inches mercury). The air temperature at the equipment outlet would be determined by the equipment's heat dissipation characteristics; for example, here stated to be 50°C above inlet for 0.1 pound per second flow which corresponds to E. Thus the density ratio at the equipment discharge σ_2 which equals σ_i at the blower inlet can be determined. For the figures given previously, it is found to be 0.0812 for E. Thus, to establish the characteristic of discharge flow volume versus pressure drop of the equipment, the point corresponding to E is found by projecting horizontally over to e_3 on the line of 0.0812-density ratio for flow volume, and then downward to intersect the horizontal projection line passing through e_1 and e_2 at e_4. Points f_4, g_4 and h_4 to establish the curve, are found in the same manner. The temperature rise across the equipment is taken as being inversely proportional to the weight flow, assuming no other means of heat dissipation or heat gain. For example, since the weight flow for H is 0.2 pound per second, the temperature rise would be only 25°C.

The next problem in the induced-flow evaluation is to bring the discharge characteristic of the equipment to intersection with the blower characteristic, the latter being based on inlet conditions determined by temperature rise and pressure drop of the air passing through the equipment at equal weight flow rates. To establish this blower characteristic in quadrant (1), points e_5 through h_5 are found in quadrant (3) on density ratio lines corresponding to equipment discharge conditions, e.g., e_5 on 0.0812-line. These points determine a curve on which points a_{20} through d_{20} may be located by projecting upward from a_3 through d_3. Then projecting over to quadrant (1) and intersecting with the vertical lines from a_7 through d_7 the blower characteristic through points with subscripts (21) is established. The operating point is found at y_4. The corresponding weight flow (0.12 pound per second) is found by projecting to 0.1-density ratio line for system resistance, finding x_1, and up to Y on the equipment characteristic. The point may also be projected by way of quadrants (2), (4) and (3) to Y on the reference characteristic. The power requirement can be established in the previously described manner, using the density ratio (0.08) found by interpolation and based on y_5 in quadrant (3). The value found is lower than for forced flow because of the lower air density at the blower inlet, although the flow volume is approximately the same.

b. Determination of Required Blower Speed

The problem of finding the required operating speed of a blower applied to an equipment can be solved by trial-and-error calculations which may be replaced by graphical procedures using the chart. The problem may arise in various instances when the flow volume and static pressure needed to operate an electronic unit under given environmental conditions are specified. For example, it may arise in the selection of a drive motor for a blower made available in the development of an equipment, or in the modification of an equipment for operation under changed environmental conditions which may be accomplished by substitution of the drive motor by one of different speed and power. The problem may also arise in determining the necessary characteristics of a speed control device to be developed for a blower-motor unit used on a specific equipment.

The specification of the required flow volume and static pressure of the blower depends on the equipment's flow and heat transfer characteristics and the arrangement used. In forced-flow applications, as discussed on page 86, the density ratio at the equipment inlet depends also on the temperature rise across the blower which would be a function of the static efficiency at the operating point (defined in terms of the percentage of the blower's maximum flow capacity) yet to be found. Therefore, an estimate of the efficiency would be required to establish the flow volume and pressure required. After determination of the blower's operating point, the estimate can be verified. However, in practice, as pointed out on page 98, the density ratio could not change appreciably between the blower inlet and outlet. Therefore, the speed evaluation can be performed with little error based on the density ratio σ_i at the blower inlet which would correspond to the environmental σ_o.

In induced-flow applications the required flow and pressure can be established directly from the equipment characteristics, thus defining the conditions at the blower inlet. These data, like in the forced-flow evaluation, establish the actual flow volume and actual system resistance at the blower inlet plotted in quadrant (1) of the chart as a point such as S (500 cubic feet per minute and 4 inches water) shown in the schematic of Figure V-14.

The pressure and power characteristics of the blower to be evaluated for speed requirement would be plotted in quadrant (5) at some reference speed for which they are known. The curves for a homologous blower could also be used, although this is not likely to be the case. The graphical procedure in the chart consists in the construction of two sets of lines, starting at point S in horizontal and vertical directions, respec-

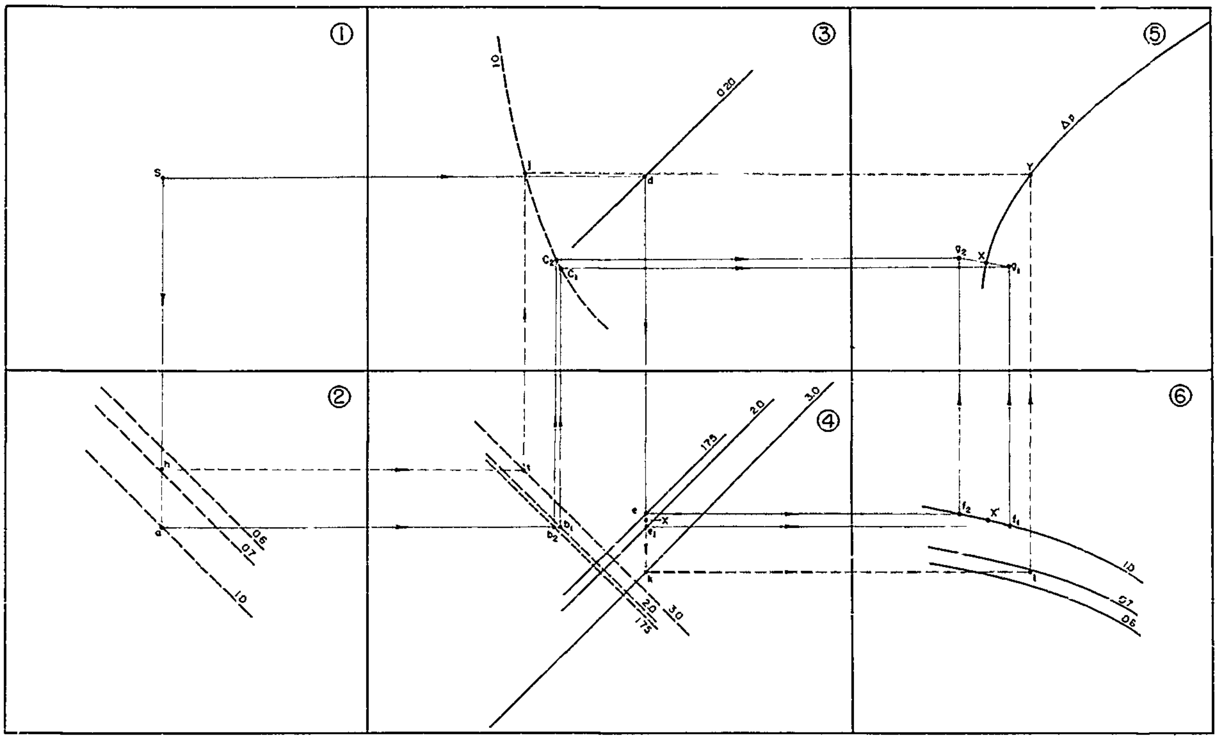

Figure V-14. Determination of Required Blower Speed and Selection of Blower Size by Use of Chart

tively, and finally intersecting in quadrant (5) at a point which, for the solution of the problem, must lie on the blower's pressure characteristic.

If the characteristics of the blower being evaluated are plotted in quadrant (5), the construction lines shown in Figure V-14 are used. Point S is projected downward to (a) on the line of 1.0-diameter ratio, and then over to b_1 in quadrant (4) located on the line of 2.0-speed ratio for flow volume. This speed ratio is chosen arbitrarily as first estimate of the probable operating speed, here assumed to be twice that on which the characteristics in quadrant (5) are based. From b_1 the line is drawn up to c_1 on the line of 1.0-width ratio, and from there over to quadrant (5). The other set of lines is started by projecting S over to (d) on the line of 0.2-density ratio (the given blower inlet condition), down to c_1 on the 2.0-speed ratio line for system resistance (in accordance with the first assumption of probable operating speed), then over to f_1 on the line of 1.0-diameter ratio for system resistance, and finally up to quadrant (5) to intersect the line drawn from quadrant (3) at point g_1. This point is seen to be not on the pressure curve and, therefore, another speed must be assumed. Since the pressure at g_1 is lower than that on the blower characteristic at equal flow volume, the assumed speed ratio of 2.0 is too great. Assuming a value of

1.8, the lines through e_2, f_2 and g_2, respectively through b_2, c_2 and g_2 are drawn. The operating point X is found at the intersection of line g_1-g_2 with the blower characteristic. The speed ratio for X is found by projecting back to X' in quadrant (6) and over to X" on the vertical line between e_1 and e_2 in quadrant (4). By interpolation, the speed-ratio line passing through X" is found (1.9). The power required to drive the blower can be found in the manner described on page 98, using the newly found speed ratio (1.9). Figure V-14 does not contain the construction.

The speed established by the above-described method may not be exactly obtainable using one of a series of standard drive units. In that case, the nearest available greater speed must be used. The increased flow, necessitating also greater power, may be utilized. The operating point can be established as described previously for evaluation of available flow rates. Otherwise, it may be desired to dissipate the excess pressure by means of throttling, as discussed subsequently on page 118. For example, if for the problem illustrated in Figure V-14, the nearest available drive speed would correspond to a speed ratio of 2.0, the operating point would be at Y, found by projecting g_1 horizontally onto the blower characteristic. The corresponding power would also have to be determined for a speed ratio of 2.0.

c. Selection of Blower Size

The problem of selecting a blower from among a group of geometrically similar, homologous blowers to be operated at a specified speed may present itself in some phases of equipment modification and development. It is solved like the problem of speed determination for a given blower, as described in the preceding, except that instead of assumptions for speed ratio, assumptions for diameter ratio are made. The dashed lines in Figure V-14 are for the correctly determined diameter ratio (1.75) to be used if a speed ratio of 3.0 is specified with flow conditions expressed by point S and the blower characteristics in quadrant (5) are those of the reference blower. The operating point is shown at Y.

The diameter so determined may actually not be available among the series of homologous blowers. Then a larger blower may have to be used and the same considerations would apply as discussed above in reference to the use of a speed greater than that determined to be necessary for a given blower.

This method would sometimes be undesirable because the operating point resulting from the selection of a blower diameter for a specified speed may fall on a part of the blower characteristic near maximum flow volume for which the efficiency is low. To improve the economy of the application, a lower operating speed can be chosen for which the dimensions of the blower would have to be larger to deliver the required air flow. Its operating point would move towards a lower percentage of maximum flow volume, with corresponding higher static efficiency.

d. Design of Blower

If a blower is to be designed to deliver a specified flow volume and pressure under certain environmental conditions, and its operation is intended to be continuous, its efficiency should be high and its flow be stable. Therefore, with the design based on the characteristic of a reference blower the operating point is specified, as shown by point P in quadrant (5) of Figure V-15. Since for the design condition the flow volume, system resist-

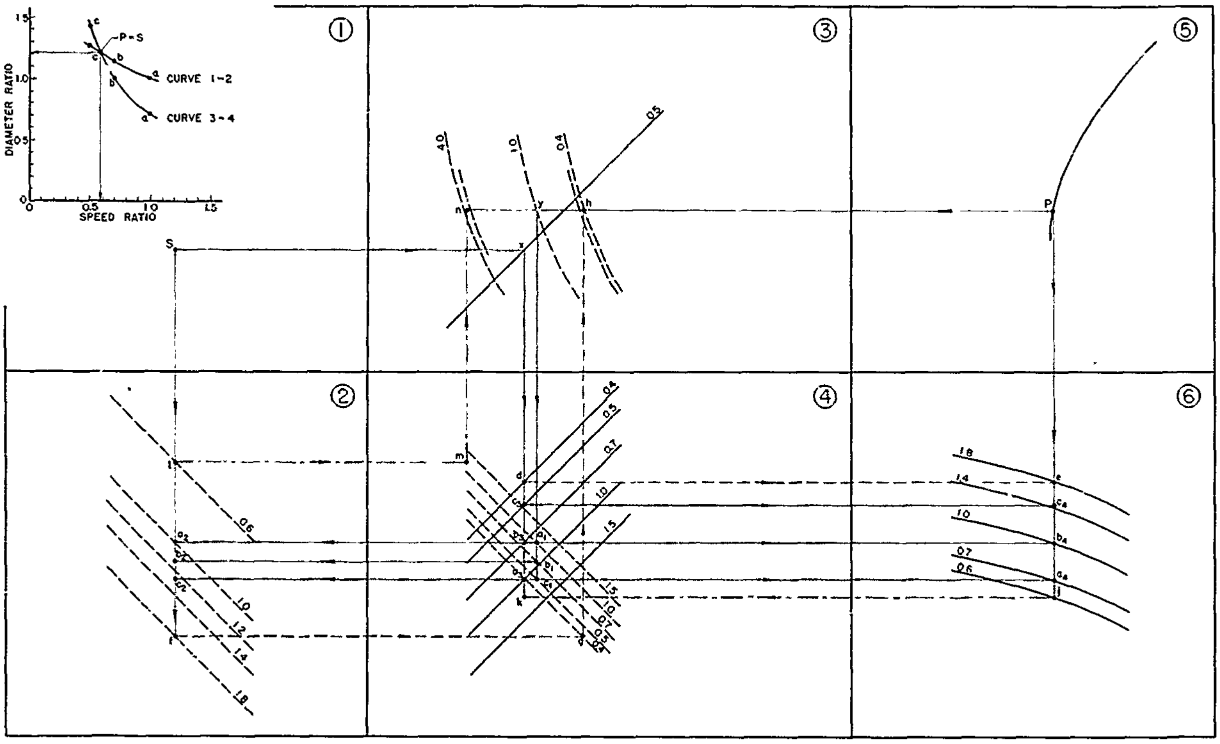

Figure V-15. Design of Blower by Use of Chart (Figure V-12)

ance (point S, Figure V-15) and density ratio at the blower inlet are also specified, the blower's speed, diameter, and, for centrifugal blowers, the width may enter the design considerations. Among the three characteristics, only one can be fixed, two must remain to be determined to meet the requirements of points S and P. If the design is to be limited to homologous blowers, or if the blower is axial-flow, the width ratio is unity. Then the objective of the design procedure becomes to find the proper combination of speed- and diameter-ratio so that proper projection lines on the chart would connect points S and P. Thus, the blower would create the pressure and would deliver the quantity of air required by the equipment. The solid projection lines in Figure V-15 illustrate the procedure for the example chosen, having defined the system's operating point S in terms of actual flow volume and resistance (400 cubic feet per minute and 1 inch water), the density ratio at the blower inlet (0.5), and the desired operating point P in terms of its location on the pressure curve, chosen for high efficiency and stability of operation (400 cubic feet per minute and 4 inches water).

Point S is projected to x on the line of the density ratio at blower inlet (0.5) in quadrant (3). Point P is projected to z on the line of 1.0-width ratio in quadrant (3), since the design is concerned only with homologous blowers. Then, the vertical projection lines from x and y must terminate in quadrant (4) on a speed ratio line for system resistance, respectively for flow volume, both having the same numerical value. The horizontal projection lines drawn from the points so established must intersect the vertical projection lines drawn from points P and S, respectively, at points located on a diameter ratio line for system resistance in quadrant (6), respectively for flow volume in quadrant (2), which both have again equal numerical value. Obviously, this could be accomplished by trial-and-error projections to assumed speed ratio lines in quadrant (4) and comparing the diameter-ratios determined by points of intersection located in quadrants (2) and (6). This procedure may be made more precise by finding on at least three sets of speed-ratio lines in quadrant (4), shown in Figure V-15 as 1.0, 0.7 and 0.5, the points corresponding to x, i.e., a_3, b_3 and c_3 in Figure V-15, and determining by projecting each to quadrant (6) the corresponding diameter ratios specified by a_4, b_4 and c_4. In an auxiliary plot of diameter ratio versus speed ratio, shown in the upper left of Figure V-15, the data so found are entered as curve 3-4. Similarly, corresponding to y points a_1, b_1, and c_1 are located and their mating points a_2, b_2, and c_2 with corresponding diameter ratios are located by projecting to quadrant (2). These data are also entered in the auxiliary plot to establish curve 1-2. The intersection of the two curves in the auxiliary plot, point P-S, gives the values of the diameter and speed ratio (1.22 and 0.58) which, if used in quadrants (2), (4), and (6), would connect points P and S with a continuous system of lines. To determine the proportions of the blower so designed, all the dimensions of the reference blower must be multiplied with the diameter ratio (1.22). To determine the speed of the designed blower, the speed of the reference blower must be multiplied with the determined speed ratio (0.58).

If the design is limited to a specific operating speed, the blower laws expressed by equations (V-6d to -9d), but only applicable to centrifugal blowers, must be used to determine a modified impeller width which is not changed in the same proportion as the diameter. Since the pressure production of the blower is assumed by the laws to be independent of impeller width and only dependent on operating speed and impeller diameter, the projection lines for system resistance can be drawn directly to define the impeller diameter. If for the design shown in Figure V-15 a speed is specified, the line for system resistance in quadrant (4), having a value equal to the ratio of the specified speed to the reference speed (0.4), locates a point d on the projection line from x. The projection of point d over to quadrant (6) intersects the vertical projection line from P in point e whose location relative to diameter ratio lines for system resistance, having known values, determines uniquely the diameter ratio of the impeller so designed (1.8). This value may then be used to determine the diameter ratio line for flow volume in quadrant (2) on which point f is located by vertical projection from point S. The following construction is from f over to g in quadrant (4) on the speed ratio line for volume flow of specified value (0.4), and up to the intersection in quadrant (3) with the horizontal projection line from P, locating point h which by its position relative to lines representing known width ratios determines the width ratio of the designed blower (0.41). The actual width of the impeller is the product of the reference blower width and the width and diameter ratios determined in the described construction (B x 0.41 x 1.8).

If the design is limited to a specific diameter, the same considerations as discussed in the preceding paragraph in reference to speed specification applies. In this case, the speed and width ratios must be found. Again the pressure requirement governs and determines the speed ratio by the projections from P to j in quadrant (6) on the line for specified diameter ratio (0.6), and then to k in quadrant (4), located by the intersection with the vertical projection from x. By interpolation the speed ratio is found (1.75), and the same value is used to establish the speed ratio line for flow volume, in quadrant (4), on which m is found by projection from l, the latter being located on the line for specified diameter ratio (0.6) in quadrant (2). The intersection of the projection lines from P and m at n in quadrant (3) permits determination of the width ratio by interpolation (3.8).

Combined Equipment-Blower-Motor Performance

An important phase in the evaluation of blower applications with electronic equipment is the determination of air flow rates over the range of environmental conditions in which the equipment is expected to operate. As described on pages 95 to 100, the analysis may be performed directly by use of the chart in Figure V-12 if a drive is used which has at every operating condition a known speed, obtained by control, or otherwise. However, if only the speed-torque characteristic of the drive is known, in addition to the equipment's flow characteristics, the analysis is more complex and requires supplementary methods.

It is generally known that constant-speed blowers are not satisfactory for equipment cooling applications over wide ranges of altitude. If they provide adequate cooling at high altitude, they usually overcool the equipment greatly at low altitude, unless the environmental temperature is very much increased. Their power requirement also increases with lower altitude and necessitates a larger drive motor. To satisfy requirements for best cooling over wide ranges of environmental conditions, the torque of the drive motor should increase about in direct proportion with the speed. All types of practical electric motors have lower torque at increased speed. However, various types of motors have natural operating speeds which may cover a range from one to three or five. Thus, when used for blower drive over an appreciable range of altitude, they provide some increase in air flow volume with increased altitude. Although not entirely adequate, such motors have been and will be used where required blower size and the corresponding power are both small so that the expense of overcooling is not prohibitive. They may also fit the situation when sustained operation of the equipment would occur only over a limited range of ambient pressure and temperature, and when variations of component temperatures are permissible.

1. Flow Rate in Specified Environment

The evaluation of available flow rates resulting from the combination of an electronic equipment, a cooling blower, and a blower drive-motor may be performed experimentally. For that purpose, the use of a test facility providing for control of environmental temperature and pressure would be necessary. Some relevant procedures are discussed in the portions of Chapter IV dealing with altitude chamber tests.

The necessity for extensive experimentation is eliminated if the following information is available: (1) the flow and resistance characteristic of the equipment, giving graphically or analytically the relation between standard pressure drop $\sigma_1 \Delta p$ and weight flow of air, (2) the variation of pressure and required driving power with flow volume of the blower at any speed and air density, permitting derivation of the blower's torque-flow volume curve, and (3) the speed-torque curve of the drive motor.

At any environmental condition, analysis of the blower's air supply to the equipment at various speeds would define the relationships of available flow volume and corresponding driving torque with blower speed. The operating point, using a specific drive motor, would be determined at a speed where the motor torque and the blower torque would be equal. Thus the flow volume, or weight flow, would also be defined.

The type of problem outlined above is not amenable to mathematical analysis. The solution could be obtained by trial-and-error calculations which, however, are likely to be very time-consuming, unless a good first estimate is available based on a set of operating conditions for which flow rate, blower speed, torque, etc. are already known. To evaluate such combined operation of three available components it is advantageous to employ calculations combined with graphical methods. In general, when the equipment is to be operated at various environmental conditions, it is best to evaluate first the available air flow at the condition of lowest air density. At this point, which usually occurs at maximum altitude, the weight flow of air would be smallest and most critical. The evaluation procedures differ somewhat, depending on whether a forced- or an induced-flow arrangement is used.

It will be noted in the following that a substantial portion of the procedures is concerned with determining air flow rates through the equipment at different assumed speeds. For that purpose the procedures described beginning on page 95, utilizing the chart in Figure V-12, may be used. However, the following methods are presented using an alternate method for determination of air flow rate at specified speed.

a. Forced Flow Arrangement

The environmental condition, given in terms of air temperature and pressure, is defined by its density ratio (Figure V-8). This is the density ratio at the blower inlet. That at the equipment inlet would be slightly different because of pressure and temperature change across the blower. However, as pointed out on page 98, these are compensating effects which usually cause small change of density ratio. Therefore, in the procedure to be presented, the density ratio at the equipment inlet is assumed to be equal to that at the blower inlet. Thus corresponding values of actual flow resistance and volume are derived from an equation or plot of $\sigma_1 \Delta p$ versus W for the equipment. For example, Figure V-16 contains the curve of Δp versus Q, derived for an equipment having the experimentally determined characteristic flow equation $\sigma_1 \Delta p = 830\ W^{2.5}$, operating at 60,000 feet N.A.C.A. standard atmosphere ($\sigma_o = 0.0942$, Table A-I-2). For any value of weight flow W, the flow volume Q is found in Figure V-9 or by equation (V-1) and $\Delta p = (\sigma_1 \Delta p)/0.0942$.

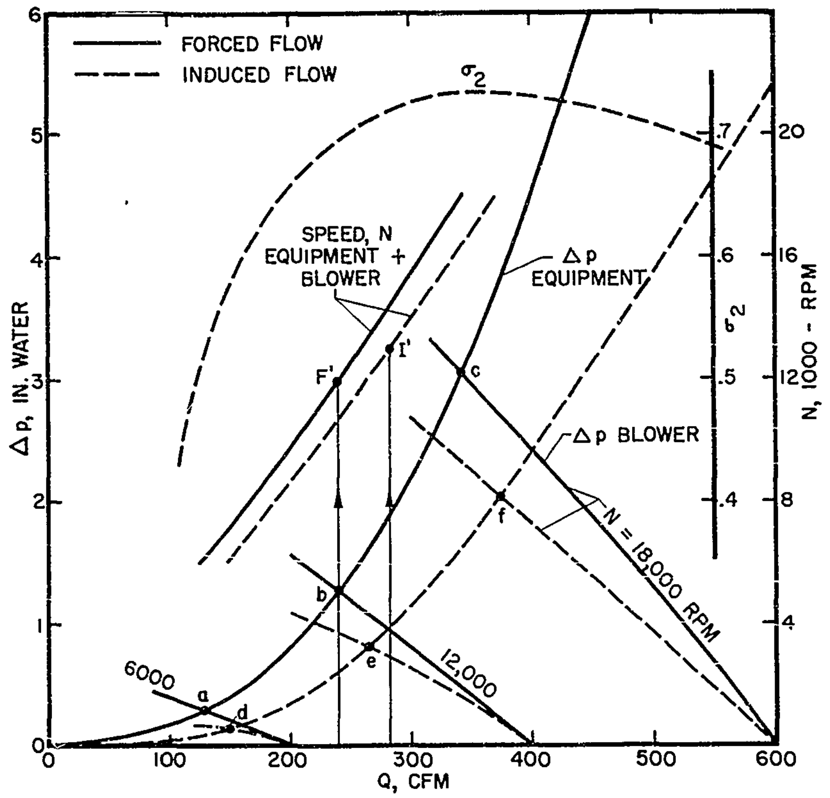

Figure V-16. Determination of Combined Equipment-Blower Characteristics for Forced and Induced Flow

The blower characteristics of pressure and power versus flow volume are available at some reference speed N_r which need not necessarily be within the practical speed range of the motor. For example, Figure V-17 contains the data for a blower operating at 6000 revolutions per minute at standard density ($\sigma_i = 1.0$). By means of the blower laws, the pressure characteristic can be converted to the values corresponding to the specified inlet air density and to one of three or four speeds within the operating range of the motor. Some knowledge of the probable operating speed is desirable so that the chosen speeds are not too greatly different. By equation (V-6),

$$\Delta p = \sigma_i\ (N/N_r)^2\ (\Delta p/\sigma_i)$$

and by equation (V-7)

$$Q = (N/N_r)\, Q_r$$

Three or four points for each speed will suffice to enable one to plot the necessary portion of the curve, as shown in Figure V-16 for three different speeds (6,000, 12,000 and 18,000 rpm). Corresponding points a, b and c are found at the intersections with the equipment characteristic which determine the operating point for each speed. At the operating point the pressure and flow volume are so related that they are equal to the requirements of the equipment. Thus three points are established on the curve of operating speed versus flow volume for the equipment-blower combination (see Figure V-16). For each point, the necessary blower torque is found by means of the reference characteristics where values of corrected horsepower hp/σ_i are found corresponding to the reference flow volumes Q_r obtained by reversing equation (V-7), as given above. The equation for the blower torque in pound-feet is

$$T_q = \left[33{,}000/(2\pi N_r)\right] (\sigma_i)\ (N/N_r)^2\ (hp/\sigma_i) \qquad \text{(V-10)}$$

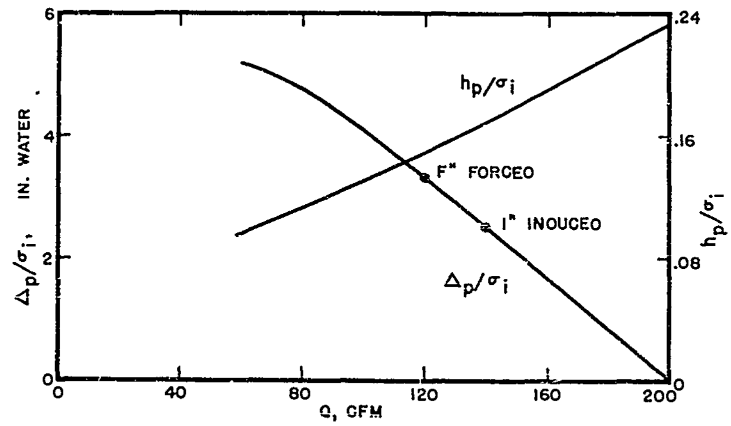

Figure V-17. Blower Pressure and Power Characteristics

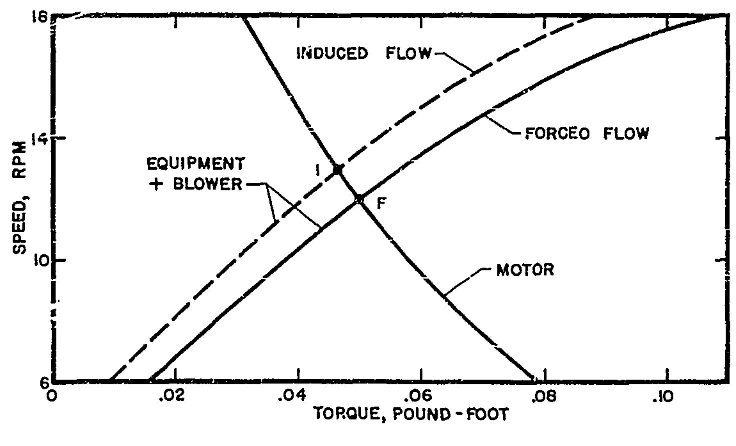

Figure V-18. Determination of Blower Operating Speed for Equipment-Blower-Motor Combination by Means of Speed-Torque Curves

The blower torque calculated for the operating point determined at each speed may then be plotted against the speed on the same grid as the speed-torque curve of the motor, as shown in Figure V-18. The intersection of the two curves at F indicates equality of the motor torque with the requirement of the blower and determines the operating speed (12,000 rpm). The corresponding flow volume is established by the speed versus flow volume curve in Figure V-16 (point F', 240 cfm) and the weight flow by Figure V-9 or equation (V-1). Also, the operating point on the blower characteristic F" in Figure V-17 (120 cfm) is found by reversing the forms of equations (V-6 and -7) given above.

If speed-torque curves of the motor are available for different air densities, they should be used instead of a single curve, as shown in Figure V-18. Then the corresponding operating point F would be obtained on the speed-torque curve for the specified environmental conditions.

b. Induced Flow Arrangement

The density ratio of the air at the equipment inlet corresponds to environmental conditions. Due to temperature rise and pressure drop across the equipment, the density ratio of the cooling air may change appreciably. Therefore, this effect must be taken into account in determining the flow volume and density ratio at the blower inlet. Thus, in order to match equipment and blower characteristics, the plot of $\sigma_1 \Delta p$ versus W for the equipment must be converted into a plot of Δp versus Q, where the pressure drop is that across the equipment at the specified inlet condition and the flow volume Q is that at the outlet of the equipment. In order to permit evaluation of Q the relationship of temperature rise with flow rate must be known. It is a function of the heat dissipation within the equipment. If no other heat dissipation occurs or if that other dissipation is constant, the temperature rise must be inversely proportional to the weight flow rate of cooling air. In order to derive the plot of Δp versus Q, the value of Δp can be found directly for any weight flow W, as indicated for forced flow. However, to find the discharge flow volume corresponding to a given weight flow, the density ratio at the discharge must be calculated based on the absolute pressure and the absolute temperature at that point. The absolute pressure and temperature are affected by the pressure drop, respectively the temperature rise, across the equipment. The dashed equipment curve in Figure V-16 is derived for the same equipment as for forced flow having such heat dissipation that $\Delta t = 1.5/W$. At any pressure drop the difference between the curves for forced and induced flow show the expansion of the cooling air passing through the equipment.

The values of density ratio σ_2 at equipment outlet are also plotted versus the discharge volume. The variation shown in Figure V-16 is typical and shows the curve to rise from a low value at low flow rate (due to large temperature rise), to a maximum, and then to drop at very large flow rates (due to large pressure drop).

The blower characteristics at three or four speeds can be transferred to the plot of Δp versus Q of the equipment like for the forced flow arrangement, except for one important difference. For each value of flow volume obtained by equation (V-7), the corresponding pressure rise is determined by equation (V-6). However, the density ratio used must be that corresponding to the same flow volume on the curve of Δp versus Q of the equipment. In this manner the three dashed blower curves for different speeds (6,000, 12,000 and 18,000 rpm) shown in Figure V-16 are obtained from the blower characteristics in Figure V-17. After obtaining the operating points (d, e, and f) by intersections, the procedure is the same as for forced flow in finding the blower speed. In calculating the blower torques, the density ratio to be used for each point is that determined by the value on the curve of σ_2 versus Q in Figure V-16 corresponding to the actual flow volume. Similarly, when the operating point determined in Figure V-18 (I at 13,000 rpm) is to be located on the reference blower characteristic (point I", 131 cfm, Figure V-17) and when the weight flow is to be calculated (σ_2 = 0.072, corresponding to I' at 13,000 rpm and 283 cfm on the speed curve in Figure V-16) the density ratio at the equipment outlet must be used.

2. Variation of Flow Rate and Blower Speed with Air Density

The evaluation of available flow rates for a given equipment-blower-motor combination at several conditions of air density may be performed for each condition by a repetition of the procedure described in the preceding outline. However, after the first determination has been made, preferably at minimum operational air density, it is convenient to use a procedure which will yield relative, but exact values. The equipment blower and motor characteristics are expressed on a percentage basis relative to the values corresponding to reference operating conditions, best taken at minimum air density. Since this requires some preparatory work, the method outlined in the following is desirable mainly when the evaluation of flow rate, blower speed, etc. is desired at possibly more than three environmental conditions.

Use is made of the chart contained in Figure V-19 (in back pocket) for finding the blower speed by matching blower and motor characteristics at a known value of equipment resistance. The speed thus obtained can be checked for its compatibility with the weight flow requirement of the equipment which would correspond to flow resistance at which the blower and motor characteristics have been matched.

a. Chart for Performance Analysis of Blower-Motor Units

The chart in Figure V-19 substitutes a graphical procedure for the evaluation of density and speed effects on blower torque and provides a simple means of correlation with motor characteristics. A schematic diagram of the chart is given in Figure V-20 to clarify the following discussion of the features and the use of the chart.

The chart in Figure V-19 consists of four quadrants, two of which provide grids for plotting the characteristic curves of the blower and the motor, respectively. The lower grid in quadrant (1) should contain a plot of the corrected blower characteristics for constant rotational speed, with the corrected torque and the corrected pressure rise of the blower, as percentages of the actual values which are required at the minimum air den-

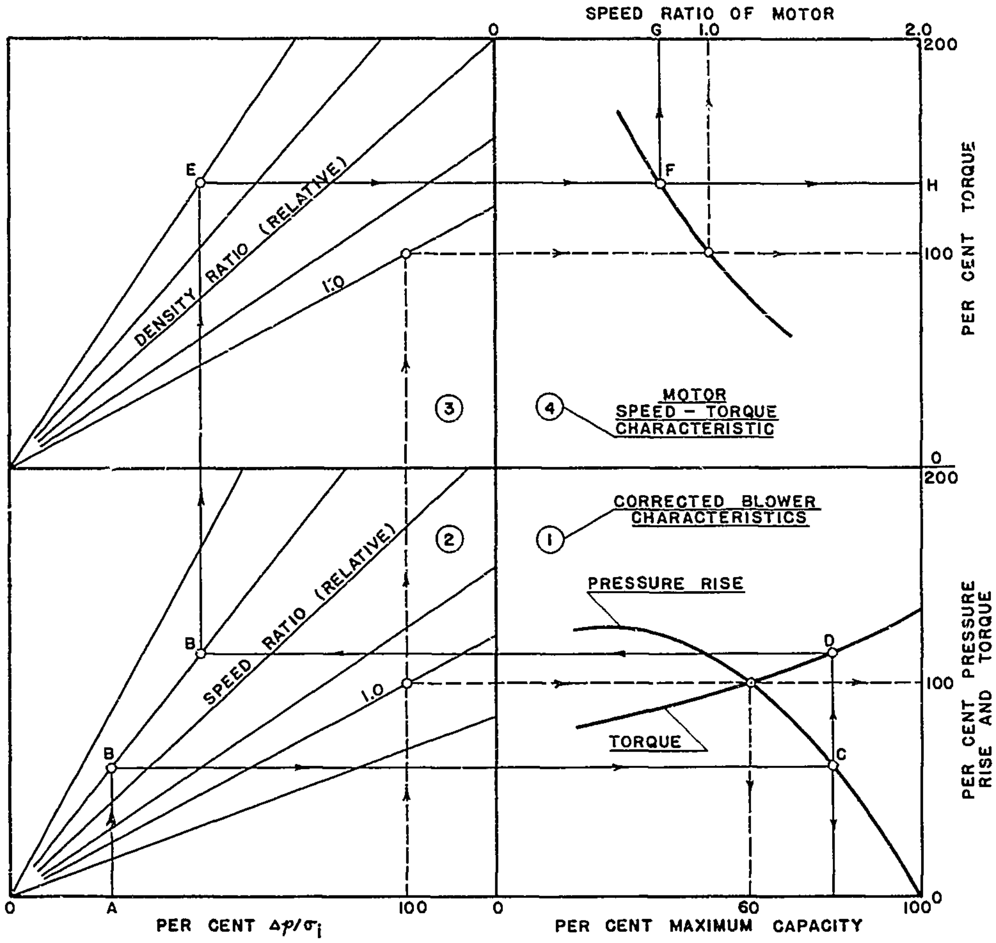

Figure V-20. Speed Determination of Blower-Motor Unit
by Use of Chart (Figure V-19)

sity of the operating range. These percentage values are plotted against per cent maximum capacity of the blower as the abscissa. At constant speed, the torque curve of the blower is the same as its horsepower curve since by equation (V-10) torque and corrected power are proportional. The upper blank grid in quadrant (4) should contain a plot of the motor speed-torque characteristics with per cent of reference torque (minimum air density) as ordinate, and the ratio of operating speed to reference speed (minimum air density) as abscissa. In Figure V-20, which is a schematic diagram of Figure V-19, typical characteristic curves for the blower and motor are shown in quadrants (1) and (4). The reference point corresponding to 100 per cent pressure rise and torque, (usually at minimum air density) is shown at 60 per cent maximum capacity of the blower. This is a typical value for a good combination, but may differ, depending on the application.

Quadrant (2) contains the parameter relative speed-ratio which is defined as the ratio of the actual speed under the conditions of operation being analyzed to the reference speed. One purpose of the speed-ratio lines in quadrant (2) is to relate the required corrected pressure of the blower $\Delta p/\sigma_i$ to the per cent design pressure rise of the reference blower having the characteristic contained in quadrant (1). Quadrant (2) is similarly used when correcting per cent design torque of the reference blower to the actual speed ratio of operation. Quadrant (3) contains the relative density ratio at the blower intake as a parameter and relates the required corrected torque of the blower to the torque made available by the motor. The relative density ratio as contained in quadrant (3) is defined as the ratio of the actual density of the air at the blower intake under the conditions of operation being analyzed to the density at the blower intake under reference conditions.

The chart in Figure V-19 is set up on a percentage and ratio basis in order to make it generally applicable to any design, independent of operating characteristics of the motor and blower, design altitude, system resistance or range of altitude over which the combined performance is to be determined. The reference point of operation may be any selected condition for which the combined equipment-blower-motor performance is known. As indicated repeatedly above, operation at minimum air density may conveniently be taken as the reference condition. By definition, for the combined system at the reference point, the corrected equipment resistance, the corrected torque and pressure rise of the blower, and the torque of the motor are 100 per cent, with the speed ratio of motor and blower both being unity. This condition is illustrated by the series of dashed lines shown on the schematic diagram in Figure V-20. As previously mentioned, the per cent maximum capacity of the blower at the reference point happens to be shown at 60 per cent in Figure V-20.

The series of solid lines shown in Figure V-20 illustrates the procedure for determining the performance of the equipment-blower-motor combination at other than reference conditions. Suppose at some reduced altitude the corrected equipment resistance $\Delta p/\sigma_i$ referred to blower inlet conditions is A per cent of the reference $(\Delta p/\sigma_i)_r$. Starting at A on the abscissa in quadrant (2), one moves vertically to B on the line for an assumed speed ratio, then horizontally into quadrant (1) to point C, the intersection with the pressure-rise curve. Reading from C downward to the abscissa of quadrant (1) gives the operating percentage of maximum capacity of the blower. From C, reading upward to D, the intersection with the torque curve, then horizontally into quadrant (2) to the assumed speed-ratio line, defined by B, then vertically into quadrant (3) to E, the intersection with the relative density-ratio line (referred to blower inlet conditions) and finally into quadrant (4) to point F, which defines the motor-speed ratio at G and per cent torque at H. The operating speed ratio for the corrected equipment resistance given by A is defined when the assumed speed ratio B and the motor speed ratio at G resulting from the described construction are equal. Hence, a trial-and-error solution is required for each operational condition of the system. At the determined operating point, actual torque developed by the motor and required by the blower is equal to the per cent torque at H in quadrant (4), multiplied by the torque corresponding to the 100-per cent point in quadrant (4). Similarly, the speed of the blower-motor combination is found by multiplying the speed ratio at G by the speed corresponding to the reference condition. The horsepower, being proportional to the product of speed and torque, is equal to the combined product of the speed ratio, the per cent torque, and the horsepower at the reference point, divided by 100. Volume flow of air at the operating point is equal to the product of the volume flow at the reference condition, the speed ratio, and the ratio of the per cent maximum capacity at the operating point to that at the reference point. The weight rate of air flow is given by the product of the weight flow at the reference condition, the relative density ratio, and the ratio of per cent maximum capacity at the operating point to that at the reference point.

It is very probable that with reference design conditions established at the maximum altitude of operation (minimum air density), the scales provided on the chart will not have sufficient range or accuracy to allow a solution to the problem over the entire range of altitude to be considered. This difficulty is readily overcome by selecting, as the new reference condition, another altitude, below design altitude, at which the operating conditions of the blower-motor unit have previously been established. To accomplish this shift in reference conditions, it is necessary to replot the corrected pressure rise and torque curves of the blower in quadrant (1), and the motor characteristic curve in quadrant (4). The value of the corrected blower pressure rise at the altitude selected as the new reference is changed to represent 100 per cent on the $(\Delta p/\sigma_i)_r$-scale. The per cent pressure rise and torque in quadrant (1) must become 100 per cent at the same per cent maximum capacity of the blower that occurred for operation under the same conditions but for the previous reference conditions. All points on the per cent pressure-rise curve in quadrant (1) must be changed in direct proportion to the change in pressure rise at the new 100 per cent point. The speed ratio and relative density ratio are again unity for the new reference point, i.e., the actual blower speed at the new reference condition becomes the reference speed. In quadrant (4) the motor speed ratio and per cent torque evaluated at the operational condition corresponding to the new reference condition must be shifted to unity and 100 per cent, respectively. Other points describing the speed-torque curve of the motor are determined from the original speed-torque curve in direct proportion to the shift of the reference point.

Figure V-21 illustrates the shifting of the reference point, as discussed above, from 60 to 80 per cent maximum blower capacity. The solid lines describe the operational conditions and the blower and motor characteristics before the shift. The dashed lines illustrate the same conditions after the shift, but now as the new reference conditions. Operating points for the new reference condition change within the chart according to the indicated shift by the letters; point A to A', B to B', etc. A check on the accuracy of procedure in making the shift in reference conditions may be made by noting that all reference points within the chart must fall at values of 100 per cent or unity, with the exception of the per cent maximum capacity of the blower which must remain at the previously established value (80 per cent).

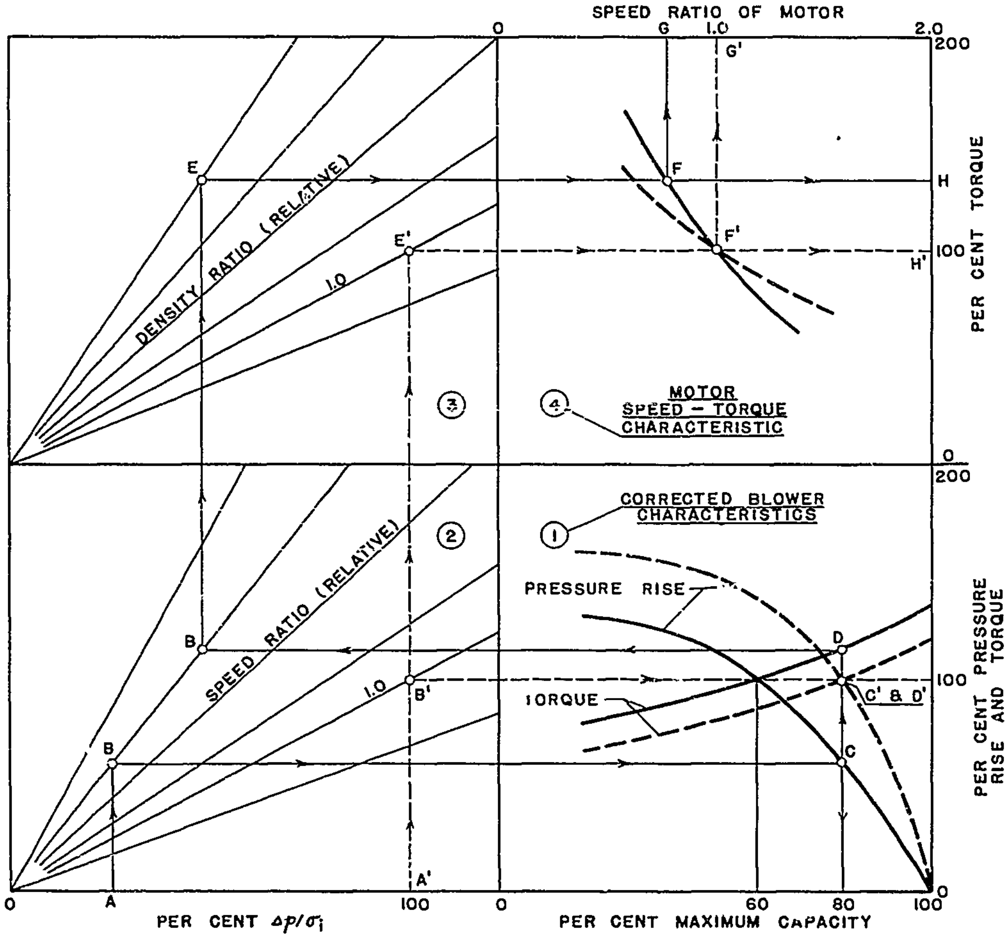

Figure V-21. Shift of Reference Point to Extend Range of Chart (Figure V-19) for Speed Determination of Blower-Motor Unit

After a shift in reference conditions, the numerical values of the various parameters within the chart are appreciably different. For example, suppose operation of a system at an altitude of 70,000 feet is used as the initial reference point. Say that at 40,000 feet the per cent $(\Delta p/\sigma_i)_r$ is evaluated, based on the 70,000 feet condition, as 15, a value sufficiently low on the pressure scales that a change in the reference point is desirable. Making the shift at this point moves the $\Delta p/\sigma_i$ of 15 per cent, based on 70,000 feet, to 100 per cent based on 40,000 feet (A to A'). Hence, for the next lower altitude, say 30,000 feet, if the per cent $(\Delta p/\sigma_i)_r$ based on 70,000 feet is 7.5, it now becomes 50 per cent for the new reference altitude of 40,000 feet. Similar considerations apply to speed ratios, density ratios, per cent torque, etc. Should, for example, the speed ratio be 0.40 at 40,000 feet relative to 70,000 feet, and then be found equal to 0.80 at

30,000 feet relative to 40,000 feet, the over-all speed ratio at 30,000 feet relative to 70,000 feet is 0.8 x 0.4 = 0.32.

b. Forced Flow Arrangement

Based on the data determined for the chosen reference conditions, it is convenient to convert the equipment characteristics into a plot of percentages of standard pressure drop $(\sigma_1 \Delta p)_r$ versus W_r, as shown in Figure V-22. This particular plot is derived for an equipment defined by $\sigma_1 \Delta p = 830\ W^{2.5}$ and the operating point determined in the description given on pages 106 to 109 (240 cfm, $\sigma_{1-r} = 0.0942$, $W_r = 0.0288$ pound per second, $(\sigma_1 \Delta p)_r = 0.118$ in. water).

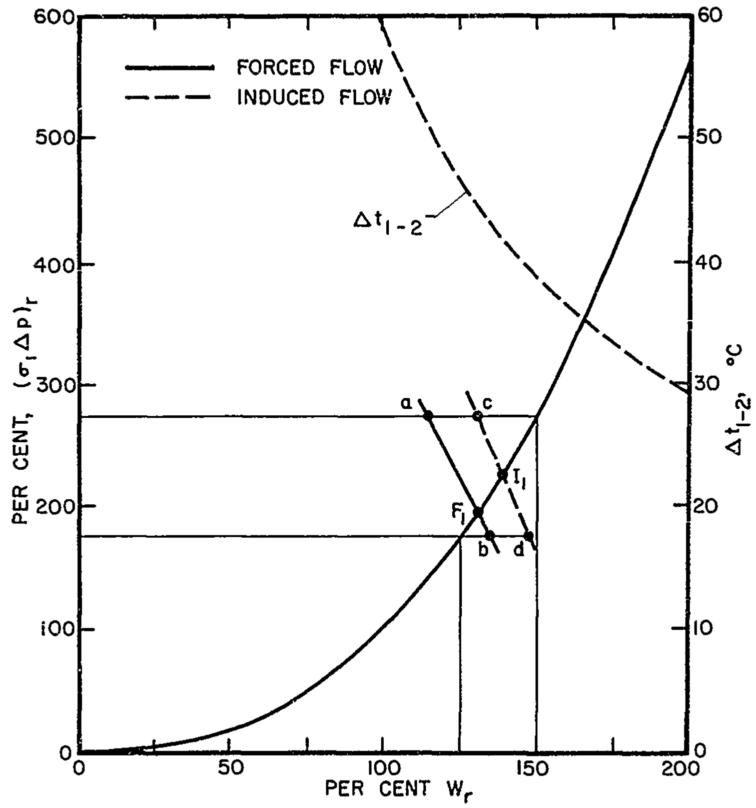

Figure V-22. Relative Equipment Resistance Characteristics

In the chart of Figure V-19, the curves of Figures V-17 and -18 would be plotted in terms of percentages, based on the operating point of the blower at the reference condition, such as F'' shown in Figure V-17 (120 cfm maximum capacity, 3.32 in. water corrected pressure drop and 0.15 hp), and the operating point of the motor at the reference condition, such as F shown in Figure V-18 (0.0494 lb-ft torque, 12,000 rpm). For the blower data the common point of the torque and pressure curves in quadrant (1) must be at 60 percent maximum capacity (120/200). (See Example V-3, page 127.)

For any different operating condition, e.g., 50,000 feet N.A.C.A.

standard atmosphere ($\sigma_o = \sigma_i = 0.1523$), the first step is to assume a percentage of reference weight flow and to determine the corresponding percentage of reference standard pressure drop [e.g., for 150 per cent W_r, 276 per cent $(\sigma_1 \Delta p)_r$]. The relative density ratio (0.1523/0.0942 = 1.618) is used to find per cent $(\Delta p/\sigma_i)_r$ based on the reference conditions by dividing the per cent $(\Delta p/\sigma_i)_r$ by the square of the relative density ratio ($276/1.618^2 = 105.5$). This is the value to be used in the chart to find by successive assumptions, as described above, the corresponding blower speed ratio (0.9) and the per cent maximum capacity (47). The per cent weight flow is then calculated as the product of the ratio of per cent capacity, the relative density ratio and the speed ratio [(47/60) (1.618) (0.9) = 114]. The value so determined should check the assumed per cent weight flow. If it does not, it is convenient to plot it against the per cent pressure drop corresponding to the assumed weight flow (a, Figure V-22) to ascertain the deviation and to permit a better next assumption for the per cent weight flow. If the second point (b, Figure V-22, assumed 125 per cent) is also close to the equipment curve, the operating point (F_1) may be found by intersecting the curve with a straight line drawn through the two points. Thus the per cent weight flow and standard pressure drop at the new altitude are determined and the corresponding values of relative speed ratio, relative density ratio, and per cent torque may be found by interpolation or use of Figure V-19, basing the trial-and-error procedure on the value of $\Delta p/\sigma_i$ shown applicable to this altitude (50,000 feet) by the manner in which F_1 in Figure V-22 has been found.

If in the analysis of a forced-flow arrangement it is necessary to shift the reference condition, as discussed on pages 113 to 115, it is merely necessary to refer all determined values to the new reference condition. This is principally necessary in making each assumption of weight flow and in checking it. Thus for any assumed weight flow, the per cent $(\Delta p/\sigma_i)_r$ referred to the new reference is the ratio of the per cent $(\Delta p/\sigma_i)_r$ referred to the original reference to the per cent $(\Delta p/\sigma_i)_r$ characteristic of the system operation at the new reference. The per cent $(\Delta p/\sigma_i)_r$ referred to the original reference is determined by use of the relative density ratio of the operating condition to the original reference condition. However, in using the per cent $(\Delta p/\sigma_i)_r$ referred to the new reference in the chart of Figure V-19, modified as previously discussed and shown schematically in Figure V-21, the relative density ratio is that of the operating condition to that of the new reference condition. As pointed out before, it is sufficiently accurate to use throughout the density ratio corresponding to environmental conditions. Also, in converting the results to the original reference conditions, they must be multiplied with the known ratio values of the new to the original reference.

c. Induced Flow Arrangement

The procedure for the induced-flow arrangement is similar to that with forced flow, but somewhat more complex. The conversion of the equipment characteristic is the same and will give the identical curve, as shown in Figure V-22, although the basic values would be different (283 cfm, $\sigma_1 = 0.0942$, $\sigma_2 = 0.0718$, $W_r = 0.0259$ pound per second, $(\sigma_1 \Delta p)_r = 0.0895$ in. water). In addition, a plot of temperature rise versus per cent weight flow is necessary with the maximum altitude condition as reference, as shown in Figure V-22 for the specific case described before ($\Delta t_{1-2} = 1.5/0.0259 = 58^{o}C$).

The curves in quadrants (1) and (2) of Figure V-19 are drawn in the same manner, based on the pertinent reference conditions (Blower: 131 cfm, 200 cfm maximum capacity, 2.9 in. water corrected pressure drop, and 0.161 hp. Motor: 0.043 lb-ft torque, 13,000 rpm).

For any different operating condition, e.g., 50,000 feet N.A.C.A. standard atmosphere (3.426 in. mercury, $-55.3^{o}C$, $\sigma_o = \sigma_1 = 0.1523$), the first step is to assume a percentage of reference weight flow and to determine the corresponding percentage standard pressure drop and temperature rise (e.g., for 150 per cent W_r, 276 per cent $(\sigma_1 \Delta p)_r$ and $\Delta t_{1-2} = 38.5^{o}C$). The corresponding actual pressure drop must be determined to permit evaluation of the density ratio at the blower inlet. For example, $\sigma_1 \Delta p = 2.76$ $(\sigma_1 \Delta p)_r = 2.76 \times 0.0895 = 0.247$ in. water, and $\Delta p = (\sigma_1 \Delta p)/\sigma_1 = 0.247/0.1523 = 1.62$ in. water. The density ratio at the blower inlet ($\sigma_2 = 0.1242$) is based on these values of pressure drop and temperature rise and the environmental conditions. In the subsequent determination of per cent $(\Delta p/\sigma_i)_r$ at the blower inlet to be used in the chart, two relative density ratios must be used since in the reduction of per cent $(\Delta p/\sigma_i)_r$ to Δp the density ratio at equipment inlet is applicable, while in the determination of $\Delta p/\sigma_i$ from Δp, the density ratio at the equipment outlet ($\sigma_2 = \sigma_i$) is applicable. The relative values must be referred to the pertinent density ratio determined for the reference condition. Thus, for example, $\sigma_1/\sigma_{1-r} = 0.1523/0.0942 = 1.62$ (like for forced flow), but $\sigma_2/\sigma_{2-r} = 0.1242/0.0718 = 1.73$. Then in finding the per cent $(\Delta p/\sigma_i)_r$ not the square of one relative density ratio, but the product of both values is used [per cent $(\Delta p/\sigma_i)_r = 276/(1.62 \times 1.73) = 98.5$].

The subsequent procedure in the use of the chart is the same as described for forced flow. In the chart only the relative value of $\sigma_2 = \sigma_1$ (1.73) is used. It is also used to check the weight flow after the compatible per cent maximum capacity (56) and the speed ratio (0.88) are determined. (per cent $W_r = (56/65.5)(1.73)(0.88) = 130$). The successive assumptions of per cent weight flow yielding points such as c and d in Figure V-22, close to the equipment curve, are sufficient to determine the operating point I_1. For the percentages of $(\sigma_1 \Delta p)_r$ and W_r so determined, the entire procedure outlined above must be repeated to determine all other corresponding values for the specified environment (50,000 ft.).

If a shift of reference conditions is necessary in the evaluation of induced-flow systems, it is made in the manner described on pages 113 to 115. To perform the evaluation at another operating condition, based on the new reference, the procedure outlined above for determination of per cent $(\Delta p/\sigma_i)_r$ is followed based on the original reference. The ratio of the value so determined to that pertinent to the operation at the new reference condition is used in the chart of Figure V-19. Also, the relative density ratio in the chart is that for the equipment outlet and is determined by dividing the actual density ratio by that pertinent to operation at the new reference condition. The results referred to the original reference are obtained in the same manner as for forced flow.

Control of Blower Performance

Ideally, the function of a cooling blower is to maintain the temperature of all critical components in the equipment constant under all operating conditions. The natural tendency of a blower equipped with a generally available electric drive motor is to provide excessive air flow at low altitude with substantially increased power requirement which governs the size of the motor. The objective in control may be temperature control and/or limitation of required driving power.

In the following, several methods of control are described. Units incorporating their use may not be readily available. However, some of them require only a limited design and development effort which may well be within the capabilities of organizations concerned with the development and application of airborne electronic equipment. Unless otherwise indicated, each method is applicable to centrifugal as well as axial-flow blowers.

1. Throttling

This method can provide close temperature control, but is not applicable over wide ranges of air density, nor does it limit the driving power. The system requirements at reduced altitude are met by introducing a pressure loss caused by a damper somewhere in the flow passage. The effect on the blower operation is illustrated in Figure V-23. It can be seen that, since at reduced altitude less volume flow is required, the blower operating point is shifted towards the pulsation region. For axial-flow units the variation in flow volume which may be so obtained without reaching the pulsation limit is relatively small. For centrifugal units the variation is greater but is usually only sufficient to cover a limited altitude range, unless the operating point at minimum air density approaches the point of

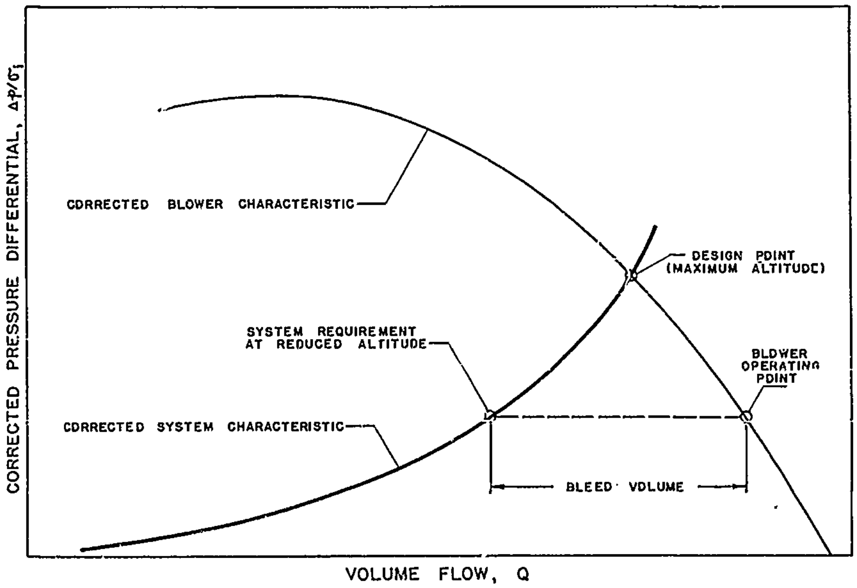

Figure V-23. Blower Control by Throttling maximum discharge. This, however, would be undesirable in respect to blower size and power requirement. In general, the required output of the drive motor increases appreciably with reduction in operating altitude, i.e., with increase in air density. Compared to operation without throttling, the power requirement is somewhat smaller.

Automatic operation of the throttling device is practical. It is most desirable to refer to an equipment temperature which must be maintained. Simplicity may be gained by reference to environmental conditions which, however, may give less accurate temperature control.

Because of limitation in the practical altitude range, necessary to avoid pulsation and oversizing the drive motor, throttling can rarely be utilized alone. It may, however, serve as a convenient means of improving the accuracy and of reducing the complication of other control systems if employed for auxiliary purposes, as discussed subsequently.

2. Bleeding

This method can also provide close temperature control but is objectionable principally because of great power requirements. The requirement of preventing overcooling at reduced altitude is met by bleeding off the air flow in excess of that needed to maintain the desired equipment temperature. The effect on the blower operation is illustrated in Figure V-24. It can be seen that the operating point is shifted toward the point of maximum discharge. Thus the range of altitude of which bleeding can be employed is great. However, at a great expense in power, requiring over-sizing the drive motor, since at minimum altitude only a very small portion of the air flow would be utilized while the blower would operate at extremely low efficiency. Thus, the power requirement would be much greater than when no control is used and the equipment is overcooled.

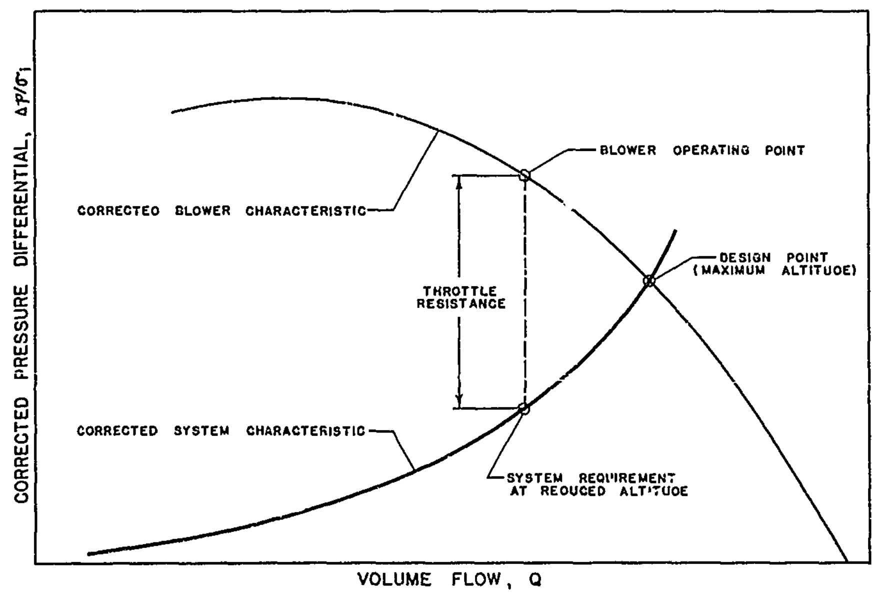

Figure V-24. Blower Control by Bleeding

The application of bleeding should be limited to auxiliary purposes. In conjunction with another principal control method, bleeding may be desirable for precise temperature control. In combination with throttling, the additional objective of keeping the operation of the blower out of the pulsation region can be attained to insure stable performance.

3. Speed Variation

The method of providing continuously variable controlled speed is best suited to match equipment requirements and blower performance at all environmental conditions. It also results in the lowest power requirements of the blower. The effect of speed variation on blower operation is illustrated in Figure V-25. The operating point at reduced altitude is specified

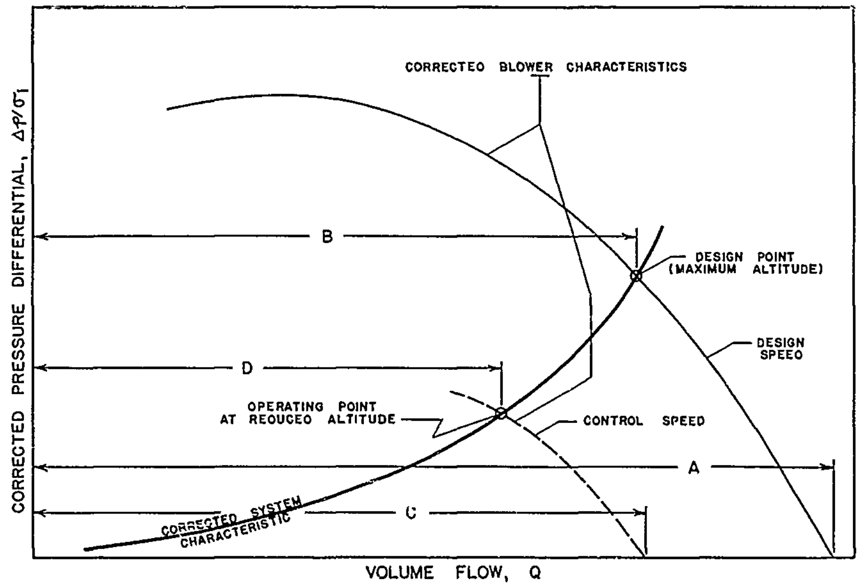

Figure V-25. Blower Control by Speed Variation

by the corrected system (equipment) resistance curve. The speed reduction is such that the volume flow and pressure requirements are met. The operating point on the blower characteristic is defined as a percentage of the maximum capacity at the given speed. Since the blower laws indicate that the static pressure varies as the square of the speed while the discharge volume varies in direct proportion to the speed, the operating point on the characteristic curve must shift, unless the system resistance and air flow requirement happen to vary in the same manner. As a rule, the shift in percentage of maximum capacity will be slight. Therefore, the blower would operate at almost the same efficiency. The relationships are illustrated by the dimensions in Figure V-25. The control speed is related to design speed in the ratio C to A. The ratio D to C, expressing the percentage of maximum capacity at control speed, is not necessarily the same as the ratio B to A, which expresses the percentage of maximum capacity at design speed. The control speed can be determined conveniently by use of the chart in Figure V-12, as described on pages 100 to 102.

The power requirements at reduced altitude decrease rapidly and are normally quite low in the vicinity of sea level, unless the air source temperature is high relative to the temperature of the heat transfer surfaces. Whatever control apparatus may be utilized, it is permissible that its efficiency at reduced altitude be lower because of the lower power requirement and improved cooling of the drive motor. Methods which appear feasible for incorporation in electric drives with speed ratio of ten to one expected to be required, are (1) a constant-speed induction motor with a magnetic fluid clutch, (2) variation of armature circuit resistance in a shunt-wound direct-current motor, and (3) variation of rotor resistance in a wound-rotor induction motor. In these applications, the control devices, such as the clutch or the series resistors, would have increasing losses as the altitude is reduced. These losses would be greatest at about half-speed where sufficient cooling air would be available to dissipate the losses readily. Automatic operation of these devices could be accomplished without undue difficulties.

Where good temperature control and greatest power economy are required, the application of continuously variable speed is most desirable. Beyond the development of the basic means of control, the method is less complicated than others which must usually employ a combination of at least two basic methods to accomplish the same effect.

With most combined control methods, part of the control is accomplished by use of a multi-speed motor. A multi-speed motor alone could provide approximate temperature control over a reasonably small range, but would also have to supply increased power at each operating speed when the air density is increased. Three- or four-speed devices may fulfill requirements over an altitude range of 60,000 feet. Combined with other auxiliary means, a two-speed motor may also accomplish the objectives of close temperature control.

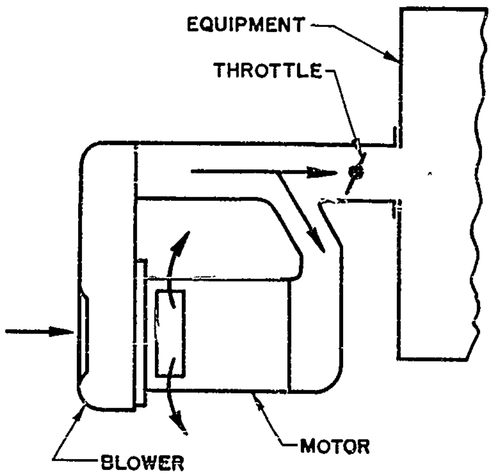

Figure V-26. Schematic of Control Arrangement with Throttling and Forced Motor Cooling

An arrangement to utilize throttling, a two-speed drive, and parallel cooling air flow through the equipment and the motor may be as shown schematically in Figure V-26. A portion of the blower discharge is utilized to cool the motor. A throttle in the equipment's flow system restricts the cooling air only to that required to maintain constant component temperatures. The excess air delivered by the blower is bypassed through the cooling passages of the driving motor. As the altitude is reduced, the weight flow through the motor increases. This permits overloading the motor until limiting internal temperatures are reached. At that altitude the speed of the unit is reduced. Over the range from 60,000 feet to ground level, a two-speed motor with a four-to-one ratio would usually be adequate.

The motor would be not much larger than one designed to operate only at the maximum altitude.

Temperature control with multi-speed motors can be accomplished by throttling or bleeding. Since the application of throttling at any given speed would then only be required over a relatively small altitude range, pulsation trouble is not likely to be encountered. The power requirement would be lower than without throttling. In contrast, if temperature control at the various speeds is to be accomplished by bleeding, the power requirements would be greater than without control. In general, a three-speed motor with throttling or bleeding would insure constant temperature for a range of about 60,000 feet altitude. With throttling, the design output of the motor at maximum speed would have to be about twice that required at the maximum altitude because of the rising power requirement with reduced altitude.

4. Impeller Width Variation

To produce the effect of an impeller of variable width by partial shielding of the impeller tip section represents a method of performance control for centrifugal blowers which has not been utilized heretofore. It provides a means of holding the power to that required at maximum altitude. With this type of control the pressure-producing ability of the blower remains essentially unaffected, while the volume flow delivered by the blower for any rotational speed and impeller diameter varies nearly in direct proportion to the free width of the impeller. Thus, the power requirements with this type of control are almost in direct proportion to the width setting and do not include required input power not utilized for the delivery of air at required static pressures.

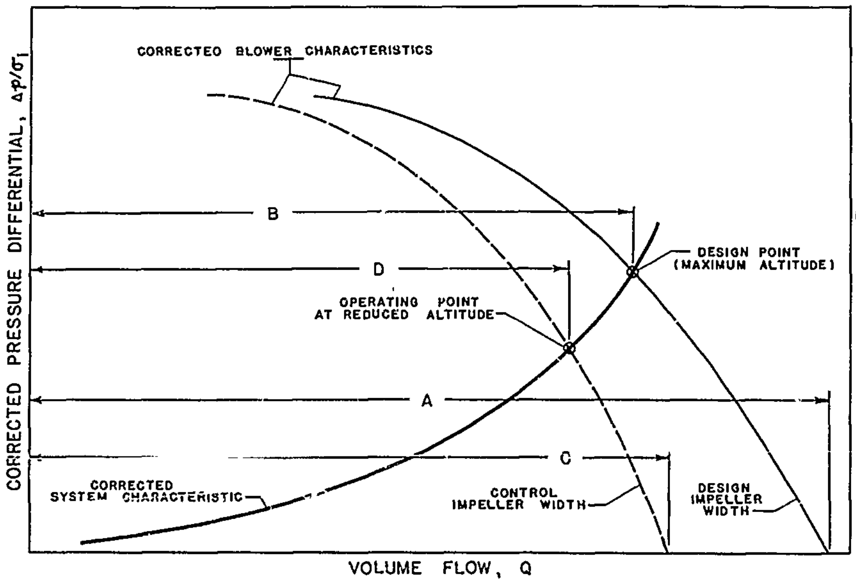

Figure V-27. Blower Control by Variation of Impeller Width

Figure V-27 illustrates the possible process of control by variation of impeller width. The reduction in impeller width is such that the volume flow and pressure requirements are met at the reduced altitude. The control impeller width is related to design impeller width in the ratio of C to A. The ratio D to C, expressing the percentage of maximum capacity at control width, is always greater than the ratio B to A, which expresses the percentage maximum capacity at design width. The advantages of width variation are best utilized when employed in conjunction with a two-speed motor and a throttle. By this combination, the required range of width variation is reduced and the blower operates within the range of maximum efficiency at all times without exceeding the power requirement at maximum altitude.

Application of this method would require a considerable development effort since a blower of relatively complex mechanical design must be employed. In this particular respect the method differs from all others previously mentioned. The former do not necessitate any modification of a standard blower, but merely require a driving device or auxiliary control elements of specific characteristics.

5. Vane Angle Variation

The variation of inlet guide vane angle and/or rotor vane angle are feasible control methods for axial-flow blowers. The variation of rotor vane angle necessitates complex mechanical design which is not likely to be justifiable for blowers of the relatively small size required to cool electronic equipment. The variation of inlet guide vane angle is practical for control over a limited range of operation. When used in conjunction with variable speed control, it serves to reduce greatly the range of required speed variation, and allows the blower to operate at, or near, its peak efficiency.

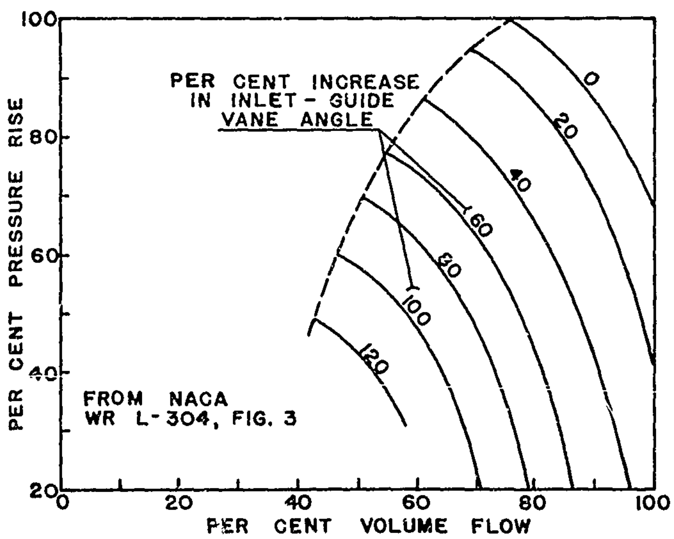

Figure V-28. Effect of Inlet Guide Vane Angle on Performance of Axial-Flow Blower

The effect of variation of inlet guide vane angle at constant speed is shown in Figure V-28. A practical variation of volume flow of two to one is indicated. This would permit the use of a two-speed motor, in addition to vane angle variation, to insure required flow variation and temperature control.

Examples

Example V-1. Application of Blower Laws

According to data from a manufacturer's catalog, a centrifugal blower having an impeller diameter of 6-5/8 inches and an impeller width of 2-7/8 inches develops 2 inches water static pressure rise, delivers 520 cubic feet of air per minute and requires 0.34 horsepower input when operating at 3000 revolutions per minute rotational speed. The air density at the blower intake is 0.075 pounds per cubic foot. It is required to determine the performance of a geometrically similar (homologous) blower of 5 inches diameter, operating under the following conditions:

rotational speed, 11,000 revolutions per minute
ambient air pressure, 5.0 inches mercury absolute
ambient air temperature, -30°C

First, it is desirable to correct the reference performance to standard conditions, as used in this manual, i.e., corresponding to ρ_r = 0.0765 pound per cubic foot. The value of σ_i corresponding to the reference performance given is

$$\sigma_i = 0.075/0.0765 = 0.98$$

so that by use of equations (V-6c, -7c, and -8c)

$$\Delta p_{corr} = \Delta p/\sigma_i = 2.0/0.98 = 2.04 \text{ inches water}$$

$$Q_{corr} = Q = 520 \text{ cubic feet per minute}$$

$$hp_{corr} = hp/\sigma_i = 0.34/0.98 = 0.347$$

and by Figure V-9 or equation (V-1)

$$W = 0.650 \text{ pound per second.}$$

For the actual conditions of operation, p_o = 5 inches mercury absolute and t_o = -30°C, therefore from Figure V-8, σ_i = 0.198. The static pressure rise of the air for the actual conditions of operation will be, by equation (V-6)

$$\Delta p \;\alpha\; \sigma_i\, D^2\, N^2$$

so that

$$\Delta p = 0.198\,(5/6.625)^2(11{,}000/3000)^2(2.04) = 3.09 \text{ inches water}$$

The volume rate of flow is calculated by equation (V-7)

$$Q \;\alpha\; ND^3$$

$$Q = (11,000/3000)(5/6.625)^3(520) = 820 \text{ cubic feet per minute}$$

and the input horsepower by equation (V-8)

$$hp \;\alpha\; \sigma_i \, N^3 \, D^5$$

$$hp = 0.198\ (11,000/3000)^3(5/6.625)^5(0.347) = 0.83$$

The weight rate of air flow is defined by equation (V-9) or Figure V-9

$$W \;\alpha\; \sigma_i \, Q$$

$$W = 0.198\ (820/520)(0.650) = 0.203 \text{ pound per second.}$$

For the above-described performance the width of the impeller is the homologous width and is

$$B = (2.875)(5/6.625) = 2.17 \text{ inches,}$$

where 2.875 is the width in inches of the reference blower.

If the blower were not homologous, i.e., the width of the impeller would not be proportioned in reference to the diameter in the same way as the width of the reference blower is proportioned to the diameter of the reference blower, corrections of the values calculated in the preceding would be required to account for width variation. For example, if the actual width is 2.5 inches, the width ratio is

$$2.5/2.17 = 1.15$$

so that the actual performance of the blower is calculated to be, by equations (V-6d, -7d, -8d, and -9d),

$$\Delta p = 3.09 \text{ inches water}$$

$$Q = 820 \times 1.15 = 943 \text{ cubic feet per minute}$$

$$hp = 0.83 \times 1.15 = 0.955$$

$$W = 0.203 \times 1.15 = 0.234 \text{ pound per second.}$$

Should this blower be operated at the same speed under standard sea-level conditions, its performance would be

$$\Delta p_{corr} = \Delta p/\sigma_i = 3.09/0.198 = 15.6 \text{ inches water}$$

$$Q_{corr} = Q = 943 \text{ cubic feet per minute}$$

$hp_{corr} = hp/\sigma_i = 0.955/0.198 = 4.82$

$W_{corr} = W/\sigma_i = 0.234/0.198 = 1.18$ pounds per second

Example V-2. Use of the Blower Design and Performance Chart to Determine Required Blower Speeds at Various Altitudes

This example illustrates the procedure used for determining blower operation necessary to match system requirements. An electronic unit having a case to be cooled by forced flow requires the performance given in Table V-3 of the cooling blower, so as to prevent overheating of its components at the various altitudes of operation.

Table V-3. Required Blower Performance for Cooling of Unit at Various Altitudes (Example V-2)

Equivalent compartment altitude, feet	Compartment density ratio, $\sigma_o=\sigma_i$	Equipment pressure drop, Δp, inches water	Blower inlet air rate, Q, cubic feet per minute
sea level	1.0	0.30	75
10,000	0.738	0.41	102
20,000	0.533	0.50	133
30,000	0.374	0.90	210
40,000	0.245	1.45	325
50,000	0.152	2.50	550

If the unit's requirements are to be met by control of the blower speed, the problem is to determine the rotational speeds required at the various altitudes of operation. The reference performance of the blower for unity width ratio, $\sigma_i = 1.0$, and 7200 revolutions per minute, is as follows:

Q	150	200	250	300	350	400
$\Delta p/\sigma_i$	4.4	4.0	3.33	2.4	1.28	0
hp/σ_i	0.226	0.266	0.311	0.358	0.405	0.455

First, it is necessary to plot in quadrant (1) of Figure V-12 points of pressure drop versus corresponding air rates required at the various altitudes. Then the reference performance of the blower is inserted in quadrant (5) of Figure V-12, with the corrected static pressure rise $\Delta p/\sigma_i$ and corrected input horsepower hp/σ_i plotted as a function of the volume rate of air flow Q. To improve the accuracy of the chart, the volume flow and corrected horsepower values are doubled when the reference performance data of the blower are plotted on the grid in quadrant (5). This requires the use of a width ratio of 0.5 in quadrants (3) and (8). Since the diameter of the impeller is constant in this example, the diameter ratio for quadrants (2) and (6) is unity in all cases. Hence, the basic control quadrant for the determination of the required rotational speed is quadrant (4), which contains speed ratios for flow volume and system resistance. The necessary procedure is that described in part (4b), page 100. To find the required power, the procedure described on page 98 is used for the operating point established for each altitude. The torque, in pound-feet, is calculated from the equation

$$\text{Torque} = 33{,}000\ (\text{horsepower})/2\pi\ (\text{revolutions per minute}).$$

The results are given in Table V-4.

Table V-4. Required Speed Variation, Power, and Torque of Blower with Altitude (Example V-2)

Altitude, feet	Speed ratio	Speed, revolutions per minute	Input power, horsepower	Torque, pound-feet
S.L.	0.30	2160	0.0084	0.020
10,000	0.41	2950	0.016	0.0285
20,000	0.53	3820	0.025	0.0345
30,000	0.84	6050	0.069	0.060
40,000	1.30	9350	0.166	0.093
50,000	2.20	15,850	0.50	0.169

Example V-3. Performance of Equipment-Blower-Motor Combination at Variable Altitude

The values for the blower and motor performance curves to be plotted in quadrants (1) and (4) of Figure V-19 are based on the characteristics given in Figures V-17 and -18 and the reference conditions indicated by points F" and F thereon. The pressure and torque of the reference blower expressed in terms of per cent maximum capacity of the blower at a fixed rotational speed, and based on reference point F are:

Per cent maximum capacity	30	40	50	60	70	80	90	100
Per cent pressure rise	156	144	124	100	75	50	25	0
Per cent torque	64	75	87	100	114	128	142	156

The speed-torque variation of the electric motor, based on reference point F", is:

Per cent torque	20	60	100	140	180	220
Per cent speed	250	153	100	66	38	15

The equipment characteristics presented in Figure V-22, based on reference point F' in Figure V-16, are used to evaluate the system operation at altitudes from 60,000 feet to sea level according to N.A.C.A. standard atmosphere. Using the procedures described on pages 115 to 116, the results given in Table V-5 are obtained. The full range of the characteristics given in Table V-5 could only be determined by shifting the reference values, as

Table V-5. Variation of Performance of Equipment-Blower-Motor Combination with Altitude (Example V-3)

Altitude	Speed-ratio	Corrected system resistance, per cent	Motor torque, per cent	Weight air flow, per cent	Per cent maximum capacity of blower
60,000	1.0	100	100	100	60
50,000	0.85	77	114	132	58
40,000	0.73	59	128	174	55
30,000	0.62	45	141	220	53
20,000	0.55	36	152	268	51
10,000	0.49	29	163	319	50
Sea level	0.43	24	170	375	49

discussed on pages 113 and 114. For example, a shift is made to 20,000 feet. It is necessary to replot the pressure rise and torque curves of the blower in quadrant (1). The operating points on the blower characteristic curves in quadrant (1) for an altitude of 20,000 feet with the reference altitude at 60,000 feet are defined from quadrant (2) by a percentage $(\Delta p/\sigma_i)_r$ of 36.2 and a speed ratio of 0.55, or directly in quadrant (1) by the per cent maximum capacity of the blower of 51. In quadrant (1) on the abscissa, at 51 per cent maximum capacity, the per cent torque and per cent pressure rise for the blower are 89 and 120, respectively. The two characteristic curves of the blower are to be shifted so that these points become 100 per cent on the ordinate of quadrant (1) at the same per cent maximum capacity of 51. All other points on the two original curves are shifted in the same proportion. The numerical values for several points on the curves, resulting from the shift, are:

Per cent maximum capacity of blower	30	40	51	60	70	80	90	100
Per cent pressure rise	129	119	100	83	63	42	21	0
Per cent torque	72	84	100	112	127.9	143.5	159.5	175.5

In quadrant (4) the operating point on the motor characteristic curve for an altitude of 20,000 feet with the reference altitude at 60,000 feet is at a per cent torque of 152 and a speed ratio of 0.55. This point is shifted so that the per cent torque becomes 100 and the speed ratio unity. All points on the original curve are shifted in the same proportion as the reference point. Values for points on the motor characteristic curve resulting from the shift in reference conditions are:

Per cent torque	80	100	120	140	150	164
Speed ratio of motor	1.42	1.0	0.66	0.34	0.19	0

The procedures to determine operating characteristics at altitudes below the new reference altitude are those described on pages 115 to 116. In order to refer the results to the original reference altitude, as shown in the results of Table V-5, the values relative to 20,000 feet altitude are multiplied with those for 20,000 feet altitude relative to 60,000 feet altitude.

CHAPTER VI

USE OF BENCH TEST DATA FOR DETERMINATION OF OPERATIONAL THERMAL CONDITIONS IN THE STEADY STATE

In this Chapter, methods of computation and analysis, based on bench test data obtained according to procedures outlined in Chapter IV, are presented. The methods are applicable to the determination of case and component temperatures and other characteristics of a unit at conditions of altitude, compartment air temperature, and radiation environment other than those at which the unit was tested.

Pressurized Units Externally Cooled by Free Convection and Radiation

The internal and external mechanisms of heat transfer of this type unit are discussed in Chapter II, page 6. They are summarized schematically in Figure VI-1. The indicated internal modes of heat transfer are affected only by the case temperature since the internal air pressure and air circulation rate would be practically constant. The external modes of heat transfer, however, are dependent on the environmental conditions. Assuming the unit so mounted that conduction from the case would be negligible, heat dissipation can only occur by radiation and convection. Their relative contributions would depend on the environmental air temperature, pressure and velocity, and the temperature and surface characteristics of surrounding objects and enclosures. If no external means are used to create air flow over the case surface, the convection is natural or free.

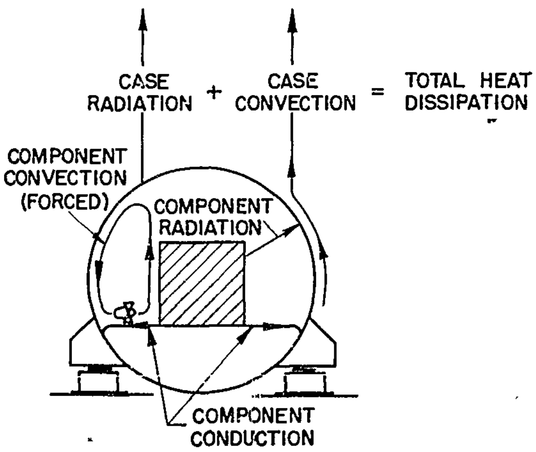

Figure VI-1. Heat Transfer Diagram of Pressurized or Sealed Unit Cooled by Free Convection and Radiation

The component temperatures are direct functions of the case temperature. Therefore, if the test data yield a correlation between individual component temperatures, and a representative case temperature, the component temperatures can be predicted under all environmental operating conditions for which the case temperature may be calculated. For that purpose, calculation methods are presented in the following which involve the use of generally accepted free convection and radiation heat transfer equations. Probable deviations of calculated values from actual values are compensated by means of correction factors determined on basis of the test data.

1. Free Convection

The rate of heat transfer q_{cv} by free convection from a heated surface to air can be evaluated by the equation

$$q'_{cv} = h'_{cv} S' \Delta t', \tag{VI-1}$$

where all terms must be in consistent units, and h'_{cv} represents the convective heat transfer coefficient, being the time rate of heat transfer per unit surface area S' and per degree temperature difference $\Delta t'$ between the surface and the air. The free convective heat transfer coefficient is dependent on the temperature level, as defined by a film temperature t'_f, the temperature difference $\Delta t'$, and the configurational characteristics of the surface, as indicated by a significant dimension L' and the magnitude of a constant C. The film temperature t'_f is the average of the air and the surface temperature, $(t'_a+t'_{sf})/2$. The coefficient may be calculated from

$$h'_{cv} = C\,(k_f/L')(a_f L'^3 \Delta t')^n, \tag{VI-2}$$

where all terms must be in consistent units. The thermal conductivity k_f and the convection parameter a_f of the air are evaluated at t_f. The convection parameter a_f is defined by $(g'\beta\rho^2 c_p/\mu k)_f$, g being the gravitational acceleration, β the coefficient of thermal expansion, ρ the density, c_p the specific heat at constant pressure, and μ the absolute viscosity, all in consistent units and at t_f. For air, a_f is proportional to $(p'^2 c_p/T'^3 \mu k)_f$ where p' and T' are the absolute air pressure and the film temperature, respectively. The exponent n is dependent on the magnitude of $(aL'^3\Delta t')$, so that if $(aL'^3\Delta t') = 10^3$ to 10^9, $n = 1/4$, if $(aL'^3\Delta t') > 10^9$, $n = 1/3$. In most situations of free convective heat dissipation from electronic components and units $n = 1/4$.

Values of the dimensionless constant C and methods for determining the correct dimension L have been established from many previous laboratory studies of freely suspended vertical plates, horizontal cylinders, vertical cylinders, and planes with the heated surface facing upward or downward. The significant dimension L should be evaluated as follows:

(1) Horizontal planes,

$$L = \Big[(\text{length})(\text{width})\Big] \Big/ \Big[(\text{length}) + (\text{width})\Big].$$

(2) Vertical planes (rectangular),

L = height of plane in vertical direction, but limited to two feet even if height is greater.

(3) Vertical planes (non-rectangular),

L = surface area of plane divided by horizontal width of plane. For circular sections,
L = 0.785 x diameter.

(4) Cylindrical surfaces, horizontal,

L = diameter of cylinder.

(5) Cylindrical surfaces, vertical,

L = height of cylinder, but limited to two feet even if height is greater.

(6) Spheres,

L = one-half diameter of sphere.

By use of the relationships represented by equations (VI-1 and -2), the empirical values for the dimensionless constant C, and the physical properties of air, the charts of Figure VI-2 are constructed. These calculation charts (in back pocket) serve for the rapid determination of free convective heat transfer from surfaces whose dimensions, position, temperature and environment are defined. Figure VI-2a covers a wider range of variables, but Figure VI-2b permits more accurate evaluation in a range usually encountered in problems concerned with electronic components or units. Rates of heat transfer are expressed in watts per square inch so that the charts can be used conveniently in electrical heat dissipation problems.

As applied to calculation of free convective heat transfer from individual surfaces, the charts are reasonably accurate. Their application to combinations of surfaces, as encountered on equipment cases, yields estimated rates of heat transfer which can be correlated with actual heat dissipation by use of corrective procedures. The uses of the charts for the evaluation of test data are illustrated in Examples VI-1, -2, and -3, page 139, and many others in this manual.

a. Combinations of Individual Surfaces. The basic assumption in the use of the charts of Figure VI-2 for calculation of free convective heat transfer from equipment cases with composite surfaces is that the summation of the heat dissipation calculated for portions of the entire surface would represent a fair estimate of the total heat dissipation. Consequently, the entire surface must be broken up into subsidiary surfaces which can be so defined that they can be identified by one of the characteristic curves, shown in the lower left quadrant of Figure VI-2, which are representative of various values for the dimensionless constant C. However, it should be realized that in making use of the chart, experimental data of individual surfaces are applied to composite configuration. Therefore, the mutual influences of the various surfaces, as well as end effects are neglected. Similarly, in determining the characteristic dimension L of each subsidiary surface by use of one of the relationships given on pages 131 and 132, the interrelationships of all surfaces describing the particular equipment case are disregarded because significant dimensions as defined for individual freely suspended surfaces are used. Here then, are two sources of possible error of not predetermined magnitude, usually in the order of less than 20 per cent, but not likely to exceed 50 per cent.

b. Evaluation of Representative Case Temperature

In addition to inaccuracies which unavoidably result in the definition of C and L for portions of an actual body, such as an equipment case, errors in computing the convective heat transfer may also result from the method by which a representative case temperature is determined. The representative case temperature is a calculated temperature obtained by the combination of the temperatures measured on the surface of the case under defined environmental conditions. It is only practical to correlate the temperatures of individual components with one representative case temperature rather than with a variety of surface temperatures which may be obtained by measurement of case temperatures at different points.

In general, it is assumed that thermocouples are spaced fairly uniformly over the case surface so that all temperature measurements represent approximately equal surface areas. Under these conditions an arithmetic average of the measured surface temperatures is sufficiently accurate for definition of a representative case temperature, providing the various measured case temperatures do not vary more than about 30° or 40°C. This permissible variation in temperature decreases as the difference between the average case surface temperature and the compartment air temperature, or the radiation-receiving wall temperature, decreases.

When the measured surface temperatures vary appreciably with respect to position on the case surface, it is necessary to evaluate the free convective heat transfer by subdividing the case surface into subsidiary areas of known average surface temperature. The choice of this area subdivision is not entirely arbitrary since it is necessary to use configurations similar to the various cases which may be analyzed by means of the charts of Figure VI-2. For example, it would not be permissible to subdivide a vertical plane into several horizontal strips of known average temperature and calculate the free convective heat transfer by use of the charts in Figure VI-2, since the state of motion and temperature of the air approaching the top horizontal or the intermediate strips would be entirely different than for the bottom horizontal strip. However, in general, it would be entirely permissible to subdivide a vertical plane into vertical strips of known average surface temperature. By this method of subdivision, the subsidiary areas so defined are fairly descriptive of freely suspended vertical planes and the methods indicated for use with Figure VI-2 are directly applicable.

In subdividing the case surface area, segments should be so defined that the natural convective air flows over them are always in parallel and never in series. The following general procedure should be applied:

(1) Vertical planes. Subdivide into vertical strips of known average surface temperature, as determined by thermocouple location. Two to four vertical strips are generally sufficient for the necessary accuracy of calculation.
(2) Horizontal planes. Do not subdivide a horizontal plane. Use an average of the various measured surface temperatures to determine free convective heat transfer.
(3) Vertical cylinders. Same procedure as for vertical planes.
(4) Horizontal cylinders. Subdivide by vertical segmentation, yielding short cylindrical areas of known average temperature.
(5) Spheres. Use average surface temperature without area subdivision, unless sphere is quite large. For the latter case, the area may be subdivided into lunes by vertical segmentation.

For the majority of surface temperature distributions actually encountered in electronic equipment, the use of the arithmetic average surface temperature without area subdivision, other than the usual breakdown into the various types of surfaces as defined on pages 131 and 132, is satisfactory and most convenient. If, however, the calculated heat dissipation from a case surface differs widely from that determined by electrical measurement (the difference of electrical input and electrical output of the unit), a check on the accuracy of the calculations of heat transfer by free convection should be made by the above-indicated method of area subdivision.

If one or several local hot spots occur on a case surface as a result of a component being attached to or located near the case, these maximum temperatures are best not considered in obtaining the average representative temperature. Other units may be so designed that the chassis blocks off the internal radiation and convection over a surface of the case, causing a region of low temperature. In such designs, the heat transfer, if any, by free convection and radiation from the low-temperature surfaces should be computed separately. If the calculated heat dissipation from such areas is not greater than five per cent of the total heat dissipation, as determined from electrical measurements on the unit, the surfaces should be omitted in obtaining the representative case temperature. When the method of area subdivision is required because of pronounced surface temperature variation, a representative case surface temperature properly related to component temperature is more difficult to define. This problem is treated in the subsequent discussion of heat transfer by radiation on pages 135 to 137.

c. Area of Individual Surfaces. The area S used in computing the free convective heat transfer from a particular portion of the total surface of a case must correspond to the dimensions used in computing the characteristic dimension L. For example, in calculating the heat transmission from the envelope of a cylindrical case, excluding the ends, where L is equal to the diameter, the area S is computed as the product of the cylinder's circumference and its length. In certain designs, fins or other surface extensions are welded to the basic case to increase its free convective heat transfer area. If the fins project perpendicularly from the case, the convective area must include the total surface of the fins, and the representative surface temperature must be evaluated by taking a weighted average including the temperatures and areas of both the fins and the basic case. In those designs which include extended surfaces made up of corrugated sheet metal, seam or spot welded to the basic case, the free convective heat transfer coefficient in the vertical passage formed by the corrugated member and the case cannot be computed accurately. The best procedure for such designs is to determine the convective heat transfer as the difference between the electrically measured heat dissipation and the computed values for the total external radiation plus the free convection from the outer corrugated surface.

2. Radiation

The rate of heat transfer q_{rd} by radiation from one surface to another can be evaluated by the equation

$$q_{rd} = 0.0037\, F_e\, F_A\, S \left[(T_1/100)^4 - (T_2/100)^4\right], \qquad \text{(VI-3)}$$

where q_{rd} is in watts, the surface area S in square inches, and the absolute surface temperatures T_1 and T_2 in $^{\circ}$K, F_e represents a dimensionless factor dependent on the radiation emissivities and the configuration of the surfaces, and F_A is a dimensionless configuration factor.

a. Calculation Charts and Their Application

Values for factors F_e and F_A as functions of the emissivities of the heat exchanging surfaces, their configuration, and environments have been determined by analytical and experimental methods for various situations. The literature also contains a fair quantity of data on surface emissivities.

To facilitate calculation of heat dissipation from electronic components and equipment cases, the basic equation for radiant heat transfer may be rewritten in the form

$$q_{rd} = \phi_1 \phi_2 S, \qquad \text{(VI-4)}$$

where ϕ_1 is a function of F_e and F_A and is, therefore, also a function of the surface emissivities, and ϕ_2 is a function of the temperatures of the heat exchanging surfaces. Generally applicable values of ϕ_1, including some approximations for various equipment arrangements and combinations of surface characteristics are given in Table VI-1. For individual electronic components as installed within an equipment case ϕ_1 may always be taken as 0.9. Charts for ϕ_2 are given in Figure VI-3 (in back pocket), in two ranges separated into Figures VI-3a and -3b. The use of Table VI-1 and Figure VI-3 for the evaluation of test data is illustrated in Examples VI-1, -2, and -3, page 139, and many other examples contained in this manual.

Table VI-1. Radiation Factor ϕ_1 for Various Combinations of Surfaces and Configurations

Case Surface	Environmental Surfaces	Confinement: Close	Intermediate	Large (Laboratory)
Black or painted	Painted or fabric-covered	0.9	0.93	0.95
Black or painted	Dural, unpolished	0.3	0.63	0.95
Alclad, unpolished	Painted or fabric-covered	0.10	0.10	0.10
Alclad, unpolished	Dural, unpolished	0.08	0.09	0.10
Dural, unpolished	Painted or fabric-covered	0.3	0.3	0.3
Dural, unpolished	Dural, unpolished	0.2	0.24	0.3

b. Emissivities and Configuration

Like for the calculation of total heat transfer by free convection, the radiant heat transfer can be interpreted as the summation of radiant heat transfer from individual surfaces. However, more frequently it is possible to evaluate entire equipment cases without subdividing them into representative surfaces which are defined by specific values of ϕ_1 in Table VI-1.

Calculation of radiant heat transfer may be as inaccurate as calculation of free convective heat transfer because several decisions are left to the discretion of the analyst. In his choice of a value for ϕ_1, by identification of surface characteristics and configuration, he is limited to known values. They may not exactly fit the situation at hand, but for lack of exact information, the value of ϕ_1 corresponding to variables which are apparently most applicable must be chosen.

c. Representative Case Temperatures

Like in the calculation of free convective heat transfer, only the use of a single representative case surface temperature is practical. For most surface temperature distributions the use of an average surface temperature for the calculation of radiant heat transfer is satisfactory. When the surface temperature of any case varies widely with position on the case surface, it may be necessary to calculate the total radiation by subdividing the case surface into subsidiary areas of known temperatures. In general, application of this method becomes necessary if 20 to 60 per cent of the case surface is at an average temperature more than 30° to 40°C greater than the remainder of the case surface. Since the shape or size of the subsidiary areas has no effect on the radiation calculations, it is permissible to subdivide the case area in any way suitable for the best definition of the temperatures of the subsidiary areas from the measured values.

The area S in the radiant heat transfer equation (VI-4) is the area of the surface of the case for which the radiation is being computed. When the representative case temperature is used as the average of the measured surface temperatures, the area S is the entire area of the case surface. With area subdivision, the term S represents only the subsidiary area. Then, the total heat dissipation by radiation from the entire case surface is found by the sum of the radiant heat transfer from all subsidiary areas. In those designs which include fins projecting perpendicularly from the case, the areas of such surfaces may usually be disregarded in determining the radiating area. However, with corrugated-type extended surfaces the radiating area and representative case temperature would be that of the external corrugated surface.

When it is necessary to employ the method of area subdivision to determine the total heat dissipation by radiation, the representative case temperature is best obtained by finding, using Figure VI-3, a single surface temperature which would give the same total heat dissipation by radiation. Hence, an effective temperature evaluated on basis of the total heat dissipation by radiation, rather than the average measured surface temperature, would be used as the representative case temperature. The representative case temperature so determined should be satisfactory for correlation with component temperatures so that their temperature variations with varying operational conditions may be predicted. In a few instances where free convective heat transfer represents a large percentage of the total heat dissipated, proper correlation of representative case temperature with component temperatures, as determined on basis of bench test data, may not be obtained if the representative case temperature is determined from the analysis of radiant heat transfer only. Under such circumstances a similar effective temperature describing the free convective heat transfer should be obtained with the aid of Figure VI-2. A study should then be made to determine which temperature or combinations of the effective temperatures yields the proper correlation with measured total heat dissipation and component temperatures. However, should the average of measured surface temperatures be reasonably accurate in accounting for the total heat dissipation by defining radiant and free convective heat transfer under bench test conditions, it should be also employed as a representative case temperature to define the variations in component temperature under other operational conditions.

3. Determination of Correction Factors for Evaluation of Extended Operational Conditions

The principal reason for the calculation of free convective and radiant heat transfer rates under bench test conditions is to determine the adequacy of the methods of calculation, by comparing the results obtained with the known value of total heat dissipation derived from electrical measurements. (When electrical output measurements are very complicated, it is usually advisable to make a thermal calibration of the load to establish by means of a measurable energy source the variation of a reference temperature gradient within the load system as function of the input, simulating the unit's output.) It is necessary to compare calculated and measured heat dissipation rates because under operational conditions in aircraft installations, the relative effects of free convection and radiation may be different than under bench test conditions. However, the only means of predicting case surface temperatures under such conditions is dependent on the ability to calculate the unit's free convective and radiant rates of heat dissipation independently and on the basis of assumed surface temperatures. That case surface temperature for which the sum of both rates of heat transfer equals the known total rate is the operational case temperature to be expected.

In the preceding discussion of the calculation of free convective and radiant heat transfer, the factors which may contribute to increase the probability of erroneous results are pointed out. Thus, when bench test data are used, the total calculated heat dissipation may differ from the known electrical heat dissipation. An attempt should always be made to eliminate errors exceeding 20 per cent by re-examination of the assumptions for significant dimensions, characteristic configuration, emissivities, and representative case temperature. In particular, the latter variable may yield reductions in error if it is re-evaluated by means of combining known surface temperatures in a different manner which, however, must still be reasonably representative. In that respect, the analyst's experience and judgment must be relied upon. Finally, adjusted values of calculated and measured heat dissipation are compared. From their ratio, a correction factor F_c is obtained which can be assumed to remain applicable to all heat dissipation calculations for the particular unit under all operational conditions. The use of the correction factor and its determination are illustrated in Examples VI-1 and -2.

It should be noted that a value of F_c which is between 1.1 and 1.2 and cannot be reduced by revision of the assumptions, as indicated above, may point to other than free convection conditions during the bench test. In that event, careful examination of precautions taken to avoid the influence of extraneous air currents is necessary. The correction factor as obtained, should only be accepted if it is ascertained that all feasible provisions were made to insure free convection conditions during the test.

4. Variation of Component Temperatures

In pressurized equipment, the internal modes of heat transfer are not changed with the operational conditions. Therefore, the case temperature is the only variable on which component temperatures depend. The interrelation between component temperature and case temperature is discussed on page 6. It is pointed out there that components of unappreciable temperature rise, i.e., those which are mostly cooled by conduction and convection, would maintain a practically constant temperature difference with the case, as the case temperature changes with operational conditions. However, high-temperature components would probably have a decreasing temperature rise above the case temperature with increasing case temperature, since an appreciable portion of their heat dissipation may occur by radiation.

Thus, the determination of the representative case temperature at specified operational conditions permits an estimate of individual component temperatures by two means. If only one bench test was performed, the temperature difference between each component and the case obtained in this bench test must also be used at other operational conditions to estimate the component temperature from a calculated case temperature. This procedure would yield somewhat higher than actual surface temperatures for components which would be expected to be cooled to an appreciable extent by radiation to the case. More accurate estimates of component temperatures would require their measurement in several bench tests with different case temperatures, according to procedures outlined in Chapter IV, page 55. With the aid of these data, more precise values can be established for each component's temperature under variable operational conditions since a curve of component temperature versus case temperature would be available for each critical component.

5. Examples

Three examples are presented in this section to illustrate how by use of the charts in Figures VI-2 and -3 and Table VI-1, the heat dissipation from an equipment case by free convection and radiation may be calculated. In Example VI-1, methods of analyzing bench test data to establish a representative correction factor F_c are illustrated. In Example VI-2, methods are presented for computing case temperatures under operational conditions different from bench test conditions. In Example VI-3, methods of determining free convective and radiant heat transfer, as well as a representative case temperature when surface area subdivision is required are illustrated.

Example VI-1. Computation of Heat Transfer from a Pressurized Case by Free Convection and Radiation, Using Bench Test Data

The purpose of this example is to illustrate the use of Figures VI-2 and -3 and Table VI-1 in computing free convective and radiant heat transfer from a pressurized unit. The calculated results are used to define the test correction factor, discussed on page 138.

The case of the unit is a horizontal cylinder having a length of 14 inches, a diameter of 12 inches, and closed ends. An internal blower is used, primarily for cooling a high-output tube. Since the air agitation created by the internal blower gives fairly uniform case and internal air temperatures, the representative case temperature is assumed equal to the arithmetic average of the various measured case surface temperatures.

The following bench test data are reported:

Temperature of room air, wall, and screens surrounding test area, t_o and t_w, 30°C

Barometric pressure, p_{bar}, 28.9 inches mercury

Radiation environment (for use with Table VI-1):

surface of case,	black, painted
surface temperature of radiation receiver, t_w	30°C
confinement,	large room, painted

Heat dissipation (including internal motor), measured by difference of electrical input and output,	122 watts
Representative case temperature (obtained from arithmetic average of 14 measurements at various locations on surface of case), t_c,	55°C

Typical component temperatures and temperature differences at hot spots, limiting values determined from manufacturers' data and tests,

Component	Hot spot, °C		Difference, °C	
	Measured	Limiting	Limiting to Measured	Component to Case
High-output tube	128	260	132	73
Blower-cooled tube	96	150	54	41
Blower motor	80	120	40	25
Resistor	77	200	123	22
Transformer	73	150	77	18

Heat Transfer Areas

Configuration of case, horizontal cylinder with closed ends, length 14 inches, diameter 12 inches.

Effective area for radiation (two ends plus cylindrical surface)

$$2\,(\pi D^2/4) + \pi DL = 2(\pi \times 144/4) + \pi \times 12 \times 14 = 754 \text{ square inches}$$

Effective area for free convection,

ends, vertical planes,	226 square inches
shell, horizontal cylinder,	528 square inches

Radiation from Case

From Table VI-1, for radiation environment of black-painted case in large room, radiation factor $\phi_1 = 0.95$

From Figure VI-5b, at ordinate for case temperature of 55°C, read across to receiving surface temperature of 30°C, and down to radiation factor	$\phi_2 = 0.12$
Radiant heat transfer per unit area,	$\phi_1 \times \phi_2 = 0.114$ watt per square inch
Total radiant heat transfer, determined from heat transfer per unit area and the effective radiation area, calculated	86 watts

Free Convection from Case

Significant dimensions of parts,	
horizontal cylindrical portion of case	L_{hz} = 12 inches
vertical ends of case, $L_{vt} = 0.785 \times D$,	L_{vt} = 9.42 inches
From Figure VI-2b in upper right quadrant at ordinate of 55°C surface temperature, read across to air temperature of 30°C, down to pressure line of 28.9 inches mercury, over to line for horizontal cylinder in lower left quadrant, up to L = 12 inches in upper left quadrant, and across to value of unit heat dissipation for horizontal cylindrical surface,	0.054 watt per square inch
Total heat transfer from cylindrical surface by free convection, (0.054 x 528)	28.5 watts
From Figure VI-2b, using same procedure as for horizontal cylinder, determine unit heat transfer from vertical ends for L = 9.4 inches,	0.071 watt per square inch
Total heat transfer from vertical ends by free convection, (0.071 x 226)	16.0 watts

Summary of Computed Heat Dissipation from Case

Radiation	86.0 watts
Free convection	
horizontal cylindrical surface	28.5 watts
vertical ends	16.0 watts
Total calculated heat dissipation	130.5 watts
Total measured heat dissipation	122 watts

Correction Factor

Since the computed total heat dissipation does not agree with the electrically measured heat dissipation, a correction factor F_c, discussed on page 138, must be applied to all radiation and free convection heat transfer computations. For this bench test,

$$F_c = 122/130.5 = 0.935$$

Summary of Corrected Heat Dissipation from Case

The corrected heat dissipations are calculated to determine the distribution among the two modes of heat transfer. They equal the computed individual values of heat dissipation multiplied by the correction factor F_c.

Radiation	80.3 watts
Free convection	
horizontal cylindrical surface	26.7 watts
vertical ends	15.0 watts
Total corrected heat dissipation	122.0 watts

The percentage distribution among the two modes is:

radiation,	65.9 per cent
free convection,	34.1 per cent

Example VI-2. Determination of Component Temperatures under Operational Conditions on Basis of Bench Test Data

It is required to determine the case and component temperatures of the unit described in Example VI-1, operating under the following conditions:

Compartment,	
equivalent altitude,	50,000 feet
air temperature,	30°C

Radiation environment,
radiation from entire surface of black-painted case,
receiver surface of unpolished Dural at 30°C,
close confinement in small installation compartment.

In this example, the conditions for radiant and free convective heat transfer from the case are very much inferior to those of Example VI-1, since the free convective heat transfer is hindered by the low absolute air pressure within the compartment and the radiation-receiving compartment wall is considerably more reflective. Hence, it is difficult to predict the distribution of heat dissipation between radiation and free convection, on basis of the values obtained in Example VI-1.

As a first step of the procedure, it may be assumed that the heat dissipation from the case by radiation is the same as under bench test conditions. The radiation per unit area of case surface would be

$$(80.3)/(754 \times 0.935) = 0.114 \text{ watt per square inch},$$

since it is assumed that the same correction factor F_c evaluated on basis of the bench test data in Example VI-1 is applicable to this condition.

Estimate of Case Temperature

From Table VI-1, for radiation environment of black-painted case surrounded by Dural surfaces in close confinement, radiation factor	$\phi_1 = 0.30$
The value of the case surface temperature may be estimated if the radiation factor ϕ_2 is known. $\phi_2 = 0.114/\phi_1$ Then by use of Figure VI-3b, opposite to the manner indicated for Example VI-1, the estimated case surface temperature is found as	96°C

Convection from Case

Significant dimensions, same as in Example VI-1	
From Figure VI-2a and -2b for altitude of 50,000 feet, estimated surface temperature of 96°C, air temperature of 30°C, and L = 12 inches, the unit heat dissipation for the horizontal cylindrical portion of the case is	0.060 watt per square inch
Total heat transfer from cylindrical surface by free convection, (528 x 0.060)	31.7 watts
From Figure VI-2b, using same procedure, unit heat dissipation for vertical ends, for L = 9.4 inches,	0.082 watt per square inch
Total heat transfer from vertical ends by free convection, (0.082 x 226)	18.5 watts
Total estimated heat dissipation by free convection, uncorrected, (31.7 + 18.5)	50.2 watts
Corrected heat dissipation by free convection, (0.935 x 50.2)	46.9 watts
Total heat dissipation by radiation and free convection, for case temperature of 96°C, (80.3 + 46.9)	127.2 watts
Percentage of cooling by radiation, (80.3/127.2)100	63.1 per cent

Using the above percentage of the total known heat dissipation of 122 watts, the radiant heat transfer is 77.0 watts. Then, for the conditions given, $\phi_2 = 77.0/(0.935 \times 754 \times 0.3)$ — $\phi_2 = 0.364$

From Figure VI-3b, for $\phi_2 = 0.364$, the case temperature is — 94°C

Convection from Case

For surface temperature of 94°C, and air temperature of 30°C,

Cylindrical portion, (0.057 x 528) — 30.1 watts

Vertical ends, (0.081 x 226) — 18.3 watts

Total calculated heat dissipation by free convection, uncorrected, — 48.4 watts

Corrected heat dissipation by free convection, (0.935 x 48.4) — 45.2 watts

Heat Balance

Radiation	77.0 watts
Free convection	45.2 watts
Total calculated heat dissipation	122.2 watts
Measured heat dissipation	122 watts

This agreement between the calculated heat dissipation and the electrically measured value, based on the difference between input and output, is sufficiently close. An additional trial would lower the case surface temperature only about 0.2°C. Hence, the case temperature of 94°C is essentially correct. In general, Table VI-1 and the working charts of Figures VI-2 and VI-3 cannot be used to an accuracy greater than plus or minus 2 or 3 per cent, so that another trial for computing the case temperature is not necessary whenever the computed heat dissipation agrees with the measured heat dissipation within these limits.

For this pressurized unit, the component temperatures and the internal air temperature may be conservatively assumed to follow directly the case temperature, so that the component temperatures may be computed from the bench test component temperatures given in Example VI-1 simply by adding the increase in case temperature. Hence, the increase in component temperatures for this example is (94-55) = 39°C. If trend curves of component surface temperature versus case temperature were available for the various components, which is preferable, a particular component temperature would be obtained from the curve at a value of case temperature of 94°C. Here, the component temperatures are taken to increase 39°C, and by use of the component temperatures obtained in the bench tests of Example VI-1, the following values are obtained:

Component	Hot spot, °C Computed	Limiting	Difference, °C Limiting to Computed
High-output tube	167	260	93
Blower-cooled tube	135	150	15
Blower motor	119	120	1
Resistor	116	200	84
Transformer	112	150	38

Among the above temperatures, it is quite certain that the highest is over-estimated more than the lowest, while all are likely to be conservative. The one of the five components, having the lowest percentage of heat dissipation contributed by direct radiation to the case, would have a surface temperature most accurately estimated by the above temperature.

Example VI-3. Determination of Heat Dissipation from Case Surface having Non-Uniform Temperature

Figure VI-4 indicates the magnitude and location of the various surface temperatures measured on a side panel of an equipment case during bench test. This single panel of the entire case is used to illustrate the methods employed for evaluation of free convective and radiant heat transfer from a surface having a non-uniform temperature distribution.

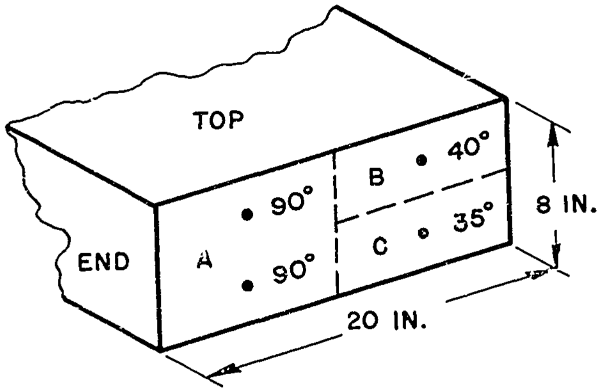

Figure VI-4. Location and Magnitude of Temperatures Measured on a Side Panel of an Equipment Case (Example VI-3)

The equipment operates under the following environmental conditions:

Compartment
- equivalent altitude, sea level
- air temperature, 30°C

Radiation Environment
- surface of case, black painted in large room, painted surfaces at 30°C

Total heat dissipation, from electrical measurements,	150 watts

Calculation of Radiant Heat Transfer

For the distribution of measured surface temperatures on the panel, the entire area is subdivided into the three subsidiary areas A, B and C, having average temperatures of 90, 40 and 35°C, respectively. The total radiant heat transfer from the side panel to the heat-receiving compartment walls is equal to the sum of the heat dissipations from the subsidiary areas.

From Table VI-1, for black-painted case in a large room,	$\phi_1 = 0.95$
From Figure VI-3b, for area A	$\phi_2 = 0.332$
for area B	$\phi_2 = 0.043$
for area C	$\phi_2 = 0.020$
The radiant heat transfers are:	
for area A (0.95 x 0.332 x 10 x 8)	25.2 watts
for area B (0.95 x 0.043 x 10 x 4)	1.63 watts
for area C (0.95 x 0.020 x 10 x 4)	0.76 watts
Total radiant heat transfer from side panel,	27.59 watts

Calculation of Free Convective Heat Transfer

Convective air flow is vertically upward. Therefore, the panel is subdivided into two vertical strips, one corresponding to area A, the other being a combination of areas B and C.

For area A,	
average surface temperature,	90°C
significant dimension,	L = 8 inches
unit heat transfer rate, from Figure VI-2b	0.223 watt per square inch
free convective heat transfer, (0.223 x 10 x 8)	17.9 watts
For area B-C	
average surface temperature, (40 + 35)/2	37.5°C
significant dimension	L = 8 inches
unit heat transfer rate, from Figure VI-2b	0.022 watt per square inch
free convective heat transfer, (0.022 x 10 x 8)	1.78 watts
Total heat dissipation from panel by free convection (17.9 + 1.78)	19.68 watts

Total Heat Transfer from Panel

The total heat transfer is the sum of the total radiant and convective heat dissipations, (27.59 + 19.68)	47.27 watts
Percentage radiation, (27.59/47.27)100	58 per cent

This procedure must be repeated for any other panel having a temperature variation in excess of the qualitative limits described on page 133.

Representative Case Surface Temperature

If it is assumed that this panel is typical of the entire case, a representative case surface temperature, as discussed on page 133, may be calculated.

For the total radiant heat transfer of 27.59 watts, the entire surface of the panel of 160 square inches, and $\phi_1 = 0.95$, $\phi_2 = 27.59/(0.95 \times 160)$	$\phi_2 = 0.181$
From Figure VI-3b for the above value of ϕ_2 and the environmental temperature of 30°, the representative case temperature is	66.5°C

If the average or weighted average surface temperature of 63.8°C is used as representative surface temperature for the entire panel, the calculated heat dissipation is 8 per cent smaller than that obtained by the above-described method of area subdivision.

Pressurized and Sealed Units with Case Cooled by Forced Convection

This general classification of units includes all those for which the case is the principal external heat dissipating surface. The two major types among these units are described in Chapter II, pages 7 and 8. They are (1) those having a plain case surrounded by a circumferential baffle which forms with the case an air flow passage (this includes the simple heat exchanger type shown in Figure II-1), and (2) those having a case-envelope heat exchanger with surface extensions between the case proper and the outer shell of the unit, similar to the type shown in Figure II-2. The heat dissipation of such units occurs mainly by forced convection to air which is either induced or forced to flow, usually by means of a blower, through the heat transfer passages which are in direct thermal contact with the inner case of the unit.

1. Heat Transfer Processes

The heat transfer processes of units with plain case and circumferential baffle are shown schematically in Figure VI-5. As indicated in Figure VI-5, the heat transfer by forced convection to the cooling air in the passage is practically equal to the total heat dissipation of the unit.

The cooling air not only receives heat from the case directly, but also from the inner surface of the external baffle. If the baffle is externally insulated, like in the bench tests suggested in Chapter IV, the heat it dissipates to the cooling air originates at the case surface and is conveyed to the baffle by radiation and by conduction through structural members. In that event, the temperature of the baffle is defined by the case temperature, the cooling air rate and the cooling air temperature. Therefore, the heat transfer characteristics of the entire unit would depend only on these three variables. However, if the baffle is not insulated, it may dissipate to the environment a portion of the heat received from the case, or it may receive heat from the environment if the environment is at a much higher temperature level than the cooling air. In the first instance the cooling air would not account for the entire heat dissipation of the unit, while in the second instance it would absorb more heat than is generated by the unit.

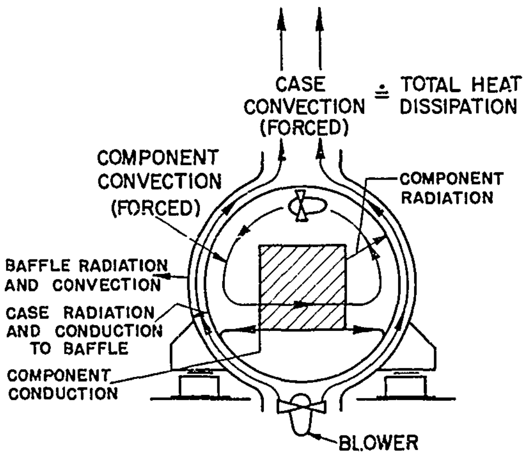

Figure VI-5. Heat Transfer Diagram of Pressurized or Sealed Unit Cooled by Forced Convection

When the environment of the unit is essentially at the temperature of the cooling air and free convective conditions exist in the vicinity of the unit, the heat transfer from the baffle to the environment can be expected to be 10 to 30 per cent of the heat transfer from the case to the baffle, the remainder being transferred from the baffle to the cooling air. Thus, the cooling air would usually account for 91 to 98 per cent of the total heat dissipation of the unit since the heat transfer from the case to the baffle should not be expected to exceed 20 to 30 per cent of the total heat generation. On the other hand, in environments where heat transfer from the baffle would be promoted by high air circulation rates and adjacent surfaces having high emissivity and temperatures more than 50°C lower, the heat loss may be greater than the maximum of 9 per cent implied above. Similarly, high environmental air and surface temperatures would result in heat gain of the baffle and the cooling air may have to account for a heat absorption rate considerably higher than the heat generation of the unit.

In this type unit the internal modes of heat transfer from components to case are seen in Figure VI-5 to be similar to those of the pressurized and sealed units cooled only by free convection and radiation, except that, as a rule, random forced convection produced by means of a circulating blower exists at the component surfaces. Thus the internal heat transfer mechanism remains fixed and is only affected by the case temperature. Therefore, component surface temperatures can be related to the case temperature by an approximately constant difference, or more correctly by trend curves established from measurements made at thermal equilibrium and at different case temperatures.

Since the case temperature is the principal parameter defining the thermal state of the unit and its components, evaluation methods based on bench test data must be principally concerned with prediction of the case temperature under specified operational conditions. In environments in which heat loss from the external baffle may occur to the extent of less than 10 per cent of the heat generation of the unit, as discussed above, insulating the baffle would affect the case temperature usually by no more than 2° or 3°C, if the unit has a heat concentration of 300 watts per cubic foot, or more, and has therefore, an appreciable air velocity in the cooling passage. Thus, neglecting baffle heat loss in the evaluation procedure would lead to conservative results. Case temperatures for specified air rates and required air rates for specified case temperatures would be slightly overestimated. This would be a satisfactory procedure to follow in all instances where the operating environment of the unit is only specified by air temperature and pressure, essentially at the same level as the cooling air. The methods given in this Chapter take this approach since they could be applicable in most instances where evaluations of operational conditions are to be made on the basis of bench test data. For environments in which appreciable baffle heat loss or gain may be expected, i.e., when environmental surface temperatures are 50° to 100°C, lower, respectively higher, than the air temperature, the methods described in Chapter VII for the determination of the heat transfer characteristics and for the evaluation of operational temperatures of this type unit should be used. As will be apparent from comparison of the methods given in the following with those of Chapter VII, neglecting external heat loss or gain may be expected, i.e., when environmental surface temperatures are between 50° to 100°C lower or higher than the air temperature, Also, the heat transfer characteristics of the unit can be expressed and used in equation form which permits reasonably reliable application of the bench test data in extrapolated ranges of case temperature, higher and lower than obtained in the tests.

The same considerations described above in reference to units with circumferential baffle also apply to those with simple case-envelope heat exchanger, as shown in Figure II-1. The only difference between the units is in the internal heat transfer mechanism. The internal baffle shown in Figure II-1 would eliminate direct radiation from components to the case and, therefore, the use of trend curves of component versus case temperature would not be as necessary, since component temperatures are more likely to maintain a constant difference with the case temperature. This would also be true for units with case-envelope heat exchangers of the type shown in Figure II-2 where also direct radiation from components to the case do not occur. In this latter type unit, direct conductive thermal contact exists between the case and the unit's outer shell. The heat transfer surfaces exposed to the external cooling air flow are at an average temperature lower than the case, but are many times greater than the case surface. The outer shell receives heat from the case mainly by conduction and dissipates it in the same manner as the outer baffle of the plain unit, as discussed previously. Consequently the same considerations for neglecting external heat loss or gain apply. The outer shell may operate at a higher temperature than the outer baffle of a comparable plain unit and may, therefore, have more heat loss. However, this heat loss would represent smaller percentage of the total heat dissipation since a unit with an extended surface type case-envelope heat exchanger having the same over-all dimensions as a plain unit with circumferential baffle would normally have more than twice the total heat dissipation. If operational environmental conditions are such that surrounding surface temperatures 50° to 100°C lower or higher than the air temperature exist, external heat loss, respectively gain, must be considered, using for the determination of heat transfer characteristics and the operational analysis the methods described in Chapter VII and VIII for this type unit. Otherwise, the methods for determining operational case temperatures described in the following in reference to the plain unit with circumferential baffle are equally applicable to all units with case-envelope heat exchangers.

Two methods of analysis, based on bench test data and neglecting external heat loss or gain, are described in the following. In the first, the test data are reduced to express the heat transfer and resistance characteristics of the unit's air flow passage in two equations which may be applied to other conditions of air flow and temperature. In the second method, the heat transfer and resistance characteristics are expressed as generalized plots derived from the test data and applicable to all conditions of air flow and temperature within their respective ranges.

2. Determination and Application of Heat Transfer and Resistance Equations

a. Heat Transfer Equation

The rate of heat absorption q_a of the cooling air in the heat exchange passage, expressed in watts, is related to the air flow rate W in pounds per second, and the inlet and exit temperatures of the air, t_1 and t_2 in °C, by the equation

$$q_a = 456\, W(t_2 - t_1) \qquad \text{(VI-5)}$$

If the external heat loss is negligible, q_a equals the total heat transfer rate from the case q_c which is the unit's rate of heat generation q in general. Thus if q is constant under all operational conditions q_a also remains constant.

The forced convective heat transfer coefficient of the heat exchange passage, bounded by the case surface and the outer baffle or shell, may be expressed as

$$h_a = C_1 W^{(1-n)}/\alpha, \qquad \text{(VI-6)}$$

and the rate of heat absorption q_a by

$$q_a = C_2 h_a(t_2-t_1)/\log_e\left[(t_c-t_1)/(t_c-t_2)\right], \qquad \text{(VI-7)}$$

where C_1, C_2 and exponent n are constants, t_c is the case temperature in ^oC, and α is a function of the average cooling air temperature, $t_a = (t_1+t_2)/2$, defining the relative variation of physical properties of air with temperature. The function α is based on a reference temperature t_{a-r} for which $\alpha = 1$. Combining equations (VI-5, -6, and -7) gives, if q is constant and no external heat loss or gain occurs, the general correlation equation

$$\alpha W^n \log_e\left[(t_c-t_1)/(t_c-t_2)\right] = \text{constant} = K \qquad \text{(VI-8)}$$

The choice of the reference temperature t_{a-r} is arbitrary. It fixes the value of α for every other condition characterized by an average air temperature t_a. Based on the variation of physical properties of air with temperature, grouped in the combination in which they affect the forced convective heat transfer coefficient, gives for the temperature range applicable to electronic equipment the relationship

$$\alpha = (1170+t_{a-r})/(1170+t_a), \qquad \text{(VI-9)}$$

where both temperatures are in ^oC.

Since the representative case surface temperature t_c is directly related to component surface temperatures, once having chosen t_{a-r} and established n and K from bench test data, the component temperatures and required air rates of the blower may be computed by use of equations (VI-8 and -9) for any operational condition of the equipment.

b. Bench Test Determinations

The bench test determination of temperatures, air rate and constants of equations (VI-8 and -9) is discussed in the following paragraphs.

(1) Air Entrance and Exit Temperature, t_1 and t_2. The methods outlined in Chapter IV, page 53, should be followed when determining bench test values of the inlet and exit temperatures of the cooling air, t_1 and t_2. When it is known that non-uniform flow conditions exist at entrance and exit, the use of a single temperature measurement at some point in a transverse section of the stream will not be satisfactory and the method of weighting the temperature on the basis of flow rate and temperature of subsidiary areas must be employed. For many units having forced, rather than induced flow, through the cooling passage, it is not feasible to probe the stream at the cooling air entrance because of the proximity of the blower to the equipment case. Then, the inlet temperature t_1 is best determined by measuring the temperature of the air entering the blower and adding to it the temperature rise of the air across the blower as calculated by equation (V-2 or -3), page 86. If the total heat dissipation from the equipment case to the cool-

ing air is known from electrical measurements of heat dissipation, it may not always be necessary or desirable to measure the cooling air exit temperature t_2 since it can readily be calculated from equation (VI-5), assuming no heat loss or gain, once the air rate has been established. In particular, when the flow conditions at the equipment exit are highly non-uniform with respect to available transverse measuring sections and considerable difficulty in measuring a true mean exit temperature is encountered, a better value may well be obtained by use of equation (VI-5). In general, however, it is desirable to have a measured value so that a check on the heat output of the components is obtained by calculating the heat dissipated to the cooling air. With bench test programs not including the electrical measurement of heat dissipation, the measurement of the exit temperature of the air is, of course, always necessary. The difficulty in measuring a true mean temperature of the entering air is not nearly so great. For induced flow, the entering air temperature is equal to that of the environment. For forced flow, the air temperature ahead of the blower, being equal to that of the environment, may be measured accurately, and little error in determining the blower discharge temperature will be introduced by use of equation (V-2 or -3). As a matter of fact, in many instances the air temperature rise across the blower is negligible.

(2) Representative Case or Heat Exchanger Surface Temperature, t_c. The methods and suggested number of temperature measurements on the case surface are outlined in Chapter IV, page 52. It is always desirable with forced convective cooling to make as many measurements as possible so that surface hot-spot effects may be included in the evaluation of a representative case surface temperature. Most equipments will exhibit fairly uniform case surface temperature and the evaluation of the representative case surface temperature is obtained simply from an arithmetic average of all measured values. Even for equipments exhibiting appreciable variation in temperature over the case or heat exchanger surface, the use of an arithmetic average of the measured values still should result in a fairly satisfactory correlation of the test data by equation (VI-8). However, in some instances an adjustment of the value of the representative case surface temperature t_c may be required in order that equation (VI-8) may satisfactorily describe the interrelation of the heat transfer variables. No general rules on procedures to be used for making these adjustments are possible to set down. In general, the exclusion of measured surface temperatures below the bulk air temperature at the same point along the flow path should aid in improving the correlation since at these points heat transfer from the case to the air stream is not possible. If adjustment of the case surface temperatures is found necessary, attention must be directed to re-examination of the correlation between representative case surface temperature and component temperature, so as to provide correct trend curves for use in predicting cooling requirements under all probable operational or environmental conditions of the equipment.

(3) Air Rate, W. The air rate W used in the correlation of bench test data must be expressed on a weight rate basis. Its measurement should be made by the procedures indicated on page 53. If known or measured on a volume rate basis, conversion to weight rate may be accomplished quite simply by use of Figure V-9 or equation (V-1).

(4) Physical Property Factor α. As previously mentioned the factor α indicates relative changes in physical properties of air with temperature, as they affect the heat transfer process. At the reference temperature t_{a-r} it is equal to unity and varies for other average air temperatures t_a as indicated by equation (VI-9). While the reference temperature t_{a-r} may be chosen arbitrarily, it is most convenient to make it equal to the average air temperature in the most typical bench test. In this test the air rate would be approximately at the average of the range of flow rates used.

(5) Constants n and K. Evaluation of n and K in the general correlation equation (VI-8) requires the use of bench test data, from which numerical values of all other variables in equation (VI-8) are obtained. The numerical value of n may be determined by measuring the slope of the line representing a log-log plot of $\alpha \log_e [(t_c-t_1)/(t_c-t_2)]$ versus the air rate W for all test runs. A graphical method is outlined in a subsequent section beginning on page 158 where the use of generalized working charts for handling the correlation equation (VI-8) is discussed. Example VI-4 contains additional information on procedures used in determining the exponent n. Having established a representative value of n, the determination of the constant K may be made by direct calculation from equation (VI-8) or by use of the working chart in Figure VI-7, described on page 158.

In order to establish reliable values of n and K the log-log plot of the test points should yield, with minor scatter, a straight line. If there is a definite indication of curvature, a mean straight line may be chosen to express the range of the test data, but must not be extrapolated in applying the data. However, in the case of curvature it is preferable to use the alternate method of data reduction on basis of generalized plots, described subsequently on page 156.

c. Flow Resistance of Heat Exchange Passage

Normally, cooling air flow is produced by a blower having characteristics such that any change in the resistance of the air flow passage affects its air handling capacity. Since the operating temperature of the equipment is directly related to any change in air flow, it is necessary to correlate bench test data on air pressure drop in such manner that operational thermal conditions at other than bench test conditions may be predicted.

Within the altitude and flow velocity ranges most likely to be encountered with a forced convective cooling system, a satisfactory correlation between system pressure drop and weight rate of air flow is obtained by use of the general equation

$$\sigma_1 \Delta p = C W^m, \qquad \text{(VI-10)}$$

where Δp is the static pressure drop of the cooling air from cooling passage entrance to exit, normally expressed in inches of water, σ_1 is the air density ratio at the cooling passage entrance, W is the weight rate of air flow produced by the blower in pounds per second, and m and C constants evaluated from bench-test data.

As a result of any change in operation of the blower, the equipment, or the aircraft, the air flow rate produced by the blower will adjust itself until the static pressure at the exit of the air discharge duct is equal to the environmental air pressure. For example, for a unit receiving air from and discharging air to the installation compartment, under all equilibrium operating conditions the air is supplied to and is discharged from the unit at the compartment air pressure, so that the pressure production required by the blower is that necessary to overcome the resistance of the air flow passage.

For purposes of correlating air flow and system resistance by use of the general equation (VI-10), care must be taken that the pressure drops measured in the bench tests correspond to those of a comparable flow system in an actual installation. The safest procedure to follow in this respect is to measure the static pressure of the cooling air when forced flow is used at the blower discharge section, or at the blower inlet section when induced flow is used. The location of the blower relative to the equipment would be exactly the same as used in the actual installation. This is to insure that the resistance of any air passage located between the blower and the cooling passage is included in the evaluation of the air flow system resistance against which the blower must operate. If, for any reason, it is not feasible to simulate actual installation configurations during bench testing, the proper procedure would be to measure the static pressure at the cooling passage and to correct later the bench test pressure drops to include the effect of any actual duct length between blower and equipment. Otherwise, of course, the resulting correlation would not be descriptive of the air flow system resistance of the actual installation. If the blower characteristics and the test air rates are accurately known, fairly reliable values for the air flow system resistance may be determined indirectly from these data, since the blower's operating characteristic is defined by static pressure rise versus air flow rate. It is not recommended that this procedure be followed for general test work even if the blower's performance is known, but it may be used when difficulties in pressure measurement preclude the collection of accurate experimental data.

Methods for actual measurement of static pressure and air rate should follow those discussed on pages 35 to 44. No additional special requirements or provisions to the indicated techniques are necessary. The values of the constants m and C are determined by plotting the bench test values of pressure drop versus air rate on log-log coordinate paper. In Example VI-4, page 164, their determination is illustrated in conjunction with the description of the use of a generalized working chart designed for this purpose. As for the determination of n in the forced convection heat transfer equation, a single value of m should satisfactorily correlate the pressure drop data with air rate, i.e., the test data should yield a straight-line log-log plot. Should this not be the case, a mean straight line may be chosen to express the resistance characteristics in the limited range of the test data. However, in general it would then be preferable to use the alternate method of applying directly the curve of $\sigma_1 \Delta p$ versus W, without attempting to evaluate m and C, as described subsequently on page 157.

d. Application of Equations

In the determination of operational conditions from bench test data, equations (VI-5, -8, and -10) can be used to predict the case temperatures and air flow rates resulting from the application of a blower, or any other air source, of known characteristics. Also, for any specified case temperature, the required air flow rate and static pressure can be ascertained for any condition of pressure and temperature at the source of air supply. In all instances, external heat loss or gain would be neglected.

As pointed out previously, the equation method is principally applicable when the test data yield straight-line log-log plots for heat transfer and resistance characteristics of the unit. Then, the use of the equations for considerably higher air weight flow rates than covered in the bench tests is permissible and would yield accurate results. Use of the equations for much lower weight flow rates than the minimum used in the tests should be made with caution since the characteristics of the flow are likely to change, thus making the equations established from the test data invalid.

For the determination of available flow rates, using a blower of known characteristics to produce induced or forced flow, the procedures described in Chapter V are applicable. Blower characteristics, system resistance characteristics defined by equation (VI-10), and the known heat dissipation rate $q = q_c$ are used to determine the available air flow rate under any environmental condition.

For the determination of blower specifications using induced flow, the required weight flow rates can be determined directly from the heat transfer characteristics defined by equations (VI-5 and -8). However, using forced flow it is necessary to use equation (VI-10) first to determine for a range of weight flows the pressure drops Δp, and to calculate for an assumed static efficiency η_s (on the order of 50 to 60 per cent) the probable corresponding values of temperature rise across the blower Δt_B from equation (V-3). This establishes for any given environmental temperature t_o the equipment inlet temperature $t_1 = t_o + \Delta t_B$ and its variation as a function of the weight flow W.

The solution of equation (VI-10) to determine Δp or W, the other being known, can also be performed graphically by the use of a working chart such as given in Figure VI-9, described in a subsequent section, page 162.

The equations describing the heat transfer characteristics can, as previously mentioned, also be solved conveniently by the use of working charts. However, the temperature rise of the case above the inlet air temperature can be determined by calculation, using equations (VI-5 and -8) in the combined form

$$(t_c - t_1) = (q/456\ W)\Big/\left[1 - 1/e^{K/\alpha W^n}\right] \qquad \text{(VI-11)}$$

Equation (VI-11) is best used for known values of weight flow. Using a blower of known characteristics, the values of W and t are defined from equation (VI-10) and the flow arrangement, as indicated above. The discharge temperature t_2, and with it the average air temperature t_a, is found from equation (VI-5), thus defining α according to equation (VI-9). Therefore, the case temperature rise (t_c-t_1) can be calculated directly. If the blower specifications are to be determined for a specified case temperature, the right-hand side of equation (VI-11) is best determined for several values of W to establish the variation of (t_c-t_1) with W. For induced flow, t_1 is known and defines, for any value of W, (t_2-t_1) from equation (VI-5) and thus the average air temperature t_a and α. Therefore, for any value of W the corresponding (t_c-t_1) can be calculated from equation (VI-11) and the case temperature t_c is defined, t_1 being invariable with W. For forced flow, the estimated variation of t_1 as a function of W must first be determined from the resistance characteristics, using equations (VI-10) and (V-3), as indicated above. Then the procedure is the same as for induced flow, defining (t_c-t_1) versus W, and from this t_c based on the probable variation of t_1 versus W. With any blower having at the operating point a static efficiency different from that assumed in establishing the variation of t_1 versus W, a slightly different case temperature t_c would be obtained. The effect would only be appreciable if the temperature rise across the blower Δt_B would be great because of a large value of $\Delta p/\sigma_1$ and/or if the efficiency of the blower would be very much lower than assumed. Consequently, it would usually be unnecessary to recalculate the case temperature.

3. Determination and Application of Generalized Heat Transfer and Resistance Plots

This second method of reducing the bench test data and applying them for the determination of operational thermal conditions represents a simplification of the equation method. It is applicable with good accuracy over the range of weight flow covered in the bench tests. The bench test data determinations discussed on pages 151 to 154 are also required for this method. However, instead of reducing the data so as to express heat transfer and resistance characteristics in the form of equations, the working curves as shown in Figure VI-6 are obtained which may be applied directly to the evaluation of operational conditions.

It is apparent from inspection of equation (VI-11) that when the effect of variation in physical properties due to temperature, expressed by α, is omitted, (an approximation introducing usually less than 2 per cent error in the required weight flow for a given case temperature), the case temperature rise (t_c-t_1) is only a function of the weight flow if q is constant. Therefore, the difference in values of the representative case temperature and of the inlet air temperature measured in the bench tests, plotted against the corresponding measured weight flow rates, gives a curve, such as B in Figure VI-6, equally valid under all operational conditions defined by air pressure and temperature. The curve would be valid even if K and n are not constant over the range of weight flow covered and no single values can be determined for them, using the previously described equation method. For the general application of the data, it is only assumed that K and n would depend principally on W, which is basically correct. Even then,

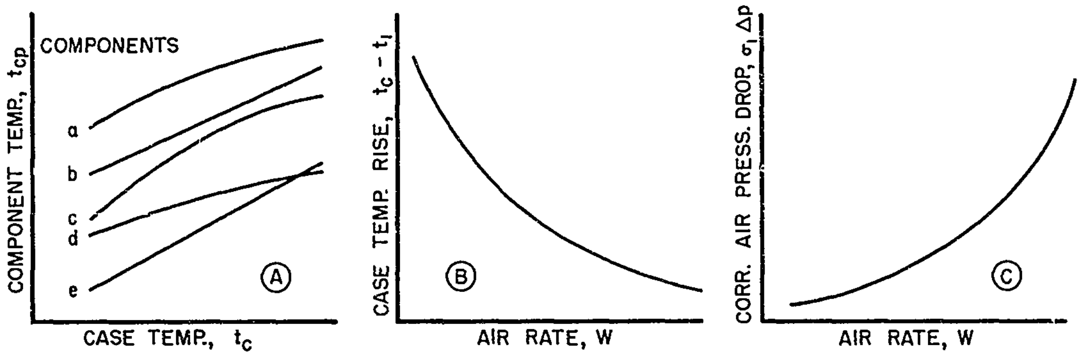

Figure VI-6. Working Curves for Pressurized or Sealed Unit with Case Cooled by Forced Convection

it is likely that the plot could be fitted for a limited range of weight flow by a simple equation having the form $t_c - t_1 = K'/W^{n'}$, where constants K' and n' differ from K and n, respectively.

Similar to the generalization of the heat transfer data, the pressure drop data can be reduced. Instead of attempting to find the values of C and m in equation (VI-10), the test data are plotted as $\sigma_1 \Delta p$ versus W, irrespective of whether C and m vary or are constant over the range of weight flows covered. This plot, such as C in Figure VI-6, can also be applied under all operational conditions within the test range of W. The actual pressure drop is determined by dividing the value of $\sigma_1 \Delta p$ determined for the given weight flow by the inlet density ratio σ_1, calculated from the operational values of air pressure and temperature.

The general application of the data plots should be limited to the weight flow range covered in the bench tests. Some extrapolation at the lower and upper ends of the curves for weight flow rates not used in the tests is permissible, but should be made with caution. If these limitations are observed, these plots are adequate to evaluate case temperatures under operational conditions with a minimum of complication and calculational effort. It is desirable to use in conjunction with these plots trend curves established in bench tests showing the variation of individual component temperatures with case temperature. Such data, as shown in plot A of Figure VI-6, are also desirable to determine the minimum flow rate to be used in the bench tests which should correspond to a case temperature at which limiting component temperatures are reached. For units of low heat concentration this flow rate may be so low that the accuracy of the measurements may be questionable. Therefore, the most accurate portions of the temperature rise and pressure drop curves established from bench tests would be utilized in evaluating operational conditions at which the air temperature is higher than during the bench tests.

For the evaluation of operational conditions where the air temperature is considerably lower than in the bench tests the use of the more complicated equation method is more reliable, although, as mentioned before, use of the equations at flow rates appreciably lower than the minimum used in the tests is not advisable. The equation method is much preferable for extrapolation of the data to appreciably higher weight flow rates than used in the tests. Such requirements exist when the available cooling air temperature under operational conditions is quite high in comparison to the desired case temperature which for existing units would usually require the selection of a new blower or a ram air supply system.

Because of the simple use of working plots, such as shown in Figure VI-6, Examples VI-4, -5 and -6 on page 164 dealing with units of the type here discussed, are concerned with illustrating the applications of the more complex equation method, using graphical means described in the following section.

4. Working Charts for Equation Method

To eliminate the necessity for repeated computations, in the applications of the equation method, as described on page 155, the working charts contained in Figures VI-7 and -9 (in back pocket) may be used. They are applicable to (1) the evaluation of bench test data to determine the constants and exponents of the correlation equations, (2) the evaluation of thermal performance under any operational condition of a unit employing a blower having defined performance characteristics, and (3) the evaluation of limiting operational conditions of the unit and the corresponding required blower performance for the purpose of selection and design of the blower and cooling air system. Uses of the charts are illustrated in Examples VI-4, -5, and -6.

a. Heat Transfer Chart

Figure VI-7 contains a multi-quadrant chart that allows graphical solution of the two basic heat dissipation equations (VI-5 and -8). For purposes of identification, the quadrants are numbered (1) through (6) and are described in the following paragraphs.

Quadrant (1). The ordinate to the left of this quadrant is the weight rate of air flow in pounds per second. The parameter plotted within the quadrant is the exponent n contained in equation (VI-8). A range of values is included which is likely to be encountered with this type of cooling. The procedure to be followed for determining n by use of this chart is discussed on page 159.

Quadrant (2). The parameter contained within this quadrant is the factor α used to account for the variation in physical properties of the air with temperature. Its value is determined by the average temperature of the air in the cooling passage, according to equation (VI-9). The numerical value of α for use in this quadrant is determined by the procedure subsequently discussed with regard to quadrant (6).

Quadrant (3). A range of values of the constant K in equation (VI-8) is plotted as the parameter in this quadrant. The numerical value of K would be expected to vary somewhat with the absolute size of the unit, configuration of cooling air passages, blower performance and bench test conditions, so that a single value of K is applicable only to one unit and one reference set of bench test conditions. Determination of the constant K should be made after the exponent n has been defined, and amounts to simply using the available test data and n to define a point in quadrant (3), or by calculation from equation (VI-8).

Quadrant (4). The difference between the representative case surface temperature and the air temperature at entrance to the cooling passage (t_c-t_1) is plotted on the right-hand ordinate with the air temperature rise (t_2-t_1) as the parameter. The air temperature rise (t_2-t_1) is indicated at the bottom of quadrant (5).

Quadrant (5). If external heat loss is negligible, for any known heat dissipation rate from the equipment and air flow rate over the cooling surfaces of the equipment, the air temperature rise (t_2-t_1) may be determined directly in quadrant (5) by reading horizontally to the left from the air rate scale to the intersection of a line describing the heat dissipation q in watts, and down to the temperature scale at the bottom of the quadrant. The value of (t_2-t_1) is the air temperature rise required for use in quadrant (4). The procedure may be reversed to define the air rate required for a given heat dissipation and temperature rise, or for the evaluation of the heat absorption of the air from known values of the air rate and air temperature rise. Quadrant (5) is a graphical solution to equation (VI-5).

Quadrant (6). By proceding along a vertical line defined by the air temperature rise (t_2-t_1), through quadrant (5) to the intersection with a line in quadrant (6) defined by the air temperature t_1 at the entrance of the cooling passage, then moving horizontally to the intersection of the line corresponding to the reference temperature t_{a-r}, and finally vertically to the top of quadrant (6), the numerical value of α required for use in quadrant (2) is obtained.

The procedure for determining n and K from bench-test data, by use of the generalized working chart in Figure VI-7, is illustrated schematically in Figure VI-8. The first step is to choose from the various air rates employed in the bench tests a typical average value, the rate being expressed in pounds per second. This air rate W is introduced on the ordinate scale of quadrants (1) and (5) in Figure VI-7, illustrated by point A in Figure VI-8. The second step consists in locating point B in quadrant (5). This point is defined by the air rate W and a known heat rate dissipated to the cooling air q, or by the air rate W and a known temperature rise of the air (t_2-t_1) measured in the bench test. Since measurements of air temperature rise are often not reliable, the electrically determined heat dissipation rate q to the cooling air would be preferable in all instances where the external heat loss from the outer surface of the unit is certain to be practically negligible.

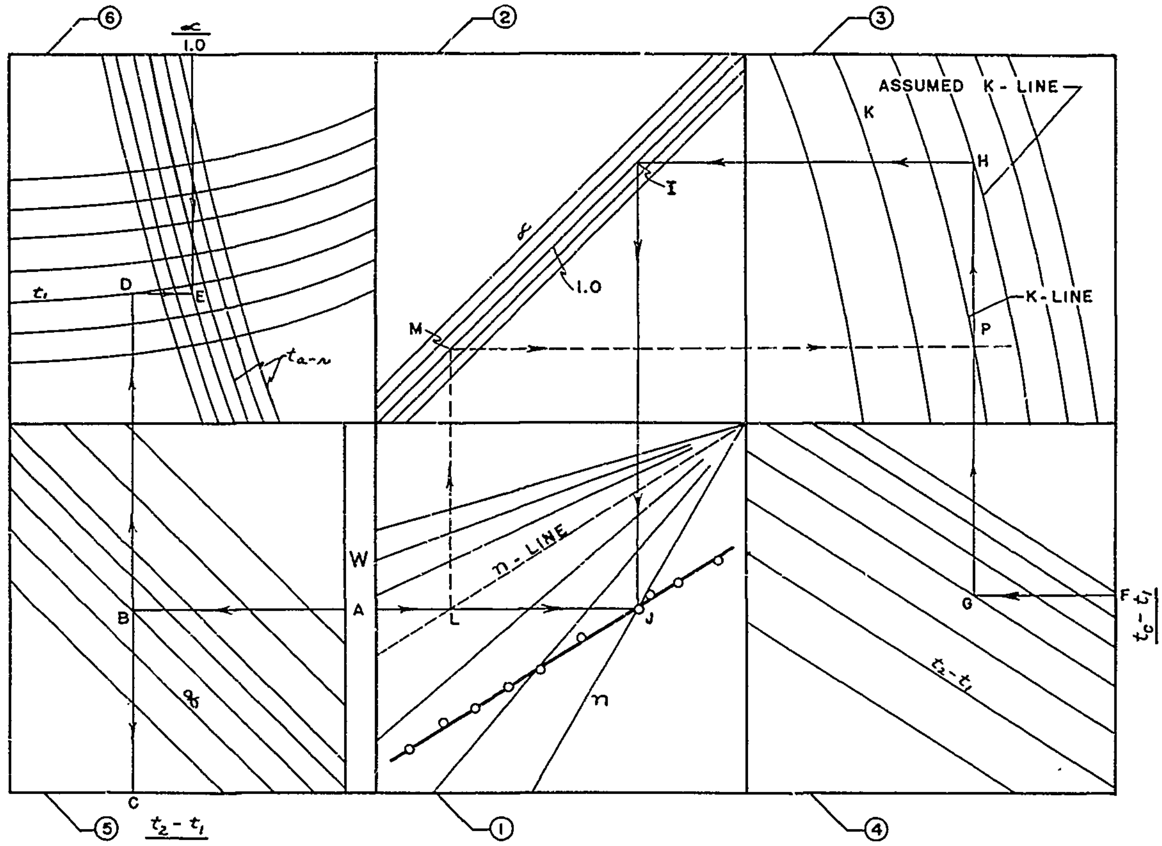

Figure VI-8. Schematic Diagram of Figure VI-7, Illustrating Determination of n and K

The next step is to define the reference average air temperature t_{a-r}. The factor α is taken equal to unity at the air rate defined by point A, since this air rate is a typical average of all bench-test air rates. From point B, a vertical line is drawn into quadrant (6) to point D, defined by the intersection with the line corresponding to the inlet air temperature t_1. From point D, a horizontal line is projected to the right and is intersected with a vertical line projected downward from α equal to unity. The intersection of these two lines, point E in Figure VI-8, defines the reference average air temperature t_{a-r}. This is the reference temperature used to evaluate K and to define α for any other air rate under all operating conditions.

Continuing the procedure for finding n and K, point F on the ordinate of quadrant (4) is located by the magnitude of the temperature rise of the heat exchange surfaces over the inlet air temperature (t_c-t_1), as defined by the test data at the air rate being considered. From point F, a horizontal line locates point G at the intersection with a line corresponding to (t_2-t_1), as defined by point C. From point G, a vertical line is drawn to point H at the intersection with the line corresponding to an assumed temporary value for K. Next, point I in quadrant (2) is determined by the intersection of the horizontal line from point H in quadrant (3) with the line corresponding to $\alpha = 1$ in quadrant (2). From point I, a vertical projection line is drawn to quadrant (1) and is made to intersect with a horizontal projection line from point A in quadrant (1). The point of intersection is J which defines the first point on a line representing the test data. From the preceding it is apparent that the location of J for the chosen test run depends on the choice of the assumed temporary value for K which is entirely at the discretion of the analyst. The best guide for the choice of a good temporary value for K is to make it such that point J, when found, should lie approximately in the horizontal third of quadrant (1) since then it is fairly certain that the other points corresponding to the other test runs would all fall within quadrant (1).

For all other air rates and associated test data the procedure is to start with the air rate W in quadrant (5) and find point B according to the same criteria as used in the first reference procedure. Then, for each air rate the value for α can be determined in quadrant (6) by the intersection of a horizontal line from the appropriate value of point D with the line for the reference air temperature t_{a-r} determined in the first reference procedure. The rest of the procedure for any test run is the same as for the reference run, using the same assumed temporary value for K and in quadrant (2) the value for α determined in quadrant (6). Thus, for each test run a corresponding point similarly to point J of the reference run, is determined in quadrant (1). An example of a series of test points so defined is shown on the heavy line in quadrant (1) of Figure VI-8. The numerical value of n is defined by that n-line exactly parallel to the line drawn through the test points. For the data illustrated in Figure VI-8, the dashed n-line runs parallel to the heavy solid line passing through the test points and, hence, has the correct value of n for the test data.

In the above procedure for determination of n, an assumed value for K was used temporarily. The correct value for K is determined after the value of n has been found. For that purpose, the reference test condition for which $\alpha = 1$ is used. The procedure is to start at point A in quadrant (1) and to determine point L at the intersection of a horizontal line from point A with the n-line which is parallel to the line representing the test points. The intersection of a vertical projection line from point L with a line for α equal to unity in quadrant (2) determines point M. From point M a horizontal line is projected into quadrant (3). This latter dashed line, as shown in Figure VI-8, is brought to intersection with the line between points G and H for the reference test run. Thus, point P is found in quadrant (3). The K-line passing through point P determines the value of K in the air flow equation representing the test data.

The values of n and K determined from the bench test data by the above procedure are applicable to the analysis of thermal performance of the cooling air passage under all operational conditions. For that purpose, the procedure is similar, except that the preliminary lines corresponding to those through points J and H in Figure VI-8 are disregarded, and instead the established n- and K-lines corresponding to those through point L and P in Figure VI-8 are used.

In using Figure VI-7 for the solution of thermal evaluation problems, the movement from one quadrant to another may be in either direction along the indicated line, except in quadrant (6), where movement in an upward direction only is necessary. For the type of problem requiring evaluation of the weight flow of cooling air for a specified operating temperature t_c of the case surfaces, the solution would involve successive assumptions of values of (t_2-t_1) and the use of the chart by starting in quadrant (4) and passing through quadrants (3), (2) and (1) to establish the corresponding weight flow W which should check the value of W determined in quadrant (5) by (t_2-t_1) and q. However, this procedure is inconvenient because the characteristics of the function expressed by the chart make the selection of suitable values of (t_2-t_1) by successive approximation difficult. Therefore, these problems are best solved by finding for various values of W corresponding values of (t_c-t_1) in quadrant (4). The remainder of the procedure would follow that discussed on page 155, depending in details on whether forced or induced flow is used.

It may occur that the numerical magnitudes of the various variables involved in the use of Figure VI-7 do not fall within the ranges provided on the chart. When this happens, the data may be brought within the ranges of the chart by adjusting the heat dissipation q, the air rate W, and the constant K, by use of an arbitrary factor ϕ. The relations correlating the values in the chart with the actual values are (1) $W_{(chart)} = \phi W_{(actual)}$, (2) $q_{(chart)} = \phi q_{(actual)}$, and (3) $K_{(chart)} = \phi^n K_{(actual)}$. For example, suppose $n = 0.4$ and $W = 0.04$. Then, as may be seen in quadrant (1) of Figure VI-7, the point on the n-line for $W = 0.04$ falls outside the quadrant and, therefore, the chart would not be useful. This may be corrected by choosing $\phi = 5$, so that with $n = 0.4$ the chart air rate is $5 \times 0.04 = 0.20$, which gives a point on the n-line in the middle of quadrant (1). For the chosen value of $\phi = 5$, the value of K to be used in the chart would be $(5^{0.4} = 1.7) \times K_{(actual)}$.

b. System Resistance Chart

A working chart for the correlation of bench-test pressure drop with air rate and the evaluation of system pressure drop for any operational condition is given in Figure VI-9. The chart permits graphical solution of the basic resistance equation (VI-10). The right-hand quadrant has air flow rate W in pounds per second as the abscissa and the exponent m in equation (VI-10) as a parameter. The left-hand quadrant has the density ratio σ_1 and the constant C contained in equation (VI-10) as parameters. System pressure drop in inches of water is plotted on the left-hand ordinate of this quadrant. The order of moving from one quadrant to another is indicated by the line shown on the chart. It may be in either direction, depending upon the type of problem.

The density ratio σ_1 is evaluated from the chart in Figure V-8, on basis of the air temperature and pressure at the entrance to the heat exchange passage. When correlating bench test data on system resistance, the exponent m in equation (VI-10) is obtained by plotting test points in the right-hand quadrant of Figure VI-9, using temporarily an arbitrarily assumed value for C. The procedure is illustrated in Figure VI-10 where a schematic diagram of Figure VI-9 is shown. For each test run σ_1 must first be deter-

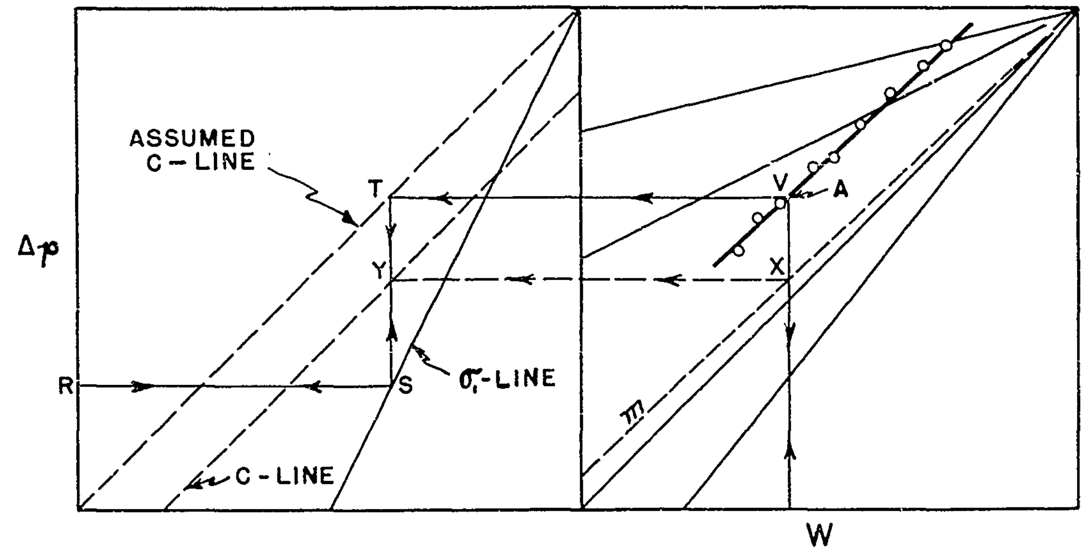

Figure VI-10. Schematic Diagram of Figure VI-9, Illustrating Determination of C and m

mined. Then, starting at point R for the measured pressure drop Δp, a horizontal line is drawn to point S on the σ_1-line for the given run. From point S a vertical line is drawn to point T at the intersection with a line for a temporarily assumed value for C. A horizontal projection from point T is made to intersect at point V a vertical projection line determined by the measured air rate W. Points similar to point V are established for other runs and determine the solid line shown in the right-hand quadrant of Figure VI-10. The numerical value of m is then defined by the m-line in the quadrant which is exactly parallel to the solid line representing the test points. This m-line is shown dashed in Figure VI-10.

After establishing the pertinent m-line, the correct C-line is found by intersecting a horizontal projection line from point X, determined by projecting point V down on the m-line, with the vertical projection line between points S and T. The point of intersection Y so established lies on the correct C-line. This line and the m-line must be used in all evaluation procedures to correlate air rate, pressure drop, and air density under all operational conditions.

For evaluation problems having defined conditions of cooling air at the inlet of the heat transfer passage, such as in bench tests and in the analysis of the air flow requirements of units with induced flow under operational conditions, the use of the chart is straightforward. However, when evaluating the air flow requirements of forced-flow units, or the performance of a combination of any type unit with a blower-motor unit of known characteristics, the magnitude of the pressure drop is not known until the inlet density and/or the flow rate are determined which, in turn, are functions of the pressure drop. Procedures for the solution of such problems are given in Chapter V. Additional details are contained in Examples VI-5 and -6 in the following section.

4. Examples

The examples given in this Section are concerned with the use of the charts in Figures VI-7 and -9, and applications of the method for which heat transfer and resistance characteristics obtained in bench tests are reduced to equations. Examples VI-5 and -6 involving the application of blowers are intended to review in part some of the principles outlined in Chapter V. In all examples, no reference is made to component temperatures since they are assumed to be functions of the case temperature only. Thus the last step in any analysis of operational conditions, not shown in the examples, would be to determine from trend curves, such as shown in plot A of Figure VI-6, the component temperatures corresponding to the operational case temperature. As an alternate, the less accurate method of constant differences between component and case temperatures can be utilized.

Example VI-4. Correlation of Bench Test Data by Use of Working Charts and Calculations

A pressurized unit having a case cooled by forced convection is bench-tested according to the recommended procedures described in Chapter IV. The unit is provided with a baffle concentric with the case. Through the annular passage so formed, cooling air is forced by the action of a blower-motor unit mounted ahead of the unit and the air-flow metering section. By throttling and bleeding of the blower to vary the air flow rate the test data of Table VI-2, in addition to the following, are obtained:

Room air temperature, all tests, t_o	21^oC
Barometric pressure, all tests, p_{bar}	29.4 inches mercury absolute
Air pressure at cooling passage exit, all tests,	zero gage
Electrically measured heat dissipation, all tests, q	400 watts

a. Heat Balance. It is assumed that all heat dissipated by the case is absorbed by the cooling air passing over the case surface. A reasonably good check on this may be obtained by comparing the measured air

Table VI-2. Test Data (Example VI-4)

Run No.	Air flow, W, pound per second	Air pressure, cooling passage entrance, Δp, inches water, gage	Air Temperatures, °C: Cooling passage entrance, t_1	Cooling passage exit, t_2	Case surface temperature,* t_c, °C
1	0.021	0.06	21.1	63.1	193
2	0.032	0.13	21.0	48.9	147
3	0.048	0.29	21.2	39.0	108
4	0.066	0.48	21.2	34.7	86.5
5	0.081	0.73	21.4	32.5	81.5
6	0.105	1.13	21.5	30.0	71.0
7	0.136	1.90	21.8	28.0	57.5
8	0.157	2.51	22.0	27.5	57
9	0.189	3.40	22.0	27.0	52.3
10	0.219	4.41	23.0	27.3	50.5

*Average of 8 thermocouple readings.

temperature rise (t_2-t_1) with that calculated from the energy equation (VI-5), or by use of quadrant (5) in Figure VI-7, since air flow rate and total case heat dissipation are known. The calculated comparison is shown in the table below. The deviation between the measured and calculated air temperature rise is shown to be within the limits of experimental accuracy for this type of heat balance and substantiates the assumption of negligible heat loss from the baffle surface by convection and radiation.

b. Correlation of Heat Transfer Data

The general correlation equation (VI-8) for heat transfer characteristics of this type of unit requires the evaluation of the exponent n and constant K. This may be done by use of the working chart in Figure VI-7, as discussed in detail on pages 159 to 162, or by direct plotting of the measured heat transfer data on log-log coordinate paper.

In using the chart in Figure VI-7, Run No. 6, having approximately the average value of the test air rates, is taken as a reference run. From quadrant (5), at the air rate $W = 0.105$ pound per second, and for $q = 400$ watts, the theoretical temperature rise $(t_2-t_1) = 8.5°C$.

Projecting upwards into quadrant (6) to the line corresponding

Table VI-3. Comparison of Calculated and Measured Air Temperature Rise (Example VI-4)

Run No.	Air rate, W, pounds per second	Air temperature rise, °C Measured	Calculated by equation (VI-5) or Figure VI-7	Difference, calculated minus measured, per cent of measured
1	0.021	42.0	41.7	-0.7
2	0.032	27.9	27.4	-1.8
3	0.048	17.8	18.3	+2.8
4	0.066	13.5	13.3	-1.5
5	0.081	11.1	10.8	-2.7
6	0.105	8.5	8.3	-2.3
7	0.136	6.2	6.5	+4.8
8	0.157	5.5	5.6	+1.8
9	0.189	5.0	4.6	-8.0
10	0.219	4.3	4.0	-7.0

to the measured entrance temperature $t_1 = 21.5°C$ and over to the right to the intersection with the vertical line from $\alpha = 1.0$ projected from the upper abscissa of quadrant (6), the point defining the reference air temperature is found. The line established for t_{a-r} is located at approximately 26°C. By calculation, the value would be found as $t_{a-r} = t_1 + (t_2 - t_1)/2 = 21.5 + 4.2 = 25.7°C$.

At the ordinate of quadrant (4) for $(t_c - t_1) = (71.0 - 21.5) = 49.5°C$, a horizontal line is drawn to the line corresponding to $(t_2 - t_1) = 8.3°C$. The vertical line drawn from quadrant (4) to quadrant (3) is stopped at the line corresponding to $K = 0.14$ which is arbitrarily assumed as a temporary value of K until n is determined. A horizontal line is drawn from the point established on the line for $K = 0.14$ and is intersected in quadrant (2) with the line for $\alpha = 1$. The vertical projection line from the point of intersection establishes a point in quadrant (1) corresponding to the air rate $W = 0.105$ pound per second.

For the air rates in the other runs the same procedure is used utilizing the values for α established in quadrant (6) by means of the line for $t_{a-r} = 26°C$. For example, Run No. 1, at $W = 0.021$ pound per second, $q = 400$ watts, gives $(t_2 - t_1) = 41.7°C$, and at $t_1 = 21.1°C$, $\alpha = 0.986$.

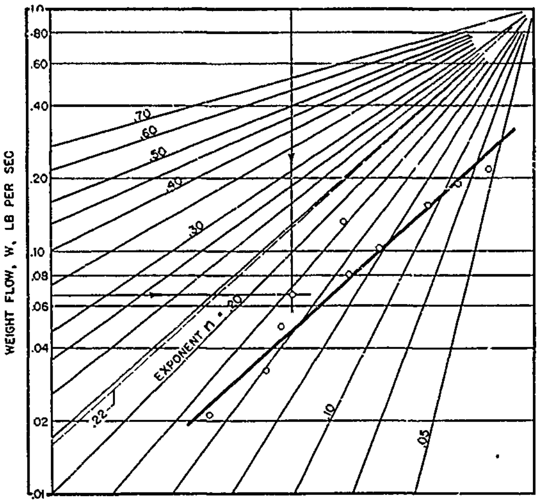

Figure VI-11. Test Data Plot in Quadrant (1) of Figure VI-7 for Determination of n (Example VI-4)

The points in quadrant (1) corresponding to the ten test runs are established in the above manner and give a plot as shown in Figure VI-11. The best line passing through the test points is indicated by the heavy solid line, and the n-line running parallel to this line is shown dotted and corresponds to n = 0.22. It should be observed that test points corresponding to low air rates are favored in establishing a best line, since under this condition overheating of the components is most likely to occur. However, the line represents a good average for high values of air rate so that n = 0.22 should satisfactorily describe the heat transfer characteristics of the cooling air passage.

The correct value for the constant K is found from the data of Run No. 6 by projecting a point in quadrant (1) on the line for n = 0.22 and W = 0.105 pound per second, vertically upward on the line for $\alpha = 1$ in quadrant (2) and from there horizontally to the right to quadrant (3). This latter horizontal projection line is brought to intersection with the previously established vertical projection line for Run No. 6 in quadrant (3) at a point through which the K-line has a value corresponding to K = 0.117.

By means of the described graphical procedure, the generalized correlation equation describing the heat transfer characteristics of this unit is established as

$$\alpha\, W^{0.22} \log_e \left[(t_c-t_1)/(t_c-t_2)\right] = 0.117,$$

where α is based on a reference temperature of $t_{a-r} = 26^{\circ}C$.

A second method for determining n and W is by direct plotting of the measured heat transfer data on log-log coordinate paper. Referring to equation (VI-8), it is apparent that by plotting $\alpha \log_e \left[(t_c-t_1)/(t_c-t_2)\right]$

versus the air rate W on log-log coordinate paper, a straight line should result, the slope of which is numerically equal to the value of n. A plot of this type is shown in Figure VI-12 using data taken from the tabular summary of all test data listed in this example.

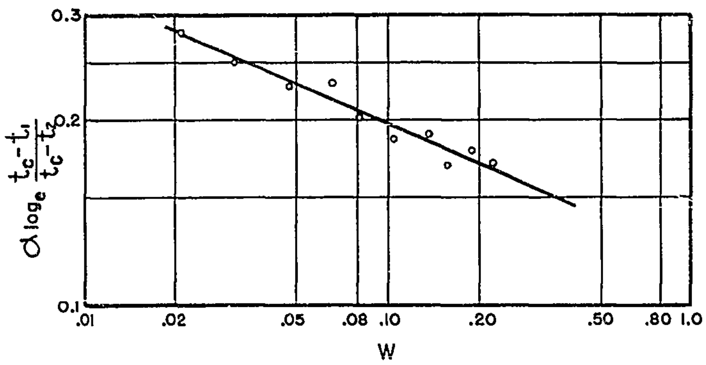

Figure VI-12. Plot of Reduced Temperature Data versus Air Rate (Example VI-4)

The points on the plot in Figure VI-12 are obtained by calculation based on an assumed reference temperature $t_{a-r} = 26^{\circ}C$. For example, for Run No. 1, the procedure is as follows:

$$(t_c-t_1) = 193 - 21.1 = 171.9^{\circ}C$$

$$(t_c-t_2) = 193 - 63.1 = 129.9^{\circ}C$$

$$\log_e \left[(t_c-t_1)/(t_c-t_2)\right] = \log_e (171.9/129.9) = 0.279$$

$$t_a = (t_1+t_2)/2 = (21.1 + 63.1)/2 = 42.1^{\circ}C$$

$$\alpha = (1170 + t_{a-r})/(1170 + t_a) = 1196/1212.1 = 0.986$$

$$\alpha \log_e \left[(t_c-t_1)/(t_c-t_2)\right] = 0.986 \times 0.279 = 0.275$$

For an average of all test points, a single straight line is drawn in Figure VI-12. The slope of this line is calculated to be 0.22, the value of the exponent n. For evaluation of the constant K, corresponding values of W = 0.10 and $\alpha \log_e[(t_c-t_1)/(t_c-t_2)] = 0.195$ are read from Figure VI-12. Therefore, from equation (VI-8)

$$K = 0.195\ (0.10)^{0.22} = 0.117$$

c. Correlation of Cooling Air Pressure Drops

The general correlation equation (VI-10) for the unit may be established on the basis of the test data by the graphical methods outlined on pages 162 to 164 using the chart of Figure VI-9, or by calculation and plotting of the corrected test data on log-log coordinate paper.

In using the graphical procedure, the values for σ_1 are found in Figure V-8. For example, for Run No. 7 the air pressure at the inlet to the cooling passage is

$$p_1 = p_{bar} + \Delta p/13.55 = 29.4 + 1.9/13.55 = 29.54 \text{ inches mercury absolute}$$

for this value and the inlet temperature $t_1 = 21.8^{\circ}C$, Figure V-8 gives $\sigma_1 = 0.96$.

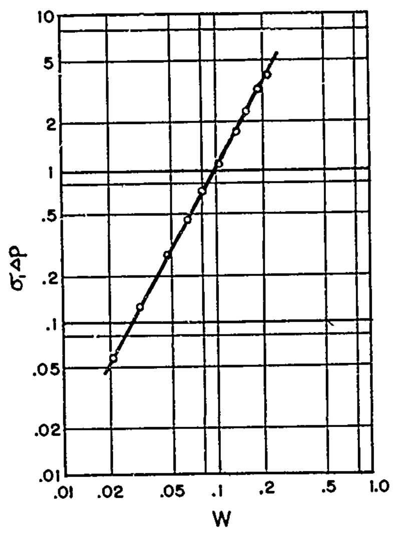

Figure VI-13. Plot of Corrected Pressure Drop Data (Example VI-4)

To plot the test data in Figure VI-9, so as to determine a characteristic line from which m may be determined, an arbitrary value of the constant C is assumed. For example, using C = 100 a plot as shown in Figure VI-13 is obtained in the right-hand quadrant of Figure VI-9. The individual points on this plot are established by drawing for each run a set of construction lines. For example, for Run No. 7 the lines start at the ordinate in the left-hand quadrant $\Delta p = 1.90$ inches water, pass horizontally to the line for $\sigma_1 = 0.96$, then vertically downward to the line for the arbitrarily chosen constant C = 100, and horizontally to the right-hand quadrant where they terminate at a point corresponding to the weight flow rate $W = 0.136$ pound per second. The line for the test points shown in Figure VI-13 is parallel to that corresponding to m = 1.85. Therefore, the value of m in the correlation equation (VI-10) would be the same.

The correct value of C is established simply by starting at a point on the line in the right-hand quadrant for m = 1.85 and the air flow rate for one particular run such as Run No. 7 and drawing a horizontal projection line to the left-hand quadrant where it is made to intersect the vertical line for Run No. 7 established by the pressure drop $\Delta p = 1.9$ inches water and $\sigma_1 = 0.96$. The value of C for the line which would pass through the point of intersection is found to be C = 73.5. Thus the pressure drop correlation equation for this unit is

$$\sigma_1 \Delta p = 73.5 (W)^{1.85}$$

The same result could be obtained, as pointed out above, by calculation and plotting of the data. First, the pressure drop data may be corrected by calculating the values for σ_1. For example, for Run No. 7, p_1 = 29.54 inches mercury absolute and t_1 = 21.8°C, as determined above. Thus, using the equation given in Figure V-8, the air density ratio at the inlet to the unit is,

$$\sigma_1 = 9.63\, p_1/(273 + t_1) = 9.63 \times 29.54/(273 + 21.8) = 0.965$$

Then, the corrected pressure drop for Run No. 7 is

$$\sigma_1 \Delta p = 0.965 \times 1.9 = 1.83 = C(W)^m$$

The data for all test runs, according to the above method, are as follows:

Run No.	Air flow, W	$\sigma_1\Delta p$	Run No.	Air flow, W	$\sigma_1\Delta p$
1	0.021	0.058	6	0.105	1.09
2	0.032	0.125	7	0.136	1.83
3	0.048	0.280	8	0.157	2.42
4	0.066	0.464	9	0.189	3.30
5	0.081	0.705	10	0.219	4.26

A log-log plot of the above data gives a straight line as shown in Figure VI-13 from which values of m and C may be found. The slope of the line is equal to m and is found by linear measurement of the horizontal and vertical intercepts to be 1.85. Using a point on the data line in Figure VI-13 corresponding to W = 0.10, a value of $\sigma_1\Delta p$ = 1.04 is found. Then, $C = \sigma_1\Delta p/W^m = 1.04/0.10^{1.85} = 73.5$.

The correlation equation established by the above methods evaluates the pressure drop only for the unit proper and does not include the resistance due to ducting or other flow restrictions between it and the blower that may be used to supply the cooling air under installation conditions. Therefore, in applying this correlation to the selection of a blower care would have to be taken to evaluate all other flow resistances since otherwise the installed blower capacity would be insufficient under operational conditions.

Example VI-5. Determination of Blower Specifications for Temperature Limitation Under Operational Conditions

A pressurized unit cooled by forced convection was bench tested. Correlation of the data gave the following constants for equations (VI-8 and -10).

n = 0.20		m = 1.9	
K = 0.13		C = 150	
t_{a-r} = 30°C		q = 200 watts	

For operation at 50,000 feet altitude, at atmospheric pressure of p = 3.44 inches mercury, and a compartment temperature of t_o = 30°C, the characteristics of a blower are to be found which will insure that t_c = 85°C. This case temperature is the highest allowable since with it several components reach their limiting temperatures for reliable operation. It is desired to determine (1) the characteristics of a blower to induce flow through the air cooling passage, and (2) the characteristics of a blower to provide for forced flow through the air cooling passage based on a probable static efficiency of 50 per cent.

a. Analysis of Induced Flow Requirement

The most convenient manner to determine the required weight flow rate for induced flow is to find the case surface temperature t_c corresponding to two or more rates of air flow and to determine then the required air rate either from a plot or by interpolating between known values. A good basis for the first choice of air rate is given by the test data since for the same difference between the case surface temperature of the unit and the temperature of the air supplied to the unit, i.e., the value of (t_c-t_1), the air rate would remain approximately constant. Here, a value of (t_c-t_1) = (85 - 30) = 55°C is desired and therefore an air rate among the test data providing for a similar value of (t_c-t_1) should be chosen as a first approximation.

The required data can be determined by use of the working chart in Figure VI-7. For example, an air rate of W = 0.04 pound per second is chosen. Using quadrants (5) and (6), for W = 0.04, q = 200, t_1 = 30, and t_{a-r} = 30, the values of (t_2-t_1) = 11°C and α = 0.996 are determined. Then by successive use of quadrants (1), (2), (3), and (4), and using the values of n = 0.20, α = 0.996, K = 0.13, and (t_2-t_1) = 11, the value of (t_c-t_1) = 50°C is found. From this, t_c = 50 + 30 = 80°C. This result indicates that a slight reduction in the air rate is necessary to obtain the desired t_c = 85°C. A second assumption for W = 0.035 gives t_c = 87°C. Since the two values approach very closely the desired value, straight-line interpolation can be used to find the desired air rate which is established as W = 0.036 pound per second. Otherwise, at least one or two additional assumptions must be made which permit plotting of the data so that t_c versus W will give a curve from which the value of W corresponding to 85°C could be picked.

In order to determine the required characteristics of the blower which would serve to induce flow over the heat exchange surfaces of the unit, the conditions of the air at entrance to the blower, i.e., at the discharge from the unit must be determined. From quadrant (5) of Figure VI-7, the temperature rise of the air passing through the unit is found to be (t_2-t_1) = 12°C so that the temperature at the entrance to the blower would be t_2 = 42°C. For determination of the absolute pressure at the inlet to the blower, the pressure drop in the unit must be determined. The air density ratio σ_1

at the inlet to the unit is defined by Figure V-8 and more accurately by the equation shown there. By calculation it is found that $\sigma_1 = 0.1092$. The pressure drop of the unit may either be calculated from equation (VI-10) or may also be determined conveniently from Figure VI-9. For $W = 0.036$, $m = 0.9$, $C = 150$, and $\sigma = 0.1092$, it is found that $\Delta p = 2.75$ inches water. Therefore, the absolute pressure at the inlet to the blower is determined by

$$p_2 = p_i = p_o - \Delta p/13.55 = 3.44 - 2.75/13.55 = 3.24 \text{ inches mercury.}$$

With above value and $t_2 = 42^oC$, σ_2 may readily be calculated and is found to be $\sigma_2 = 0.099$ which is also the value of σ_i on basis of which the performance of the blower must be corrected. Consequently the corrected static pressure of the blower is

$$\Delta p/\sigma_i = 2.75/0.099 = 27.8 \text{ inches water.}$$

The required volume flow of the blower is determined from equation (V-1), based on the assumption that the standard performance of the blower at ground level conditions is given for an air density equal to 0.0765 pound per cubic foot. This may not always be the case. However, any performance data may readily be converted by correcting them according to the methods indicated in Chapter V. Here, the volume flow requirement of the blower is obtained by

$$Q = 784\ W/\sigma_i = 784 \times 0.036/0.099 = 285 \text{ cubic feet per minute.}$$

b. Analysis of Forced Flow Requirement

If forced flow is used, the properties of the air at the entrance to the unit are not exactly defined since they are dependent on the pressure drop through the unit which in turn is determined by the air flow rate necessary to produce the desired case surface temperature. Therefore, the process is one of trial and error and can be simplified by the use of graphical methods. As a starting point in the series of assumptions which must be made, it may be assumed that the pressure the blower would have to create would be equal to the pressure drop through the unit when induced flow is used. However, it must be realized that this is usually on the low side since particularly with units of appreciable resistance such as the one with which this problem is concerned, a temperature rise which is not negligible occurs as the air passes through the blower. In the present example, the air density ratio at the inlet to the blower is the same as at the inlet to the unit when induced flow is used. Therefore, $\sigma_i = 0.1092$. If for a first assumption the pressure drop is assumed to be $\Delta p = 3$ inches water, $\Delta p/\sigma_i = 3/0.1092 = 27.5$, and an average blower efficiency of 50 per cent is assumed, Figure V-11 shows the temperature rise across the blower to be $\Delta t_B = 11^oC$. To define the flow through the unit the air density ratio at the exit of the blower must be determined since it is the value of σ_1 to be used in the system resistance equation (VI-10) or in the chart of Figure VI-9. Thus,

$$p_1 = 3.44 + 3/13.55 = 3.66 \text{ inches mercury}$$

$$t_1 = 30 + 11 = 41^\circ C$$

$$\sigma_1 = 9.63 \times 3.66/(273 + 41) = 0.112.$$

Using Figure VI-9 in reverse, i.e., starting at $\Delta p = 3$, and proceeding to the right to $\sigma = 0.112$, then downward to $C = 150$, then to the right to $m = 1.9$, and downward from there, the corresponding weight flow is found to be 0.0394 pound per second. From the chart in Figure VI-7, using $W = 0.0394$ and $t_1 = 41$, the value for the case surface temperature t_c is found in the manner indicated above. For the first assumption it is $t_c = 91^\circ C$. In view of this, it appears that it is necessary to increase the flow through the unit which would result in higher pressure drop. Therefore a new assumption of pressure drop is made for which the same procedure is followed. Based on the uniform assumption of a static efficiency $\eta_s = 0.50$, the following data are obtained in this manner.

Δp	t_1	W	t_c
3	41	0.0394	91
4	45	0.0471	89
6	52	0.0590	87
8	59	0.0693	90
9	63	0.0735	93

The plot of these data shown in Figure VI-14 indicates that a minimum value of case temperature of about 87°C is reached and that it appears to be im-

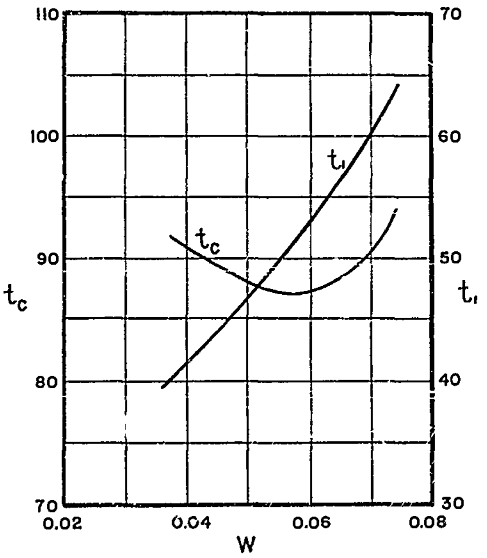

Figure VI-14. Variation of Case Temperature with Air Flow Rate of Forced-Flow Blower (Example VI-5) possible to obtain a case temperature of 85°C. This is due to the fact that with increasing air flow the pressure drop of the unit would increase and therefore the temperature rise across the blower would also go up as indicated by the plot of t_1 versus W. This is an important criterion which may prevent the use of a forced-flow blower for cooling of a unit of relatively high resistance. Such a unit would require the use of an induced-flow blower.

From the data in Figure VI-14 it is also apparent that if the efficiency of the blower would be higher than 50 per cent it is conceivable that the temperature rise may be sufficiently small that a case surface temperature of 85°C could be obtained. At the minimum point where t_c = 87°C, Δt_B = 22°C. Consequently if Δt_B could be reduced to 20°C it would appear that t_c could also be reduced by 2°C. The temperature rise across the blower being inversely proportional to the efficiency, a value of η_s = 0.55 is indicated. In fact, using the same procedure as before, it is found that the air rate is essentially the same as previously determined for t_c = 87°C, i.e., W = 0.0591 pound per second, giving t_c = 85°C if η_s = 0.55. The pressure drop is also unaltered. Therefore the characteristics of the blower must be

$$\Delta p/\sigma_1 = 6/0.1092 = 55 \text{ inches water}$$

$$Q = 784 \times 0.0591/0.1092 = 424 \text{ cubic feet per minute.}$$

It is apparent from the above data that the use of a forced-flow blower for the unit treated in this example would be disadvantageous since its pressure-producing ability would have to be twice that of the induced-flow blower and its discharge volume almost twice as great. Both factors would contribute to making the size of the blower greater as well as to increasing its power requirements.

Example VI-6. Determination of Case Surface Temperature Under Operational Conditions with Blower of Known Characteristics

A pressurized unit having a case cooled by forced convection was bench tested and the following constants were determined:

$n = 0.25$	$m = 1.85$
$K = 0.15$	$C = 50$
$t_{a-r} = 20°C$	$q = 400$ watts

Subsequently, selection was made of an induced-flow blower to provide a case surface temperature t_c = 85°C at 40,000 feet altitude where the compartment pressure due to flight speed effect was assumed to be 5.80 inches mercury and the air temperature 30°C.

The characteristics of the blower selected for the unit are shown in Figure VI-15, based on standard conditions (ρ_r = 0.0765 pound per cubic foot). It is desired to check the operation of the system for 60,000 feet altitude with an assumed compartment pressure of p_o = 2.50 inches mercury

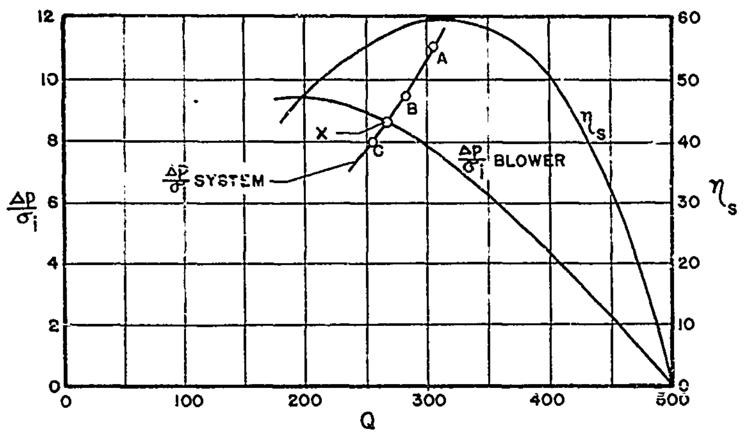

Figure VI-15. Characteristics and Method for Determination of Operating Point of Induced-Flow Blower (Example VI-6)

and an air temperature of $t_o = 0^oC$. It is also desired to determine whether under these conditions forced-flow may be more advantageous than induced-flow using the given blower. The speed of the blower remains constant under all conditions. Essentially, the methods discussed in Chapter V, page 160 to 163, are applicable with minor modifications.

a. Analysis of Induced-Flow Application

Under the conditions for which the blower was selected, the desired weight flow rate was $W = 0.0605$ pound per second and the air density ratio was $\sigma_1 = \sigma_o = 0.184$. For the specified operating conditions, $\sigma_1 = 0.0881$. Since the air density ratio is reduced to roughly one-half of the value for the conditions under which the blower was selected, it would appear that the air rate under these extended operational conditions would be roughly one-half of that previously determined. For $W = 0.03$, and $q = 400$, $(t_2-t_1) = 29$, as found in quadrant (5) of Figure VI-7. Therefore, the inlet temperature to the blower would be $t_2 = t_c = 29^oC$. From Figure VI-9 and the given constants, using $W = 0.03$ and $\sigma_1 = 0.0881$, the pressure drop of the unit is found to be $\Delta p = 0.862$ inches water. Consequently, the inlet pressure to the blower is $p_i = 2.50 - (0.862/13.55) = 2.44$ inches mercury. Corresponding to these values, the air density ratio at the entrance to the blower $\sigma_i = 0.0776$. Then, the corrected pressure rise of the blower would have to be

$$\Delta p/\sigma_i = 0.862/0.776 = 11.1 \text{ inches water,}$$

and correspondingly

$$Q = 784 \times 0.03/0.0776 = 303 \text{ cubic feet per minute.}$$

These values plotted on the coordinates of the blower characteristics in Figure VI-15 give point A which is seen to be considerably above the characteristic curve. Therefore, the weight flow at the operating point is expected to be considerably lower. Two additional assumptions of the air rate W give points B and C in Figure VI-15 so that the system characteristics can be plotted. At the same time, a plot of W versus Q may be made (not shown). Then at the intersection marked by X the resulting flow volume is found for which the corresponding air rate is determined from the plot of W versus Q. The value so found is $W = 0.0262$ pound per second. By use of the working chart in Figure VI-7 in the previously indicated fashion, for $W = 0.0262$, $t_1 = 0°C$ and the other known values, it is found that $(t_c - t_1) = 108°C$ which for $t_1 = 0°C$ makes $t_c = 108°C$.

Thus it is seen that in spite of the considerable decrease in the cooling air temperature, the reduced density due to reduced atmospheric pressure at 60,000 feet altitude is responsible for an appreciable decrease in the air rate which would result in overheating of the unit, if the speed of the blower remains constant.

b. Analysis of Forced-Flow Application

It is to be determined what case surface temperature the same blower as used for induced flow would produce under forced-flow conditions for $p_o = 2.50$ inches mercury and $t_o = 0°C$. For this analysis it is desirable to convert the blower characteristics into a plot of $\sigma_1 \Delta p$ versus W, where Δp is the pressure rise across the blower and $\sigma_1 = \sigma_d$, the air density ratio at the discharge of the blower. Thus, this value would be equivalent to the system resistance characteristics defined by $C(W)^{1.85}$. Regardless of the flow rate, the air density ratio at the inlet to the blower σ_i is that of the installation compartment, previously calculated to be $\sigma_o = 0.0881 = \sigma_i$. The subsequent procedure is to assume a volume flow rate for which corresponding values of static pressure, efficiency, and temperature rise across the blower can be determined so as to define $\sigma_1 \Delta p$. From the previous analysis of the induced-flow application it would appear that the desired flow rate should lie between 300 and 200 cubic feet per minute. Assuming a flow rate of Q = 300 cubic feet per minute, corresponding values of $\Delta p/\sigma_i = 7.9$ inches water and $\eta_s = 0.595$ are determined from the plot of the blower characteristics in Figure VI-15. For good accuracy it is desirable to calculate the temperature rise across the blower which would be, according to equation (V-3),

$$\Delta t_B = (0.202/\eta_s)(\Delta p/\sigma_i) = (0.202/0.595)(7.9) = 2.68°C$$

The pressure rise across the blower is

$$\Delta p = \sigma_i\,(\Delta p/\sigma_i) = 0.0881 \times 7.9 = 0.696 \text{ inch water}$$

Thus, the absolute discharge pressure and temperature of the blower are defined. The air rate itself is determined by the inlet conditions to the blower. The following values are obtained:

$$p_d = p_1 = p_o + \Delta p/13.55 = 2.50 + 0.696/13.55 = 2.55 \text{ inches mercury}$$

$$t_d = t_1 = t_o + \Delta t_B = 0 + 2.68 = 2.68°C$$

$$\sigma_d = \sigma_1 = 9.63\, p_1/(273 + t_1) = 9.63 \times 2.55/275.7 = 0.0892$$

$$\sigma_1 \Delta p = 0.0892 \times 0.696 = 0.062 \text{ inch water}$$

$$W = \sigma_1 Q/784 = 0.0892 \times 300/784 = 0.0341 \text{ pound per second}$$

$$C (W)^{1.85} = 50 (0.0341)^{1.85} = 0.0962 \text{ inch water}$$

By the same method corresponding values for flow rates of 250 and 200 cubic feet per minute can be determined and give plots as shown in Figure VI-16.

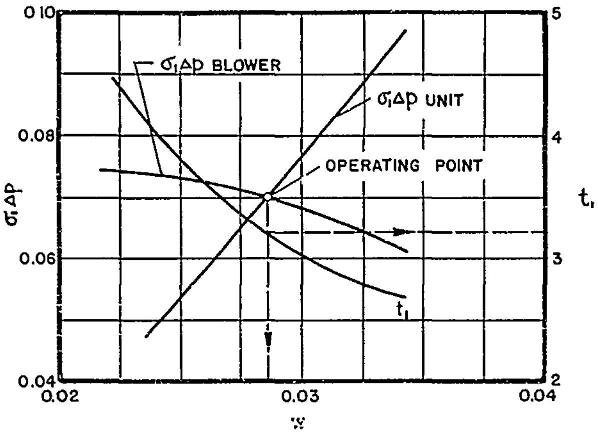

Figure VI-16. Discharge Characteristics and Method for Determination of Operating Point of Forced-Flow Blower (Example VI-6)

At the intersection of the lines in Figure VI-16 for $\sigma_1 \Delta p$ of the blower and $\sigma_1 \Delta p = C(W)^{1.85}$ for the unit, the operating point is found which indicates a corresponding air rate $W = 0.0286$ pound per second and a blower discharge temperature of $t_1 = 3.25°C$. Using these values in the working chart in Figure VI-7, the case surface temperature rise is found to be $(t_c - t_1) = 102°C$ from which the case temperature of the unit is defined as $t_c = 105.3°C$ which is only 2.7°C lower than the case temperature obtained with induced flow. Thus, it is apparent that for this type of unit with relatively low pressure drop there is little difference between the effects of induced and forced flow. Because of slightly greater density of the air supplied to the forced-flow blower, its air-handling capacity is better than that of the induced-flow blower. Since the heat exchange system also would operate at somewhat greater air density, a tendency for reduced pressure drop exists. By comparison of the air rates for the two types of flow it may be seen that the blower unit operating at the same speed would provide roughly 10 per cent more air flow if operating under forced-flow conditions. The temperature rise of the air passing through the blower, in the order of 3°C, would have relatively little significance compared to the temperature difference between the incoming air and the surface of the unit which would be in the order of 100°C. The effect on component temperatures, which would change approximately as much as the case temperature is of negligible importance in respect to component reliability and life.

Pressurized and Sealed Units with Integrated or Separate Heat Exchanger Cooled by Forced Convection

Units of this type are usually capable of maintaining the same component temperatures for greater heat concentrations per unit volume than other types of sealed units. This capacity for increased heat dissipation at a given temperature level results from the use of an integrated or separate heat exchanger which provides additional surface for heat transfer. In comparison with a unit having its case cooled directly by forced convection, greater surface areas are available for heat transfer from internal circulating air and for heat transfer to external cooling air. In comparison with a unit having a case-envelope heat exchanger, the surface area is increased and the thermal resistance is lowered mainly for the internal heat transfer process. The heated air circulating inside the equipment is made to flow over the heat exchange surfaces by means of an internal blower and is cooled by air flow created by either an external blower or ram action. A schematic arrangement of a unit with an integrated heat exchanger is shown in Figure II-3, page 10, and of a unit having a separate heat exchanger, being part of a central system, in Figure II-4, page 10.

The three basic heat exchange processes occurring with this type of equipment are (1) heat dissipation of the electronic components to the air circulating through the component space by action of the internal blower, (2) heat exchange in the integrated or separate heat exchanger, from the internal air to the external or cooling air, each on opposite sides of the heat exchanger's surface (caused by the temperature differential existing between the internal and external air flows), and (3) heat exchange between the internal air and components and the equipment's environment by transfer through the equipment's case or, in addition, through duct walls when a separate heat exchanger is employed. Figure VI-17 illustrates schematically the internal and external air flow paths as well as the general heat exchange processes listed

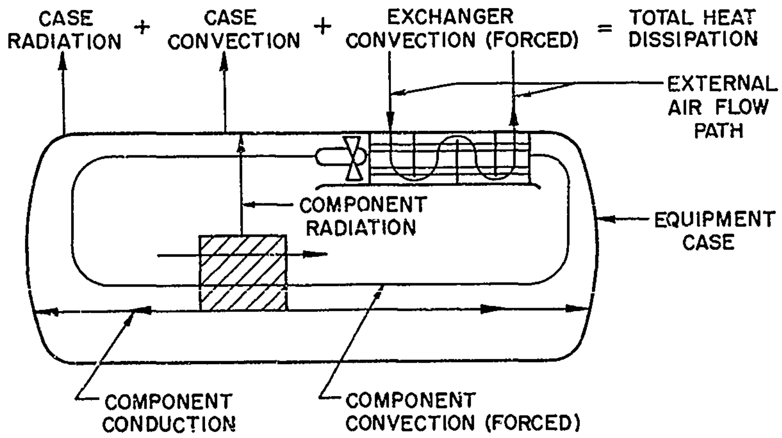

Figure VI-17. Heat Transfer Diagram of Closed Unit with Integrated Heat Exchanger above. The heat dissipation from the components to the internal air flow occurs principally by forced convection during passage of the internal air through the component space. The heated air is moved by action of the internal blower to the integrated heat exchanger, or to a separate heat exchanger by way of additional ducting. The air temperature decreases while passing through the exchanger. Then, the air is discharged to the component space. The total heat dissipated from the unit equals the heat transfer by forced convection to the external cooling air plus any heat transfer from the case surface to its environment, generally by radiation and free convection. Heat gain from the environment to the internal structure of the equipment may occur in some installation. Then, the heat to be dissipated in the heat exchanger would equal the heat gain through the case surface plus the heat dissipated by the components within the equipment.

The heat sink for the heat removed from the internal air during passage through the heat exchanger is air, designated as external cooling air. This air may be supplied to the heat exchanger by an external blower, or by ram action. The external blower may be located at the entrance or exit of the heat exchanger, depending upon whether forced or induced flow is desired.

The location of an integrated heat exchanger within a unit depends upon the required size of the core, the general arrangement of the heat-producing components, and the path of the internal air flow. It may be located in one end of the case, possibly opposite to the end housing the internal blower. Or, if the spatial requirement of the heat exchanger is not great, and its configuration is of the "pancake" type, the heat exchanger may be suspended from an upper panel near one end of the case. From the standpoint of internal pumping power, it is always desirable to place the exchanger near or at the top of the case so that the internal air flow is aided to the greatest extent by natural convection.

In the following, the correlation of bench test data and their use for prediction of thermal performance at other operating conditions are discussed. The methods for a unit having an integrated heat exchanger are the same as for a unit utilizing a separate heat exchanger except for the evaluation of heat loss or gain of the internal air while passing through the ducts to and from the separate heat exchanger. Measurements and bench test procedures for this general type of equipment are described in Chapter IV. In Example VI-7 measurements made on a unit during bench test and the procedure for reducing the test data to a generalized form are given. In Example VI-8 the use of the generalized data obtained from bench test, for the evaluation of required external blower performance under altitude conditions is illustrated. The prediction of operating temperatures of components at other than bench test conditions is illustrated in Example VI-9.

1. Heat Transfer in the Heat Exchanger

The integrated or separate heat exchanger in a unit of this type must transfer from the internal air to the external cooling air a quantity of heat q_{ex} equal to the difference between the heat q dissipated by the heat-producing components and the heat q_c lost to the environment through the equipment case. Hence,

$$q_{ex} = q - q_c \tag{VI-12}$$

The case heat transfer rate q_c is negative when heat is gained by the unit from the environment, making the total heat that must be dissipated by the heat exchanger equal to the sum of the heat produced by the components and that gained from the environment.

When employing an integrated or separate heat exchanger the unit would be pressurized or sealed. Therefore, the internal pressure would be independent of altitude and the flow rate at which the internal blower circulates air through the unit would remain constant for all operational conditions. Hence, for a given unit with known heat dissipation and having a heat exchanger of fixed size and configuration, the temperature level of the component space is determined by the temperature and flow rate of the external cooling air. The average temperature level of the air passing through the exchanger determines the physical properties of the air which affect the heat transfer characteristics of the exchanger secondarily only, since these characteristics vary little for the temperature range encountered in the operation of the exchanger.

The difference in temperature level between the internal and external air is of major significance in defining the heat transfer ability of the exchanger. The maximum temperature difference in the heat exchanger is the difference between the temperatures of the internal and external air at the entrance to the heat exchanger, $(t_{it-1}-t_{e-1})$. The ratio of the heat transfer rate of the heat exchanger q_{ex} and the inlet temperature differential $(t_{it-1}-t_{e-1})$ defines a heat exchanger parameter H_{ex} in watts per $^{\circ}C$ as

$$H_{ex} = q_{ex}/(t_{it-1} - t_{e-1}) \tag{VI-13}$$

For a heat exchanger of fixed size and configuration, having a constant circulation rate on one side of its heat transfer surface, H_{ex} may be shown to be only dependent on W_e, the external cooling air rate. Therefore, the equation

$$H_{ex} = q_{ex}/(t_{it-1} - t_{e-1}) = f\ (W_e) \tag{VI-14}$$

is valid for all operational conditions. The functional relationship $f(W_e)$ is evaluated from bench test data by plotting H_{ex} versus the external air rate W_e. For any temperature condition, the plot serves to define the required cooling air rate for the required heat dissipation rate.

The temperature drop of the internal air across the heat exchanger $(t_{it-1}-t_{it-2})$ is related to the heat transfer rate q_{ex} and the internal air rate W_{it} by the heat balance equation

$$q_{ex} = 456\ W_{it}(t_{it-1} - t_{it-2}) \tag{VI-15}$$

Similarly, the temperature rise of the external cooling air $(t_{e-2}-t_{e-1})$ is related to q_{ex} and W_e by

$$q_{ex} = 456\, W_e(t_{e-2} - t_{e-1}) \qquad \text{(VI-16)}$$

With temperature expressed in °C and air rate in pounds per second, the heat transfer rate q_{ex}, determined by equations (VI-15 and -16), is in watts. The mean temperatures of the internal air at entrance and exit to the heat exchanger, i.e., t_{it-1} and t_{it-2} respectively, must be measured during bench test of the equipment. In general, probing the inlet and exit sections of the heat exchanger is required in order to define mean values of air temperature. On the external cooling air side of the exchanger the temperatures t_{e-1} and t_{e-2} and the external cooling air rate W_e must be measured during bench test. Probing of the air stream at discharge from the exchanger is required to define a mean value for t_{e-2}, but a single measurement of the inlet temperature t_{e-1} should be sufficient. Knowledge of these temperatures and the external cooling air rate allows evaluation of the heat exchanger's dissipation rate q_{ex} by equation (VI-16), and of the internal air rate W_{it} by equation (VI-15).

2. Evaluation of Heat Transfer Through Equipment Case

As illustrated in Figure VI-17, heat may be transferred from the components to the equipment case, or vice versa, by conduction through the chassis and by direct radiation. Also, the internal air flow undoubtedly creates some forced convective heat transfer over the inside surface of the case. In general, the entire heat transfer process is quite complex and does not permit direct evaluation. The heat transfer between the case surface and the equipment's environment may be evaluated by procedures outlined on pages 131 to 139, since usually the heat is transferred principally by free convection and radiation.

A working method for evaluation of case heat loss or gain is obtained by plotting the case heat loss q_c as a function of the difference between the average internal air temperature t_{it-av}, and the case surface temperature t_c, all data having been determined from bench test of the equipment. The average internal air temperature is defined by

$$t_{it-av} = (t_{it-1} + t_{it-2})/2, \qquad \text{(VI-17)}$$

the average of the entrance and exit temperature of the internal air for both component space and heat exchanger. The case heat transfer q_c occurring during bench test is best evaluated by calculation of free convection and radiation between the case surface and the test environment. Correction factors to be applied in the evaluation of radiation and free convection, discussed on page 138, may be selected on basis of bench tests of geometrically similar closed equipments where all heat is dissipated through the case surface. If such tests are not available, the correction factors must be estimated or may possibly be ignored. Otherwise, the case heat transfer may be evaluated from the heat balance equation (VI-12), once the heat dissipation in the heat exchanger is known. After that the approximate correction factor

F_c to be applied to radiation and free convection from the exterior of the case, calculated by the method outlined on page 138, may be determined. As a general rule it is recommended that the entire case surface be utilized for evaluation of free convection and radiation.

Bench tests of the equipment should cover a sufficiently large range of conditions so that a variation in the temperature differential ($t_{it-av}-t_c$) is obtained which contains values of this differential likely to be encountered under aircraft operational conditions. During bench test, insulation of the case would probably be required for units dissipating more than 10 per cent of the total heat generated through the case surface, since in all likelihood, even at equal internal air temperature, higher case temperatures would prevail under operational conditions at high altitude and in close confinement because of reduced convection and radiation from the case surface.

The recommended procedure of plotting case heat transfer rate q_c as a function of the internal temperature differential ($t_{it-av}-t_c$) allows direct evaluation of q_c only if the case surface temperature t_c and the average internal air temperature t_{it-av} are known. In the application of this method a trial-and-error process is necessary to define the case surface temperature which meets the requirement that the heat transfer rate between the internal air and the case surface and that between the case surface and the unit's environment be equal. Equality of these two heat transfer rates defines the steady-state case temperature.

Ducts required with a separate heat exchanger provide additional surface for transfer of heat to or from the environment. This surface should be included in evaluation of case heat transfer, using essentially the same procedures.

3. Heat Transfer in Component Space

The surface temperatures of the individual components within the unit may be defined in terms of the average temperature of the air within the component space t_{it-av}. Trend curves giving average or hot-spot surface temperatures of the individual components as a function of t_{it-av} are determined from bench test data, where t_{it-av} is varied by changing the rate of external cooling air flow. These trend curves allow evaluation of component surface temperatures under any operational condition, since the internal air rate through the component space remains constant. An example of trend curves so established may be seen in Figure VI-20, Example VI-7. It is apparent that, in general, permissible thermal conditions can be maintained by control of the average air temperature within the component space.

4. Resistance of External Cooling Air Passages

A knowledge of the flow resistance of the external cooling air passages of the heat exchanger is necessary for complete evaluation and prediction of thermal performance at other than bench test conditions. This resistance should be the total of that of the passages within the heat exchanger and of the inlet and exit air headers. Procedures for generalizing bench test data on resistance are the same as for other type units cooled by forced convection. The working equation is

$$\sigma_1 \Delta p = C (W_e)^m \qquad \text{(VI-18)}$$

where Δp is the flow resistance of the exchanger in inches of water, and σ_1 the air density ratio at the entrance to the exchanger. The constant C and the exponent m are evaluated from bench test data of Δp and σ_1 for several values of the external air rate W_e in pounds per second. The procedure for evaluating C and m is discussed on page 154. A useful alternative to that procedure is to plot the term $\sigma_1 \Delta p$ obtained from bench test data versus the external air rate W_e, so that the pressure drop Δp may be determined directly from this plot for any operational condition if the external air rate is known.

If forced flow is used, evaluation of temperature rise across the blower is important, since it not only affects the density ratio σ_1 but also the inlet temperature differential $(t_{it-1}-t_{e-1})$ of the heat exchanger.

The methods required for determining the external air rate when the blower and its drive motor have prescribed or known performance characteristics are discussed in Chapter V.

5. Summary of Working Curves Necessary for Analysis of Thermal Performance

A summary of working curves derived from bench test data of unit with integrated or separate heat exchanger, needed for thermal evaluation of the unit at other than bench test conditions is illustrated in Figure VI-18. Plot A presents trend curves which show the effects of the average temperature of the internal air on the surface temperature of all thermally critical components. The working curve for evaluation of case heat transfer q_c is shown in plot B. Plots C and D describe the heat transfer parameter and flow resistance of the heat exchanger, respectively, as affected by the flow rate of external cooling air. Additional information on blower and motor characteristics is necessary when the external cooling air is to be supplied by a specific blower unit and if it is desired to evaluate the performance of the equipment-blower-motor combination under aircraft operation conditions.

6. Working Procedure for Units with Internal Baffle

Some units with integrated or separate heat exchanger may have a baffle which encloses the component space completely and may be concentric with the equipment case. It forms an annular passage for the return flow of internal air from the component space to the heat exchanger. An arrangement of this type is illustrated in Figure VI-19, page 186. With this arrangement a more rigorous analysis of the heat transfer processes may be made. In particular, the case heat transfer may be evaluated more accurately.

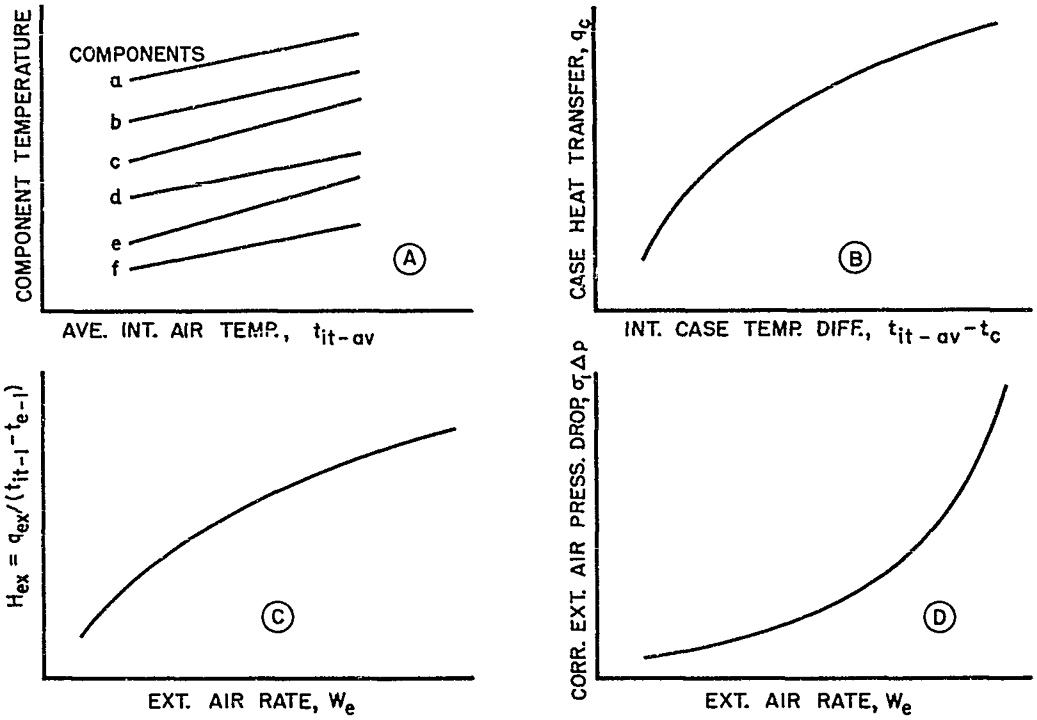

Figure VI-18. Summary of Working Curves for Unit with Integrated or Separate Heat Exchanger

The temperature of the heated air leaving the component space t_{it-3} may be measured during bench test. Furthermore, because of the presence of the internal baffle it may be assumed that all heat dissipated by the components is absorbed by the internal air passing through the component space and over the inside surface of the baffle. Hence, a heat balance across the component space yields

$$q = 456\, W_{it}(t_{it-3} - t_{it-2}), \qquad \text{(VI-19)}$$

where t_{it-3} represents the mean air temperature at entrance to the return flow passage formed by the baffle and the case. Equation (VI-19) allows definition of the internal air rate W_{it} from bench test data. This additional heat balance permits a check on the accuracy of the heat balance of the heat exchanger by equations (VI-15 and -16).

The internal air returning to the heat exchanger through the annular passage undergoes a temperature change due to heat transfer through the case surface, which is defined by the heat balance equation

$$q_c = 456\, W_{it}(t_{it-3} - t_{it-1}) \qquad \text{(VI-20)}$$

The case heat transfer may be correlated in terms of the average temperature differential available for transfer of heat to or from the case surface. This temperature differential is defined by the difference between the average temperature of the air returning in the passage formed by the case and the baffle, $t_{it-b-av} = (t_{it-3}+t_{it-2})/2$ and the average case surface temperature t_c. The case heat transfer rate q_c, plotted as a function of $(t_{it-b-av}-t_c)$ derived from bench test data, establishes a working curve for evaluation of case heat transfer under all operational conditions of the equipment. The ratio $q_c/(t_{it-b-av}-t_c)$ should remain essentially constant for all values of case heat transfer. The calculation of a case temperature which satisfies the requirement that q_c across the inside thermal resistance equals that across the outside thermal resistance requires a trial-and-error procedure.

Except for the above-described variations, all other evaluation procedures are the same as previously discussed for units without internal baffle.

7. Discussion of Evaluation Procedures for Prediction of Thermal Performance

Two general types of thermal evaluation problems are encountered with units cooled by an integrated or separate heat exchanger. The first problem is an evaluation of the cooling air rate W_e and the corresponding performance of a blower or a ram air induction system required to maintain the average temperature within the component space at or near a specified level which will insure against overheating the electronic components. The second type of problem occurs when a blower-motor combination having known performance characteristics is used to supply air to the heat exchanger, either by forced or induced flow, and it is desired to evaluate the operating temperatures of the electronic components. Either type of problem requires a trial-and-error solution in which one variable is assumed and must be checked by defining the performance of the entire heat transfer system.

Suppose, for example, the external cooling air rate W_e, the temperature of the external cooling air at the entrance to the exchanger t_{e-1}, and the generalized bench test performance data are known and that it is desired to determine the average temperature of the internal air within the component space. First, the temperature of the internal air at the entrance to the heat exchanger t_{it-1} is assumed. This allows evaluation of q_{ex} by equation (VI-14) or a plot such as C in Figure VI-18 since the external cooling air rate W_e is known. Next the case heat transfer is evaluated from the heat balance equation (VI-12). The internal air temperature at exit from the heat exchanger t_{it-2} is determined from equation (VI-15), using the value of W_{it} established in the bench tests. This allows evaluation of the average internal air temperature t_{it-av} by equation (VI-17) and the required case surface temperature t_c from the generalized case heat transfer plot, such as curve B in Figure VI-18. Knowing t_c and the environmental temperature of the equipment, one may calculate the heat transfer between the case surface and the environment. This value must agree with the previously defined case

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8. Examples

Example VI-7. Description of Bench Test and Reduction of Test Data to Generalized Form for the Prediction of Thermal Performance of a Pressurized Unit with Integrated Heat Exchanger and Internal Baffle

a. Description of Equipment and Procedure of Measurements

An electronic unit having an integrated heat exchanger is bench-tested to determine the thermal performance data necessary for the prediction of thermal operation at other than bench test conditions. A schematic arrangement of the equipment and its cooling system is shown in Figure VI-19. The case consists of a closed cylinder 12 inches in diameter and 12 inches long. The equipment is cooled by an internal heat exchanger of the tubular type, containing 230 tubes each of 0.20-inch outside diameter, 0.19-inch inside diameter, and 8 inches long. The tube bundle dimensions are 4 by 4 by 8 inches.

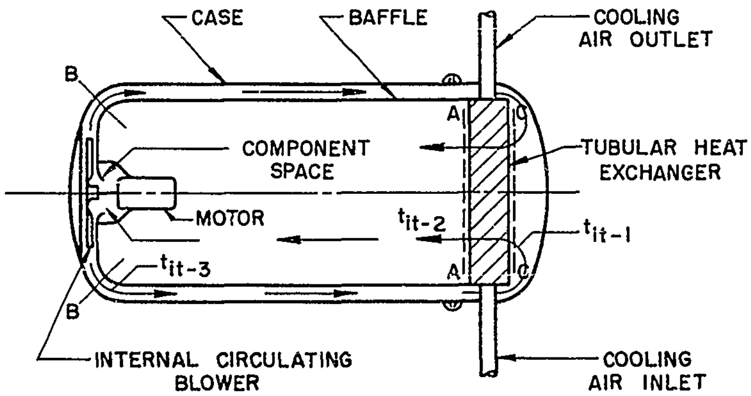


Figure VI-19. Schematic of Unit with Integrated Heat Exchanger and Internal Baffle (Example VI-7)

A blower located inside and at one end of the case induces air flow over the heat-producing components and discharges it through a shroud to the return air flow passage formed by a baffle concentric with the equipment case. The diameter of the baffle is 11 inches. This annular passage directs the air flow to the rear of the heat exchanger where its flow direction is turned 180 degrees, allowing the air to pass over the outside of the heat-producing components. The components' heat dissipation is transmitted to the internal circulating air by convective heat transfer over the component and baffle surfaces. Radiant heat transfer to the baffle surface is picked up by the internal air flow as it passes over the baffle surface. During bench test the external cooling air is forced through the inside of the heat exchanger tubes by a blower located in an auxiliary air flow apparatus. The external cooling air is discharged directly to the room.

Internal air temperature at the exit from the heat exchanger t_{it-2} is determined by averaging ten temperatures measured at equally spaced positions by a thermocouple probe inserted in the equipment in plane A-A. It is assumed that the air flow velocity over the tube bundle is uniform, so that the mean air temperature is defined by an average of the measured temperatures. Calculations indicate the temperature rise across the blower to be negligible. Thus, the temperature of the air leaving the component section is equal to that at the blower discharge. Five thermocouples are located in the air flow passage formed by the shroud at the blower discharge. They are equally spaced on an 8-inch diameter at B-B and the average of the indicated readings is taken as the mean air temperature at exit from the component section t_{it-3}. Internal air temperature at entrance to the heat exchanger t_{it-1} is determined from 10 thermocouple readings taken adjacent to the end plate of the equipment case at points immediately ahead of entrance to the tube bundle in plane C-C. The average case surface temperature is determined by 10 thermocouples equally spaced over the case surface. Temperatures of the external cooling air are measured at the inlet and exit of the heat exchanger by a single thermocouple located in the center of the inlet duct and a thermocouple probe located in the exit duct, positioned during test to give six readings. An average of the latter readings is assumed equal to the exit mean air temperature, since the air approaches the inlet header to the heat exchanger in an essentially uniform fashion.

Static pressure measurement of the external air flow is taken in the inlet and exit ducts several inches ahead and behind the heat exchanger. The rate of external air flow is metered by an auxiliary air flow apparatus.

Surface temperatures of 11 critical components are measured using the methods described in Chapter III.

b. Test Data

The following bench test data are reported:

Altitude of test location,	sea level
Temperature of room air and walls, t_o	20°C
Temperature of screens surrounding test area, t_w	20°C
Density of enclosed air in case sealed at room air temperature, ρ_o	0.0765 pound per cubic foot

Radiation environment (for use with Table VI-1):	
Surface of case	black painted
Temperature of radiation-receiving surfaces	20°C
Confinement	large room
Configuration of case, horizontal cylinder with closed ends. Length, L = 12 inches, diameter, D = 12 inches.	
Heat transfer areas of case	
Effective area for radiation, two ends plus cylindrical surface	680 square inches
Effective area for free convection	
Ends, vertical planes	226 square inches
Case, horizontal cylinder	454 square inches
Significant dimensions for free convection from case	
Cylindrical portion of case, L_{hc}	12 inches
Vertical ends of case, L_{vt}	9.42 inches
Heat dissipation to internal air (difference of electrical input and output)	800 watts
Temperature of external cooling air at inlet to heat exchanger, t_{e-1}	20°C
Ambient air pressure, p_o	14.7 pounds per square inch absolute

The air rates, average temperatures, and external cooling air pressure drops measured for the eight test runs are given in Table VI-4.

c. Reduction of Test Data

The calculational procedure for reducing the test data to a generalized form is illustrated in detail below for test run No. 1. Test data for other runs are reduced in the same manner.

The heat dissipated by the heat exchanger to the external cooling air is evaluated from the heat energy balance equation (VI-16). Using the data given in Table VI-4 for Run No. 1,

$$q_{ex} = 456\,(0.036)(56.8 - 20) = 604 \text{ watts.}$$

Since the total heat dissipation of the equipment is 800 watts, the heat loss from the equipment case is

$$q_c = 800 - 604 = 196 \text{ watts.}$$

The inlet temperature difference at the heat exchanger is

Table VI-4. Measured Air Rates, Temperatures, Pressure Drops, and Component Temperatures (Example VI-7)

Run No.	1	2	3	4	5	6	7	8
Air rate, W_e, pound per second	0.0360	0.0640	0.0920	0.119	0.150	0.183	0.200	0.211
External air temperature at heat exchanger exit, t_{e-2}, °C	56.8	41.8	35.5	32.2	29.9	28.2	27.6	27.2
Internal air temperature at heat exchanger inlet, t_{it-1}, °C	96.3	85.0	79.4	73.6	70.4	65.3	63.8	62.9
Internal air temperature at heat exchanger exit, t_{it-2}, °C	73.7	61.1	55.2	50.1	44.7	38.9	37.0	36.5
Internal air temperature at exit of component section, t_{it-3}, °C	103.6	91.2	85.0	79.9	75.1	69.5	67.9	66.9
Case surface temperature, t_c, °C	63.5	57.0	55.5	53.3	49.5	48.0	47.0	45.6
Component temperatures, °C								
No. 1	123.5	114.2	105.1	99.1	99.0	89.4	87.7	86.5
No. 2	138.7	126.7	119.0	118.0	110.8	103.0	101.8	101.7
No. 3	213.4	204.1	195.0	186.2	185.2	180.0	177.8	176.6
No. 4	103.5	90.6	85.4	78.0	75.5	69.4	67.8	66.6
No. 5	98.5	86.2	80.1	75.2	70.1	64.6	62.9	61.7
No. 6	183.6	171.2	162.0	160.0	158.2	150.3	151.2	146.7
No. 7	171.1	160.1	157.0	146.2	144.6	138.4	136.9	138.0
No. 8	153.6	144.2	135.0	132.1	125.2	122.5	117.8	116.7
No. 9	223.6	211.2	202.0	200.1	198.1	189.5	190.2	186.7
No. 10	83.6	70.3	69.0	62.1	55.1	51.6	50.9	48.6
No. 11	198.0	186.2	178.1	172.1	173.3	168.4	159.5	163.0
Static pressure of external cooling air ahead of heat exchanger, inches of water	0.075	0.19	0.34	0.53	0.80	1.11	1.30	1.42

$$t_{it-1} - t_{e-1} = 96.3 - 20 = 76.3^{\circ}C$$

Hence, the generalized parameter for the heat exchanger is

$$H_{ex} = 604/76.3 = 7.92 \text{ watts per } {}^{\circ}C$$

The circulation rate of the internal air is found from an internal heat balance, given by equation (VI-19), which gives for Run No. 1

$$W_{it} = 800/\ 456\ (103.6 - 73.7) = 0.0587 \text{ pound per second.}$$

The average temperature of the air passing through the component section is

$$t_{it-av} = (103.6 + 73.7)/2 = 88.6^{\circ}C$$

The free convective and radiant heat transfer from the case surface to the environment is calculated by procedures illustrated in Example VI-1, page 139. From Table VI-1, $\phi_1 = 0.95$. From Figure VI-3b, for the case temperature of $63.5^{\circ}C$ and an environmental temperature of $20^{\circ}C$, $\phi_2 = 0.203$ watt per square inch. Hence, q_c by radiation equals $0.95 \times 0.203 \times 680 = 132$ watts. Figure VI-2b is used for evaluation of free convective heat transfer. Using data of Run No. 1, q_c by free convection is 84 watts. Thus, $q_c = 132 + 84 = 216$ watts, as calculated, in comparison with 196 watts obtained from the heat balance. The correction factor F_c to be applied to the calculated case heat loss is, then, $F_c = 196/216 = 0.91$.

The average temperature differential for heat transfer from the return air to the case surface is

$$t_{it-b-av} - t_c = (103.6 + 96.3)/2 - 63.5 = 36.5^{\circ}C$$

Pressure drops across the external path of the heat exchanger are corrected to a common datum plane by multiplying actual pressure drop by the corresponding density ratio σ_1 evaluated at entrance to the exchanger. For Run No. 1, $\sigma_1 = 0.985$ (from Figure V-8) and $\sigma_1 \Delta p = 0.985\ (0.075) = 0.074$ inch water.

Following a similar procedure for all other test runs yields the reduced test data of Table VI-5.

These data are generalized for use in predicting thermal performance at other than bench test conditions by plotting (1) component temperatures versus average internal air temperature in the component space t_{it-av}, (2) $H_{ex} = q_{ex}/(t_{it-1} - t_{e-1})$ versus external air rate W_e, (3) corrected pressure drop of heat exchanger $\sigma_1 \Delta p$ versus external air rate W_e, and (4) case heat loss q_c versus $(t_{it-b-av} - t_c)$. These plots are shown in Figures VI-20, -21, -22, and -23, respectively. The correction factor F_c for calculation of case heat loss and the internal air rate W_{it} are assumed constant for thermal evaluation work and taken equal to the average of the values obtained from bench test. The average values are $F_c = 0.91$, and $W_{it} = 0.0580$ pound per second.

Table VI-5. Reduced Test Data (Example VI-7)

Run No.	1	2	3	4	5	6	7	8
External cooling air rate, W_e, pound per second	0.0360	0.0640	0.0920	0.119	0.150	0.183	0.200	0.211
Heat dissipation rate to heat exchanger, q_{ex}, watts	604	636	650	662	677	684	693	693
Heat dissipation rate from case, q_c, watts	196	164	150	138	123	116	107	107
Inlet temperature differential to heat exchanger, $(t_{it-1}-t_{e-1})$, °C	76.3	65.0	59.4	53.6	50.4	45.3	43.8	42.9
$q_{ex}/(t_{it-1}-t_{e-1})$, watts per °C	7.92	9.78	10.94	12.35	13.44	15.10	15.82	16.15
Internal air rate, W_{it}, pound per second	0.0587	0.0583	0.0589	0.0589	0.0577	0.0574	0.0568	0.0577
Average internal air temperature in component space, t_{it-av}, °C	88.6	76.1	70.1	64.0	59.9	54.2	52.5	51.7
$t_{it-b-av}-t_c$, °C	36.5	31.1	26.7	23.4	23.2	19.4	18.8	19.3
Correction factor, F_c, for calculated heat loss from case	0.91	0.93	0.90	0.89	0.91	0.92	0.89	0.94
Corrected heat exchanger pressure drop, $\sigma_1 \Delta p$, inch water	0.074	0.187	0.335	0.522	0.789	1.095	1.284	1.403

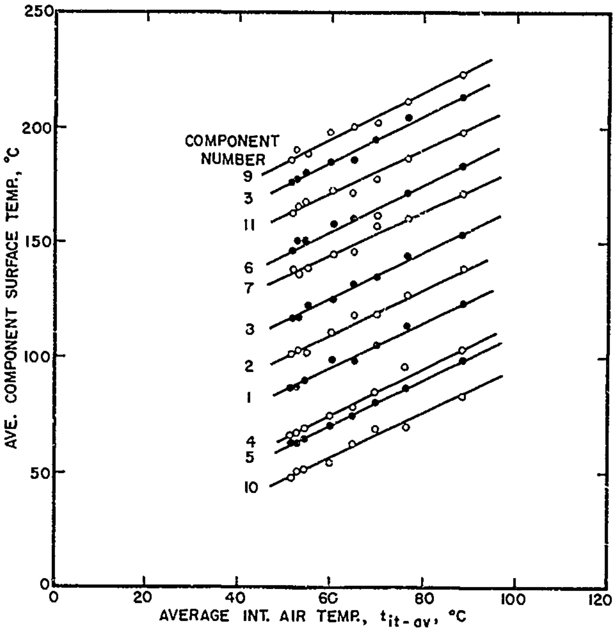


Figure VI-20. Component Temperature Trend Curves (Example VI-7)

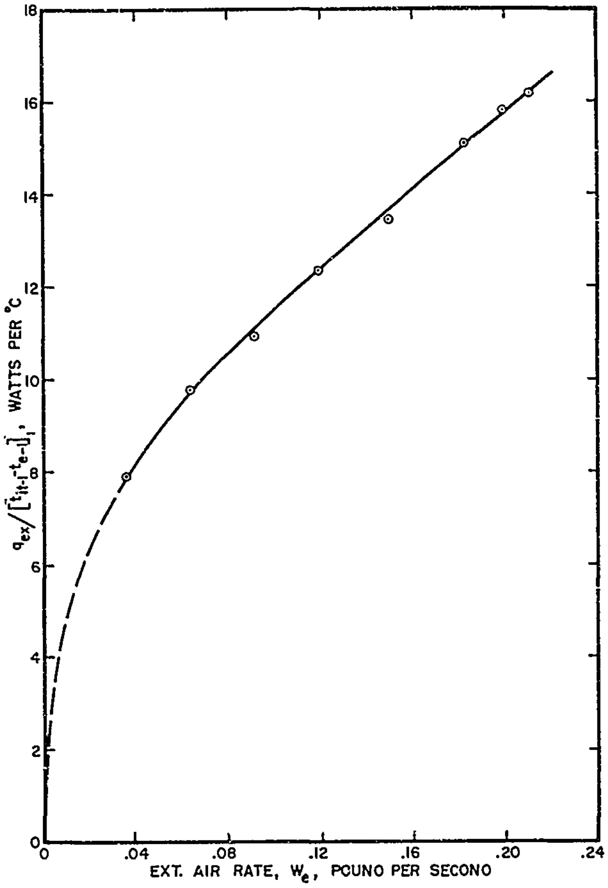


Figure VI-21. Heat Transfer Parameter of Heat Exchanger (Example VI-7)

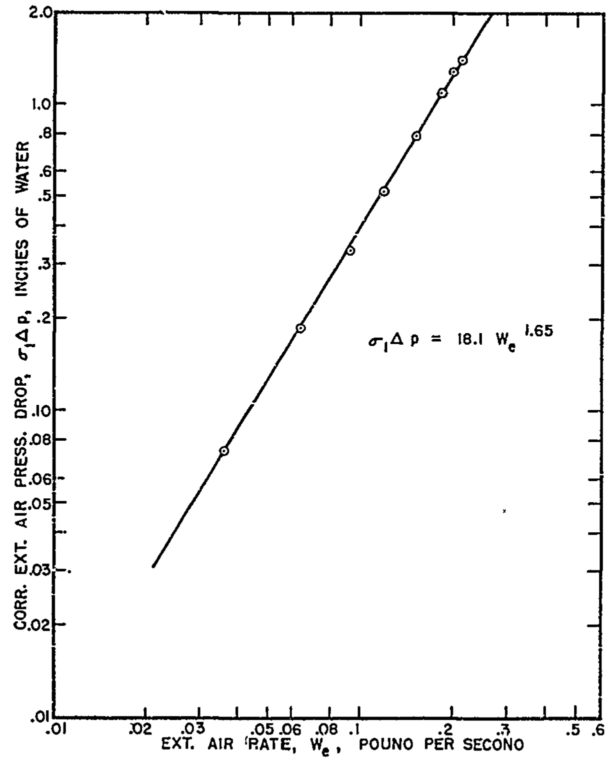


Figure VI-22. External Resistance Characteristics of Heat Exchanger (Example VI-7)

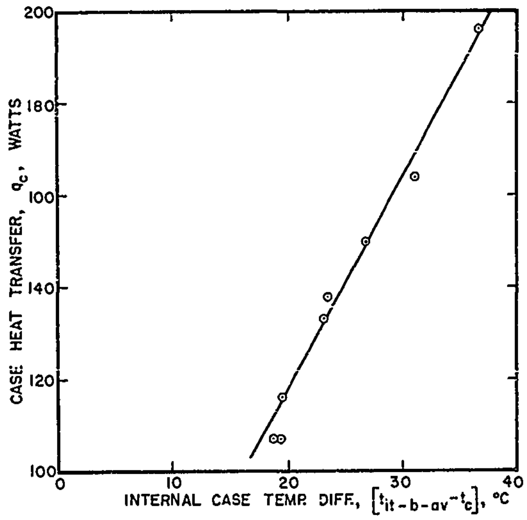


Figure VI-23. Case Heat Loss Characteristics of Unit (Example VI-7)

Example VI-8. Determination of Performance Specifications of Cooling Air Blower for a Pressurized Unit with Integrated Heat Exchanger and Internal Baffle

The pressurized unit with integrated heat exchanger described in Example VI-7 and shown schematically in Figure VI-19 is to be operated in a compartment of an aircraft at a pressure altitude of 50,000 feet and an ambient air and compartment wall temperature of $0^{\circ}C$. To avoid overheating of any components within the unit the average internal air temperature in the component space t_{it-av} is limited to $80^{\circ}C$. It is required to define the performance specifications of an external cooling air blower suitable for inducing sufficient flow of air through the heat exchanger that the specified average internal air temperature be maintained.

No special provisions are made for inducing air flow over the case, and the unit's view of compartment walls is essentially unobstructed. It is assumed that free convective and radiant heat transfer occur between the case surface and the environment. The case surface is unpolished Dural and the compartment walls are fabric-covered. Dimensions, flow passage arrangement, etc., are the same as those defined in Example VI-7.

A trial-and-error solution is required because of unknown case surface temperature and case heat loss. Known operational conditions are:

Compartment air pressure, p_o = 3.426 inches mercury absolute (found from Table A-I-2, page 324, for altitude of 50,000 feet)
Compartment air and wall temperature, $t_o = 0^{\circ}C$
Average internal air temperature in component space, $t_{it-av} = 80^{\circ}C$
Gross heat dissipation, q = 800 watts

From Example VI-7,

Correction factor for calculated case heat loss, $F_c = 0.91$
Internal air flow rate, W_{it} = 0.0580 pound per second

From Table VI-1, page 136,

Radiation factor for case surface, $\phi_1 = 0.30$

The temperature rise of the air across the component space is by equation (VI-19)

$$t_{it-3} - t_{it-2} = 800/(456 \times 0.018) = 30.3^{\circ}C$$

Since the average internal air temperature in the component space is fixed at $80^{\circ}C$, then the temperatures at entrance and exit of the component space are

$$t_{it-2} = 80 - 30.3/2 = 64.85^{\circ}C$$

$$t_{it-3} = 80 + 30.3/2 = 95.15^{\circ}C$$

Next, an assumption of case temperature is required. Say, $t_c = 74^oC$.

From Figure VI-3b, for $t_c = 74^oC$ and $t_o = 0^oC$, $\phi_2 = 0.334$ watt per square inch
From Example VI-7, surface area for radiant heat transfer = 680 square inches
Calculated radiant heat transfer from case = $\phi_1 \phi_2 S = 0.3 \times 0.334 \times 680 = q_{rd} = 68.1$ watts
From Figure VI-2b, for $t_c = 74^oC$, $t_o = 0^oC$, and $p_o = 3.426$ inches mercury, free convective unit heat transfer rate for cylindrical portion of case = 0.072 watt per square inch,
for ends of case = 0.098 watt per square inch
From Example VI-7, surface area for free convective heat transfer, cylindrical portion = 454 square inches, ends = 226 square inches
Calculated free convective heat transfer = $0.072 \times 454 + 0.098 \times 226 = q_{cv} = 54.8$ watts
$q_c = F_c (q_{cv} + q_{rd}) = 0.91 (68.1 + 54.8) = 111.8$ watts

If the assumed value of $t_c = 74^oC$ is correct, the heat loss from the case surface should be equal to the heat loss of the internal air in the return air flow passage. From equation (VI-19), using the calculated value of q_c = 111.8 watts and W_{it} = 0.0580 pound per second, the temperature change of the internal air in the baffle passage is

$$t_{it-3} - t_{it-1} = 111.8/(456 \times 0.058) = 4.22^oC$$

$$t_{it-1} = t_{it-3} - 4.22 = 95.15 - 4.22 = 90.93^oC$$

$$t_{it-b-av} = 95.15 - (4.22)/2 = 93.04^oC$$

$$t_{it-b-av} - t_c = 93.04 - 74 = 19.04^oC.$$

Corresponding to this last temperature differential, Figure VI-23, containing the case heat loss characteristics of the unit developed in Example VI-7 from bench test data, gives q_c = 113.5 watts. This shows that for the assumed case surface temperature of 74°C the calculated heat loss from the internal air to the case surface exceeds the calculated heat loss from the case to the environment by 1.7 watts. Thus, the correct equilibrium case temperature would be slightly above the assumed value. In general, agreement of values for q_c within 2 per cent would be good enough. In this example, a second series of calculations gives a slight change of the equilibrium case temperature to $t_c = 74.3^oC$, the case heat loss to q_c = 112.0 watts and the internal air temperature at entrance to the integrated heat exchanger to $t_{it-1} = 90.9^oC$.

Since, with induced flow the blower is located in the discharge duct of the external air path, the temperature of the external air at entrance to the heat exchanger t_{e-1} is equal to the ambient air temperature $t_o = 0^oC$. Consequently

$$t_{it-1} - t_{e-1} = 90.9 - 0 = 90.9^oC.$$

Also,

$$q_{ex} = 800 - q_c = 800 - 112 = 688 \text{ watts,}$$

and

$$q_{ex}/(t_{it-1} - t_{e-1}) = 688/90.9 = 7.57 \text{ watts per } ^\circ C$$

For this value Figure VI-21, containing the heat transfer characteristics of the heat exchanger developed from bench test data in Example VI-7, the required external air rate is found to be $W_e = 0.033$ pound per second.

From Figure VI-22 developed in Example VI-7, the corrected pressure drop of the external air across the heat exchanger for $W_e = 0.033$ is $\sigma_1 \Delta p = 0.065$ inch water. Since $\sigma_1 = \sigma_o$, then from Figure V-8, for $p_c = 3.426$ inches mercury and $t_o = 0^\circ C$, $\sigma_o = \sigma_1 = 0.121$, and $\Delta p = 0.065/0.121 = 0.537$ inch of water. This is the static pressure rise the blower must produce to create the required air flow through the heat exchanger. At entrance to the blower,

$$p_i = p_2 = p_1 - \Delta p = 3.426 - 0.537/13.55 = 3.39 \text{ inches mercury}$$

$$t_i = t_2 = t_1 + q_{ex}/(456\ W_e) = 0 + 688/(456 \times 0.033) = 45.7^\circ C$$

$$\sigma_i = \sigma_2 = 0.101 \quad \text{(from Figure V-8)}$$

The required volumetric capacity of the blower is, by equation (V-1)

$$Q_B = 784 \times 0.033/0.101 = 256 \text{ cubic feet per minute}$$

The required corrected static pressure rise of the blower is

$$\Delta p/\sigma_i = 0.537/0.101 = 5.31 \text{ inches of water}$$

Hence a blower must be selected having a static pressure producing ability of 5.31 inches of water at a volumetric capacity of 256 cubic feet per minute when the inlet density to the blower is standard, i.e., $\rho_i = 0.0765$ pound per cubic foot ($\sigma_i = 1$). Assuming a static blower efficiency of 50 per cent, the horsepower input to the blower required at the operating point at 50,000 feet altitude is

$$hp = Q\ \Delta p/(6350\ \eta_s) = 255 \times 5.31/(6350 \times 0.5) = 0.0427$$

This would require an electrical input to the drive motor on the order of 100 watts.

When case heat transfer is to the environment, any increase in the case heat loss aids in reducing the size and power requirements of the external blower. In this example the case heat loss can be increased by painting the case surface black. From Table VI-1, for a black-painted case, fabric-covered walls, and intermediate confinement, $\phi_1 = 0.93$. Following the procedure illustrated in the example, it is found that the case surface temperature drops from 74.3° to 59°C, the case heat loss increases from 112 to 178 watts, and the external air rate drops from 0.033 to 0.027 pound per second. The pressure drop of the air across the heat exchanger is 0.386 inch of water. The blower must handle 210 cubic feet of air per minute at a corrected static pressure rise of 3.8 inches water. Power input to the blower drops from 0.044 to 0.025 horsepower.

Example VI-9. Determination of Operating Surface Temperatures of Internal Components in a Pressurized Unit with Integrated Heat Exchanger and Cooling Air Blower of Known Performance Characteristics

The unit whose description and bench-test performance are given in Example VI-7 is operated in a compartment having a pressure altitude of 45,000 feet, p_o = 4.356 inches mercury absolute. The temperature of the air and surrounding walls is 20°C. The performance characteristics of the blower supplying the external cooling air by forced flow to the integrated heat exchanger are known, so that the volumetric capacity, pressure drop, and power required may be directly evaluated. It is required to determine the operating temperatures of the various internal components for this condition of operation.

For the performance characteristics of the external blower and by the procedure illustrated in detail in Example VI-5, for forced flow, the following data are determined.

Volumetric capacity of blower at intake, Q = 411 cubic feet per minute
Air rate delivered by blower, W_e = 0.075 pound per second
Static pressure rise of air across blower, Δp = 1.765 inches water
Corrected static pressure rise of blower, $\Delta p/\sigma_i$ = 12.33 inches water
Efficiency of blower, η_s = 56 per cent
Power required to drive blower, hp = 0.204 horsepower
Temperature rise of air across blower, Δt_B = 4.5°C

The heat dissipation in the component space is 800 watts and the equipment case is painted black.

The temperature of the external cooling air at entrance to the integrated heat exchanger is $t_{e-1} = t_o + \Delta t_B = 20 + 4.5 = 24.5$°C. From Figure VI-21 of Example VI-7, $q_{ex}/(t_{it-1}-t_{e-1}) = 10.28$ for W_e = 0.075 pound per second.

At this point in the analysis an assumption of the case heat loss is required. Let q_c = 150 watts. Hence,

$$q_{ex} = 800 - q_c = 800 - 150 = 650 \text{ watts},$$

and

$$t_{it-1} - t_{e-1} = 650/10.28 = 63.2^\circ C.$$

The temperature of the internal air at entrance to the integrated heat exchanger is $t_{it-1} = t_{e-1} + 63.2 = 24.5 + 63.2 = 87.7$°C. The temperature of the internal air leaving the heat exchanger is, then

$$t_{it-2} = t_{it-1} - q_{ex}/456\, W_{it} = 87.7 - 650/(456 \times 0.058) = 63.1^\circ C,$$

since the internal air rate W_{it} remains equal to the bench test value of 0.058 pound per second (see Example VI-7). The temperature of the internal air leaving the component space is

$$t_{it-3} = t_{it-2} + q/456\ W_{it} = 63.1 + 800/(456 \times 0.058) = 93.4^{\circ}C.$$

The average temperature of the internal air passing through the component space is $t_{it-av} = 63.1 + 15.2 = 78.3^{\circ}C$.

The next step is to evaluate the case heat loss to check the assumed value of 150 watts. The magnitude of the areas, significant dimensions, etc. for free convective and radiant heat transfer are listed in Example VI-7, page 186. From Table VI-1, $\phi_1 = 0.93$. Assume $t_c = 65^{\circ}C$. From Figure VI-3b, $\phi_2 = 0.212$, hence, radiant heat transfer $q_{rd} = F_c\phi_1\phi_2 S = 0.91 \times 0.93 \times 0.212 \times 680 = 121.9$ watts. From Figure VI-2b, for the cylindrical portion of the case, watt per square inch by free convection = 0.042, for ends of case, watt per square inch = 0.059, total free convective heat transfer $q_{cv} = 0.91(0.042 \times 454 + 0.059 \times 226) = 29.5$ watts. Case heat loss $q_c = 29.5 + 121.9 = 151.4$ watts. The average temperature of the air in the return air flow passage $t_{it-b-av} = 0.5(93.4 + 87.7) = 90.5^{\circ}C$ and $t_{it-b-av} - t_c = 25.5^{\circ}C$. Therefore, from Figure VI-23, the heat loss from the return air flow to the case equals 143 watts. Since the inside and outside case heat transfer rates do not agree, another assumption of case surface temperature is required. A second trial gives a balance at $t_c = 64^{\circ}C$ and $q_c = 147.5$ watts. This heat loss of 147.5 watts differs slightly from the originally assumed value of 150 watts. However, the values check within less than 2 per cent and, therefore, the internal air temperature $t_{it-av} = 78.3^{\circ}C$ may be assumed as correct. A greater difference would necessitate repetition of the procedure based on a new assumption of q_c equal to the calculated value.

The component temperatures are found from the trend curves established from bench test data and the calculated average internal air temperature of $78.3^{\circ}C$. The trend curves for this particular equipment are given in Figure VI-20, Example VI-7. For example, the surface temperature of component (6) would be, according to its trend curve, 175 C when $t_{it-av} = 78.3^{\circ}C$.

Vented Units with Closed Case Cooled by Free Convection and Radiation

In this type unit, as discussed in Chapter II, page 11, the components operate in an environment which is at ambient atmospheric pressure. The construction of the case permits pressure equalization with the ambient atmosphere. The internal and external heat transfer mechanisms of this type unit are shown schematically in Figure VI-24. Generally speaking, the heat transfer modes of the unit resemble those of the pressurized unit cooled externally by free convection and radiation, except that the internal heat transfer system is variable with altitude while that of the pressurized unit is essentially constant. Also, the configuration of the case is usually different, being rectangular prismatic, while pressurized units most frequently have cylindrical cases. The heat transfer patterns of both types are essentially the same, except that vented units are not likely to have forced convection internally. Heat is being dissipated from the components to the case by convection, radiation and conduction, and subsequently from the case to the at-

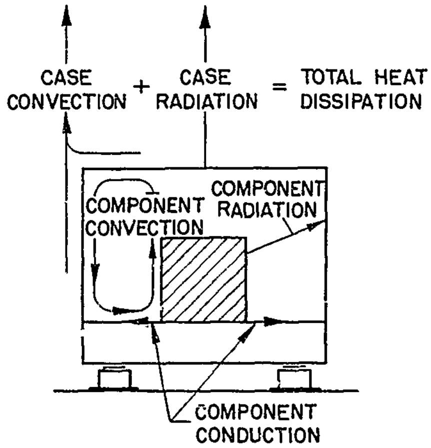


Figure VI-24. Heat Transfer Diagram of Vented Unit with Closed Case Cooled by Free Convection and Radiation

mosphere principally by free convection and radiation. Consequently the determination of the case temperature under operating conditions, by calculations based on bench-test data, is the same for both types. The methods and principles of analysis are discussed on pages 132 to 138, and are illustrated in Examples VI-1, -2 and -3 with reference to the pressurized unit.

1. Determination of Component Temperatures

On the basis of bench-test data alone, component temperatures at operational conditions different from test conditions can only be calculated approximately, because the internal heat transfer pattern is extremely difficult to analyze and to break up into the three contributing modes, i.e., conduction, convection, and radiation. While conduction and radiation are only affected by the case temperature and the component temperature, free convection is also affected by the internal pressure of the unit which is variable with environmental altitude conditions. Therefore, the extent to which each mode participates in the total heat dissipation from each component is also variable with environmental altitude conditions.

Since the closed vented unit has usually a relatively moderate rate of total heat dissipation, complete thermal evaluation for the operation of every component is frequently not necessary. If required, it can only be made on the basis of altitude chamber test data, as discussed in Chapter VII. Bench test data provide sufficient information which can be made the basis of approximate calculations to ascertain whether certain critical components may undergo changes in temperature which may result in their failure under extended operational conditions.

It is usually of importance to determine the surface temperatures of three types of components under extended operational conditions. They are (1) high-temperature components like tubes surrounded by other high-temperature components, (2) high-temperature components like transformers surrounded by other high-temperature components, and (3) low-temperature components like condensers surrounded by high-temperature components. For the first two types it is necessary to ascertain whether conditions would be such that they can dissipate their heat without reaching excessively high surface temperatures. The third type must be studied for the effect which the surrounding high-temperature components may have in causing an increase of surface temperature due to reduced convection under operation conditions encountered in flight.

A high-temperature component like a tube surrounded by other high-temperature components is capable of dissipating heat by radiation only from a relatively small portion of its surface which is usually the top. Because of the surrounding similar components, no temperature gradient may exist in the chassis, thus eliminating heat dissipation by conduction. Therefore, the principal mode of heat transfer is convection. Free convective heat transfer is quite difficult to evaluate for a component in this position and, therefore, it is best to estimate by calculation the heat transfer from the portion of the surface capable of dissipating heat by radiation, and to determine then the convective heat transfer by difference. If this estimate is made for bench test conditions, using the component and case temperatures determined under these conditions, a distribution of heat dissipation between free convection and radiation can be determined. Under other operational conditions, such as at higher altitude, the distribution will change in favor of radiation since convective heat transfer is impeded with increase in altitude. Thus, if the case temperature at the higher altitude is known, calculations must be made to establish the resulting surface temperature of the component on basis of a heat balance between the component and the case and the internal atmosphere of the unit. The normal procedure would be to assume a vastly increased percentage of radiant heat transfer from the component to the case and to calculate for this partial rate of heat dissipation the required surface temperature of the component. Then, the convective heat transfer from the component may be determined on basis of this calculated surface temperature and a local internal air temperature, estimated as the average of the component surface temperature and the known case temperature. The result of this calculation added to the originally assumed rate of radiant heat transfer must equal the known rate of heat dissipation of the component. Thus, by successive approximation, an approximate component temperature can be determined for which this calculated heat balance is obtained. The procedure is based on the assumption that the radiant heat dissipation calculated at test conditions for the part of the surface of the component in interchange with the other surrounding high-temperature components remains constant with changed altitude conditions since the other high-temperature components are also assumed to increase in temperature. The procedure utilized in making these calculations for individual components is illustrated in Example VI-10. Figures VI-2 and VI-3 and Table VI-1, discussed previously, are used. The procedure is only applicable to tubes having negligible heat transfer by conduction.

The method of analysis for high-temperature components such as transformers is quite different since their principal mode of heat dissipation is conduction. This makes calculation relatively simple since the rise of case temperature over test conditions is also indicative of the temperature rise of the transformer. Radiant heat dissipation from transformers would be low under bench test conditions. A relatively small percentage of heat dissipation may occur by free convection. This free convective heat transfer would be practically eliminated under high-altitude conditions and, therefore, the heat transfer by conduction would have to be increased by the quantity of heat dissipation previously attributed to free convection. Normally, the analysis of the distribution of heat dissipation between free convection and conduction would not be necessary. It is sufficient to adjust the temperature difference between the transformer surface and that of the case, measured under test conditions, by increasing it about 10 to 20 per cent to compensate for the convective cooling which would be practically eliminated at high altitude.

Low-temperature components such as condensers, when surrounded by high-temperature components may be subjected to critical conditions under high-altitude operation. This may be due to the fact that the internal air temperature of the equipment would increase beyond the permissible limit of the component, or that the component would be subject to large conductive or radiant heat gain from surrounding hot components. For purposes of analysis, approximations must be made on the basis of assumptions which provide for heat gain of the low-temperature component solely by radiation and dissipation of heat from the component solely by free convection. This assumes that the thermal conductivity of the low-temperature component's material is low and little conductive heat transfer occurs. The assumption is conservative and will tend to yield a high component temperature if conductive cooling exists. The methods of analysis are shown in Example VI-11. They are not applicable when heat gain by conduction is likely to occur.

If a low-temperature component gains heat by conduction and radiation from surrounding high-temperature components, the final operating temperature would be determined by a heat balance between these two modes of heat transfer and free convection. The calculation of conductive heat transfer is complex, and for most components only a qualitative estimate can be made.

In all computational methods described above, the heat dissipations by free convection and radiation need only be calculated under test conditions. For the determination of component temperatures at extended operational conditions proportionality factors can be applied for free convection as well as radiation. As far as free convective heat transfer is concerned, the dimensional characteristic L and the configurational characteristic C are unchanged. Therefore, free convective heat transfer will vary for any given component as the value of ϕ_3 determined from the use of the upper and lower right quadrants in Figure VI-2. For radiant heat transfer, the configuration and the emissivity function ϕ_1 also remains constant and, therefore, the radiant heat transfer for any particular component under changed operating conditions will vary as the temperature function ϕ_2. Use of these relationships simplifies calculations considerably. The mentioned Examples VI-10 and -11 illustrate the procedures. It should also be noted that in the examples the local air temperature within the unit in the vicinity of the component is assumed to be the average of the case temperature and the component temperature. It should be recognized that this is not absolutely correct but will represent a fair approximation for each component analyzed. Accurate calculation of the internal air temperature distribution under extended operational conditions is not possible.

2. Examples

The following examples are intended to illustrate the procedures for the determination of the operational thermal conditions of closed vented equipment cooled internally and externally by free convection and radiation.

Example VI-10. Determination of Operational Surface Temperature of High-Temperature Component, Surrounded by Other High-Temperature Components in a Closed Vented Unit

A closed vented unit is bench tested to determine the operational thermal characteristics of a tube mounted vertically in the chassis and surrounded by similar high-temperature components. The component is cooled by internal radiation, free convection and conduction, and the equipment is cooled by radiation and free convection from the equipment case surface. The position of the component (A) to be studied relative to the equipment chassis and case and relative to the other high-temperature components is illustrated in Figure VI-25.

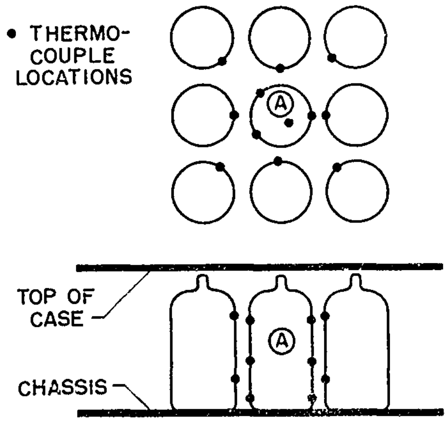


Figure VI-25. Relative Locations of High-Temperature Components in Closed Vented Unit (Example VI-10)

Part A. Analysis of Bench Test Data

Bench Test Data

Measured heat dissipation of tube,	6.0 watts
Dimensions of tube	
diameter,	1.0 inch
height,	2.0 inches

surface area of top of tube,	0.785 square inch
surface area of cylindrical part of tube,	6.28 square inches
Average measured temperature of tube surface as obtained from several thermocouple readings,	197°C
Average measured temperature of surrounding components,	174°C
Surface of case inside and outside, black-painted average measured temperature,	55°C

Calculation of Radiation to Surrounding Components

From Table VI-1	$\phi_1 = 0.90$
From Figure VI-3a, for tube surface temperature of 197°C and surrounding surface temperature of 174°C,	$\phi_2 = 0.33$
Heat transfer by radiation to surrounding components, (0.9 x 0.33 x 6.28)	1.87 watts

Calculation of Radiation to Case and Chassis

From Figure VI-3a, for tube surface temperature 197°C and case surface temperature of 55°C	$\phi_2 = 1.396$
Calculated radiation to case and chassis based on radiation from top surface of tube to case and an arbitrary increase of 50 per cent to account for radiation from cylindrical tube surface to case and chassis, therefore, (0.9 x 1.396 x 0.785)1.5	1.48 watts
Total calculated radiation, (1.87 + 1.48)	3.35 watts

Free Convection

Free convection, determined by difference between measured heat dissipation and calculated radiation, (6.0 - 3.35)	2.65 watts

Heat Balance

Source	Watts	Per cent total dissipation
Radiation to surrounding components	1.87	31.2
Radiation to case and chassis	1.48	24.7
Free convection	2.65	44.1
	6.00	100.0

Determination of Free Convection Proportionality Factor

average surface temperature of component,	197°C
average case temperature,	55°C
Average local internal air temperature in vicinity of tube, (average of average case temperature and average component surface temperature), (55+197)/2	126°C

From Figure VI-2a, using first two quadrants, proportionality factor for free convective heat transfer,	$\phi_3 = 17.3$

Part B. Determination of Thermal Performance Altitude of 40,000 Feet

The determination of the component temperature at altitude is based on the assumption that the radiant heat dissipation to adjacent components remains constant and that the decrease in convective heat transfer at altitude is overcome by an increase in radiation from the components to the chassis and case. By use of the same methods as outlined in the discussion on closed pressurized equipment cooled by free convection and radiation and Examples VI-1 and -2, the case surface temperature at 40,000 feet altitude is calculated to be 84°C.

Component Surface Temperature

The free convective heat transfer is expected to decrease. As a first trial, it is assumed as 50 per cent of the bench test value, (0.5 x 2.65)	1.33 watts
By heat balance, the total radiation must then be, (6.0 - 1.33)	4.67 watts
Based on assumption of constant radiation to adjacent components, the radiation to case and chassis is, (4.67 - 1.87)	2.80 watts
As in Part A, the radiation to the chassis and case assumed as 1.50-times the radiation from the top of the tube, $\phi_2 = 2.8/(0.9 \times 0.785 \times 1.5)$	$\phi_2 = 2.64$
From Figure VI-3a, for $\phi_2 = 2.64$ and case surface temperature of 84°C, find component temperature,	270.5°C
Average local internal air temperature, (270+84)/2,	177°C
From Figure VI-2a, for above temperature, proportionality factor at altitude,	$\phi_3 = 9.8$
By proportion, based on calculated free convective heat dissipation under bench test conditions, free convection at altitude, 2.65(9.8/17.3)	1.50 watts

Heat Balance

Radiation to surrounding components,	1.87 watts
Radiation to case and chassis,	2.8 watts
Free convection,	1.50 watts
Total calculated heat dissipation	6.17 watts
Measured heat dissipation	6.00 watts

Since heat balance between calculated and measured heat dissipation is not obtained, re-calculation of the component surface temperature is required. The temperature was too great. Therefore, as next trial, ϕ_2 is reduced in the ratio of the measured to the total calculated heat dissipation. Thus, $\phi_2 = 2.64(6.00/6.17)$. For the corresponding component surface temperature of 265°C the following heat balance is obtained.

radiation to surrounding components,	1.87 watts
radiation to case and chassis,	2.66 watts
free convection,	1.46 watts
total calculated heat dissipation,	5.99 watts
measured heat dissipation,	6.00 watts

Agreement between the measured and calculated values is sufficiently close and the approximate component temperature is 265°C under the specified altitude conditions.

Example VI-11. Determination of Operational Surface Temperature of Low-Temperature Component Mounted Adjacent to a High-Temperature Component in a Closed Vented Unit

The location of a condenser relative to a tube is shown schematically in Figure VI-26. The condenser is assumed to be non-heat producing and of a material having low thermal conductivity.

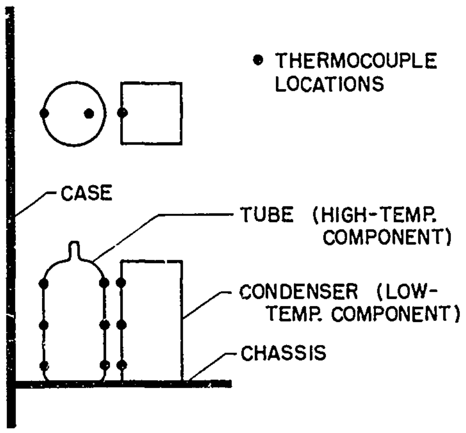


Figure VI-26. Relative Locations of Case and High- and Low-Temperature Components in Closed Vented Unit (Example VI-11)

Part A. Analysis of Bench Test Data for High-Temperature Component (Tube)

Bench Test Data

Measured heat dissipation	10.0 watts
Dimensions	
diameter,	1.0 inch
height,	2.0 inches
surface area of top,	0.785 square inch
surface area of cylindrical part	6.28 square inches
Average surface temperature, determined from several thermocouple readings,	188°C
Surface of case	
inside and outside, black-painted	
average measured temperature	55°C

Radiation from Tube

From Table VI-1, neglecting the effect of the adjacent low-temperature component,	$\phi_1 = 0.90$
From Figure VI-3a, for 188°C and 55°C,	$\phi_2 = 1.26$
Surface area, (6.28 + 0.785)	7.07 square inches
Total radiant heat transfer, (0.9 x 1.26 x 7.07)	8.02 watts

Free Convection from Tube

Free convection, by difference (10.0 - 8.02)	1.98 watts
Average local internal air temperature near tube, (55 + 188)/2	121.5°C
From Figure VI-2a, using first two quadrants, proportionality factor for free convective heat transfer,	$\phi_3 = 15.6$

Part B. Temperature of High-Temperature Component at Altitude of 40,000 Feet

By the procedures outlined in Examples VI-1 and -2 on pressurized equipment cooled externally by free convection and radiation, the case surface temperature at 40,000 feet altitude is calculated to be 84°C.

Based on a first assumption that cooling by free convection at 40,000 feet altitude would be 50 per cent of bench test value, free convective heat dissipation, (0.5 x 1.98)	0.99 watt
Radiation, by difference, (10.0 - 0.99)	9.01 watts
radiation factor, $\phi_2 = 9.01/(0.9 \times 7.07)$	$\phi_2 = 1.415$
From Figure VI-3a, for case surface temperature of 84°C and $\phi_2 = 1.415$, component surface temperature,	210°C
Average local internal air temperature near tube, (85 + 210)/2,	147.5°C
From Figure VI-2a, using first two quadrants, proportionality factor,	$\phi_3 = 6.1$
By proportion, based on calculated free convection and proportionality factor under bench test conditions, free convection at altitude, 1.98(6.1/15.6)	0.77 watt

Heat Balance

Radiation,	9.01 watts
Free convection,	0.77 watts
Total calculated heat dissipation,	9.78 watts
Measured heat dissipation,	10 watts

Following the same procedure the heat balance is found to check at a surface temperature of 211°C, which may be accepted as the approximate value of the tube's surface temperature at 40,000 feet altitude.

Part C. Temperature of Adjacent Low-Temperature Component (Condenser) at Altitude of 40,000 Feet

From bench test data,	
average measured surface temperature of component,	120°C
average local internal air temperature near condenser, (120 + 55)/2	87.5°C
From Figure VI-2a, convective proportionality factor,	$\phi_3 = 6.8$
From Figure VI-3a, for high-temperature component at 188°C and low-temperature component at 120°C, radiation proportionality factor,	$\phi_2 = 0.797$

The radiant heat transfer from the high-temperature component to the low-temperature component is assumed to be dissipated from the low-temperature component by free convection. Therefore, the free convection from the low-temperature component is proportional to 0.797. If for the first trial it is assumed that the free convective heat transfer from the low-temperature component at 40,000-feet altitude is 50 per cent of the bench test value, the radiation proportionality factor at altitude $\phi_2 = 0.5 \times 0.797 = 0.399$. At 40,000-feet altitude, the surface temperature of the tube is 211°C, determined in part B. Then from Figure VI-3a, for 211°C and $\phi_2 = 0.399$, the low-temperature component has a temperature of 185°C.

Checking assumed free convective heat dissipation,	
case temperature 40,000 feet altitude,	84°C
average local internal air temperature near condenser, (185 + 84)/2	134.5°C
from Figure VI-2a, free convective heat transfer proportionality factor,	$\phi_3 = 5.0$
calculated percentage of free convective heat transfer based on bench test conditions, (5.0/6.8)100	73.5

This is not in agreement with the assumed percentage of 50. Hence, a second trial is required. By repeating the process it is found that free convection at 40,000 feet is equal to 65 per cent of that under bench test conditions, corresponding to a surface temperature of 176.5°C for the low-temperature component. The following summary of calculated temperatures in °C is obtained:

	Bench Test	40,000-Feet Altitude	Temperature Increase
Case surface	55	84	29
High-temperature component	188	211	23
Low-temperature component	120	176.5	56.5

Vented Units with Open Case and Through-Flow of Atmospheric Air by Natural Convection

The heat transfer pattern of this type unit, as discussed in Chapter II, page 12, is complex and cannot be analyzed on the basis of bench test data alone. The heat transfer diagram in Figure VI-27 summarizes the various

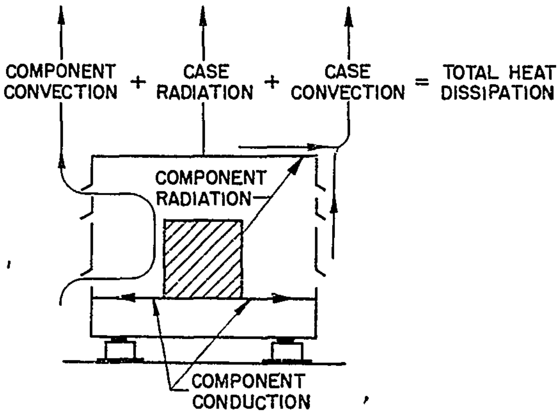


Figure VI-27. Heat Transfer Diagram of Vented Unit with Open Case and Through-Flow of Atmospheric Air by Natural Convection

participating modes of heat transfer, as described on page 12. The principal difficulty arises in the evaluation of direct convection from individual components to atmospheric air. An approximate method of analysis to determine the general component temperature level at any operational condition is outlined in Chapter VIII, page 304. However, for its utilization bench-test data alone are not sufficient and at least two altitude chamber tests are required. This method is limited to the determination of an average component temperature and does not give information on the exact variation of surface temperatures of individual components with changes in operational environment. This latter data can only be obtained by altitude chamber tests at the operational ambient temperature and pressure. In such tests, the procedures and precautions discussed in Chapter IV, page 68, must be followed if reliable thermal data are desired.

Vented Units with Open Case and Forced or Induced Through-Flow of Atmospheric Air

Units of this type in which air flow is produced from the environment over component surfaces are normally not operated so as to be subject to electrical malfunctioning due to low internal air pressure. Also, the unit's heat transfer surfaces are not extensive enough to dissipate the generated heat by radiation and free convection at a temperature level permissible for the safe operation of individual electronic components. Cooling by through-flow of air is direct, allowing heat dissipation to the air without passage of heat through secondary heat transfer surfaces, such as equipment cases or heat exchangers which unavoidably introduce additional thermal resistance. Therefore, this air cooling method is superior to all others from the standpoint of least quantity of air and pumping power required.

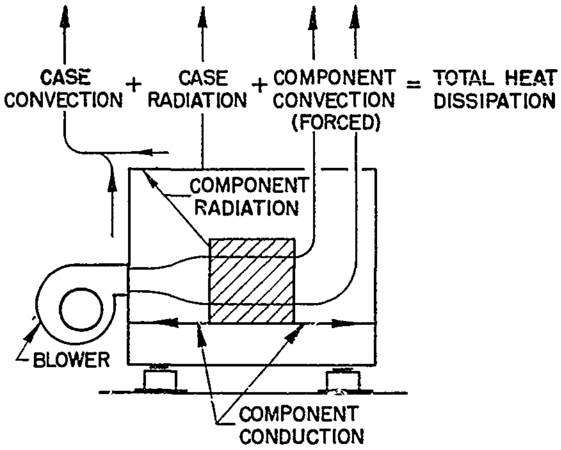


Figure VI-28. Heat Transfer Diagram of Vented Unit with Open Case and Forced Through-Flow of Atmospheric Air

The various modes of heat transfer contributing to dissipation of heat generated by the unit when cooled by this method are illustrated in Figure VI-28. The principal mode of heat transfer from the components is forced convection, created by air flow produced by a blower or ram action. In addition, should the temperature of the unit's case be lower than the average temperature level of the components, heat is transferred from the components to the case by radiation directly and by conduction through the chassis. The case surface dissipates heat to the environment by convection and radiation. The rate of total heat dissipation equals the sum of that given to the through-flow of air and that dissipated from the surface of the case. Heat gain through the equipment case occurs when environmental temperatures exceed the case surface temperature. This heat gain must be dissipated by the through-flow of air, in addition to the heat generated by the components.

Units of this type have, in general, induced flow and the case so designed as to admit air through multiple openings in side panels and/or in one end. Air discharge occurs through a single outlet in the end opposite that to which the blower is connected. Other equipments may employ forced flow of air through a single inlet, with the heated air discharging through a single outlet or several louvered openings in the side panels and/or the end of the case. The air flow pattern obtained with multiple discharge openings in side panels is not the most effective for cooling by through-flow of air. Frequently, it is the result of remedial application of blowers to equipments originally intended to be cooled by free convection and radiation only. It may also be resorted to when the design of the unit does not permit internal location of the blower and modification for inducing through-flow.

Evaluation procedures discussed in subsequent paragraphs apply specifically to units having cases with closed side panels, admitting and discharging air through opposite ends, or to those discharging the air through a single outlet but having air flow induced through multiple openings in end and side panels of the case. Evaluation procedures for equipments having flow discharging through multiple openings in the case are discussed separately on pages 214 to 215.

1. Forced Convection Heat Transfer to Through-Flow of Air

Of the total heat q dissipated by an equipment, the major portion q_a is absorbed by the through-flow of air, while the remainder q_c is dissipated through the case surface to the environment. A heat balance for the equipment is then

$$q = q_a + q_c \tag{VI-21}$$

The case heat transfer q_c becomes negative when heat gain from the environment occurs.

Air flowing through an open unit comes in contact with many geometrically dissimilar components having different surface temperatures. Thus, a complex heat transfer pattern exists which cannot be analyzed in detail by conventional methods. However, it may be postulated that a rational analysis of the heat transfer process is possible by introducing a new variable, referred to as the effective temperature of the components. It is assumed that this temperature may be defined from individually measured component temperatures so as to allow simulation of the actual heat transfer process by means of a surface having a uniform temperature, equal to the effective temperature t_{ef} and dissipating heat to the through-flow of air.

The use of an effective component temperature for evaluation purposes permits two possible procedures for generalization of bench test data. The working chart in Figure VI-7 for closed units with case cooled by forced convection, could be employed to generalize the heat transfer characteristics by substituting the effective temperature t_{ef} for the case surface temperature t_c designated in the chart, and using the heat dissipation to the through-flow of air q_a for the heat dissipation. This method requires definition of a constant exponent n. Undoubtedly, this characteristic would not be exhibited by all units cooled by through-flow of air because the nature of the heat transfer processes of the individual components may differ appreciably.

A working method not requiring definition of a constant exponent, but based upon similar heat transfer principles is obtained by rearranging equations (VI-6 and -8), page 151. From equation (VI-6), neglecting variation in physical properties with temperature, an equivalent heat transfer coefficient for the entire unit may be expressed as some arbitrary function of the air flow rate W, i.e.,

$$h = f''(W) \tag{VI-22}$$

Thus, equation (VI-8) is equivalent to

$$q_a \propto (t_2 - t_1) f''(W) / \log_e \left[(t_{ef} - t_1)/(t_{ef} - t_2)\right] \qquad \text{(VI-23)}$$

The heat dissipation to the air is related to the temperature rise of the air by the heat balance equation

$$q_a = 456\, W\, (t_2 - t_1) \qquad \text{(VI-24)}$$

Hence, combination of above equations (VI-23 and -24) yields

$$\log_e \left[(t_{ef} - t_1)/(t_{ef} - t_2)\right] = f'(W) \qquad \text{(VI-25)}$$

which may be rearranged using equation (VI-24) to give in simplified form

$$(t_{ef} - t_1)/q_a = f(W) \qquad \text{(VI-26)}$$

The difference between the effective temperature t_{ef} and the inlet air temperature t_1 divided by the heat dissipation to the through-flow of air q_a is a function only of the rate of cooling air flow W. Equation (VI-26) indicates that a plot of the parameter $(t_{ef}-t_1)/q_a$ versus the air rate W, determined from bench-test data, provides a generalized basis for evaluation of the effective temperature for any environmental condition and flow rate of cooling air associated with operation of the unit.

2. Evaluation of Effective and Individual Temperatures of Components

In order that the prediction of operational thermal conditions from bench test data on the basis of an effective temperature be feasible, two requirements must be met. They are: (1) effective temperatures derived from individual component temperatures in bench tests must satisfactorily correlate the heat transfer parameter $(t_{ef}-t_1)/q_a$ with the cooling air rate W for all environmental conditions to which the equipment is subjected, and (2) the effective temperature must satisfactorily correlate individual component temperatures so that knowledge of effective temperature for any operational condition of the equipment will permit definition of component temperatures.

Component temperatures employed in the evaluation of the effective temperature may be all hot-spot or all average surface temperatures. It is recommended that the effective component temperature of a unit be determined as the arithmetic average of the measured temperatures of those components producing 75 per cent or more of the total heat dissipation. However, practical limitations may exist on the number of component temperatures possible to determine during bench test of an equipment and, in general, temperatures measured on a relatively small number of components should satisfactorily serve to define the effective temperature. Regardless of the magnitude of the heat dissipation and the number of components whose surface temperatures are to be measured, every attempt should be made to select representative components located throughout the unit. Among them should also be those components known to be thermally critical for the operational conditions to which the unit is to be exposed. The selection of components used to define the effective temperature may have to be revised repeatedly for some units in order to obtain data correlation. Factors to be considered must be location in the air flow passages, magnitude of heat dissipation, surface area of exposure to air flow, probable radiation and conduction to case surface, and possible effects of internal baffles or partitions separating heat producing from non-heat producing components.

Individual component surface temperatures should be correlated by plotting the difference between component temperature and inlet air temperature $(t_{cp}-t_1)$ versus the difference between effective temperature and inlet temperature, $(t_{ef}-t_1)$. The trend curves so obtained are applicable to the definition of component temperatures during operation of the equipment at other than bench test conditions.

3. Evaluation of Heat Dissipation to Through-Flow of Air

Recommended procedures for thermal evaluation necessitate a breakdown of the total heat dissipated by the unit into that absorbed by the through-flow of air and that transferred through the case of the unit. Because of possible uncertainties in the calculation of case heat transfer from knowledge of case surface and environmental temperatures, it is recommended that the heat dissipation to the through-flow of air be determined by the heat balance equation (VI-24). This procedure requires bench test data of air rate W, inlet air temperature t_1, and exit air temperature t_2. Since most equipments of this type have air flow induced by action of a blower, the inlet temperature t_1 is equal to that of the ambient air, and the exit air temperature t_2 may be measured readily at the discharge of the blower, correcting for any probable temperature rise during passage through the blower. Probing the discharge air and correcting the ambient temperature for rise due to air compression, whether by action of a blower or ram, would normally be required with forced flow. Procedures for probing air streams to determine mean values of temperature are discussed in Chapter III, page 21, and in Appendix II, page 343.

4. Case Heat Transfer

Heat transfer between components and the equipment case occurs principally by direct radiation and conduction through the chassis. Since the effective temperature t_{ef} represents to a considerable extent an average temperature level of the components, the difference between this temperature and the average case temperature defines an average temperature differential for case heat transfer. Therefore, it is recommended that the case heat transfer rate q_c be generalized by plotting its magnitude, derived from bench test data as the difference of q and q_a, as a function of the internal temperature differential $(t_{ef}-t_c)$. Case temperature t_c is determined during bench test of the equipment, and should be measured at a number of positions on the case surface to allow evaluation of an average temperature. During bench tests of most units, the case temperatures are likely to be similar to those encountered at other environmental conditions. Hence, insulation of the case to increase the case temperature would not normally be required.

However, it is advisable to make a preliminary evaluation of operational conditions after the first test data are available so as to obtain an estimate of the probable case temperature. Should it be high, it may have to be reproduced in bench tests by use of insulation.

An average value of the correction factor F_c applied to the evaluation of case heat transfer at other than bench test conditions may be obtained from bench test data. Heat transfer between the case surface and the equipment's environment should be evaluated for each bench test run by procedures discussed on pages 132 to 138 and the charts shown in Figures VI-2 and VI-3. The ratio of q_c, determined from heat balance equation (VI-21), to the calculated value of q_c from the working charts defines the correction factor for each test. The average of the correction factors for all bench tests should be used when predicting thermal performance of the unit under operational conditions.

5. Equipment Resistance to Air Flow

Knowledge of the resistance to through-flow of air is necessary to render bench test data useful for predicting thermal performance of the equipment. Resistance data obtained from bench test of the equipment should be correlated by the generalized equation

$$\sigma_1 \Delta p = C\, W^m, \qquad \text{(VI-27)}$$

where σ_1 is the ratio of the air density at entrance to the equipment to standard air density, Δp is the pressure drop of the air while passing through the equipment, generally expressed in inches of water, W is the air flow rate in pounds per second, and C and m are constants. Correlation of bench test data to the form of equation (VI-27) may be obtained by use of the resistance chart in Figure VI-9. The density ratio σ_1 is defined by use of Figure V-8. If it is determined that resistance data of any equipment are not accurately correlated by equation (VI-27), numerical values of $\sigma_1 \Delta p$ should be plotted as a function of the air rate W, and the resultant plot could then be used as a generalized chart for evaluation of pressure drop for any operational condition of the equipment.

6. Summary of Working Curves and Discussion of Evaluation Procedures for Prediction of Thermal Performance

The generalized working curves derived from bench test data necessary for thermal evaluation of open units having through-flow of cooling air are summarized in Figure VI-29. Plot A is the correlation of the temperature rise of individual components $(t_{cp}-t_1)$ with the effective temperature rise $(t_{ef}-t_1)$. Plot B shows the heat transfer parameter $(t_{ef}-t_1)/q_a$ as a function of the flow rate of cooling air W. Plot C gives the correlation of case heat transfer rate q_c in terms of the difference between effective temperature and case temperature $(t_{ef}-t_c)$. Plot D gives the corrected flow resistance $\sigma_1 \Delta p$ as function of the flow rate W. The plot may be used as a working curve or may be expressed by an equation.

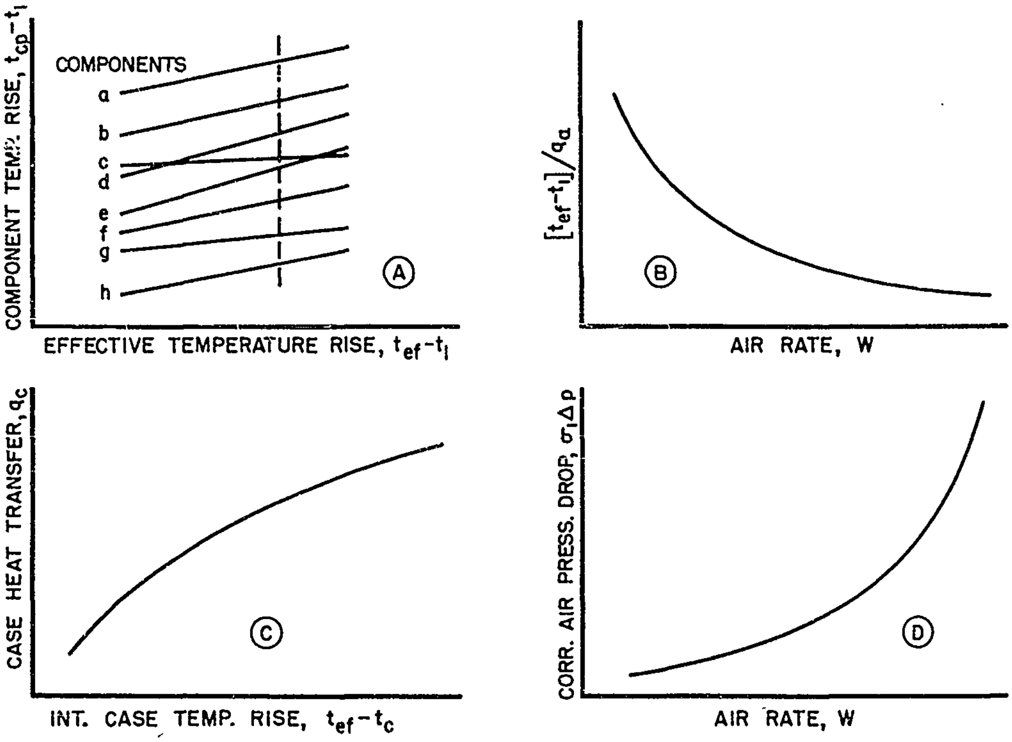


Figure VI-29. Summary of Working Curves for Open Unit with Forced or Induced Through-Flow of Atmospheric Air

Evaluation procedures for prediction of thermal performance require solution by trial-and-error calculation. In one type of thermal evaluation problem the cooling air is to be supplied to a unit by a blower of known performance characteristics and the unit is to be operated in a compartment of specified pressure, temperature and environmental conditions. It would then usually be desired to determine the operating temperatures of the components within the unit. A solution to this type problem may be obtained by determining first the flow rate of cooling air supplied by the blower. This requires use of the resistance relationship established for the equipment and of the procedures for defining the quantity of air delivered by the blower as discussed in Chapter V, page 105. Once the air rate W has been established the heat transfer parameter $(t_{ef}-t_1)/q_a$ is determined from the working curve illustrated in plot B of Figure VI-29. At this point in the evaluation process an assumption of some other variable must be made, which generally is the case heat transfer rate q_c. This permits evaluation of q_a, equal to $(q-q_c)$, and then the effective temperature t_{ef} from the parameter $(t_{ef}-t_1)/q_a$, since the inlet air temperature is known or may be determined. With the assumed case heat transfer rate q_c and the calculated effective temperature t_{ef}, the working curve of case heat transfer illustrated in plot C of Figure VI-29 is used to define an average case temperature t_c. Finally, the correctness of the assumed value for q_c is verified by evaluating the heat transfer between the equipment case and its environment, using the working charts in Figures VI-2 and VI-3 and the correction factor F_c. The over-all calculation process is repeated until agreement between the assumed and calculated case heat transfer rate is within 3 per cent. The effective temperature t_{ef} so determined defines individual component temperatures from the component trend curves illustrated in plot A of Figure VI-29.

In another type of problem it may be required to determine the cooling air rate and the corresponding blower performance necessary to maintain a maximum permissible temperature level within the unit. The evaluation procedure described above would also be followed, with the exception that the air rate becomes the primary dependent variable.

Typical problems illustrating the outlined evaluation procedures are contained in Examples VI-13 and -14, pages 219 to 221.

7. Units with Forced Flow Discharged Through Multiple Openings in the Case

Ordinary bench tests of units of this type are not practical since the quantity and temperature of the air leaving each opening would have to be determined experimentally to define the total heat dissipation to the through-flow of air. Principally for this reason, test data allowing thermal evaluation of the unit at other than test conditions are best obtained by insulating the case, except in the vicinity of the discharge openings, and testing the unit in a chamber or box, as described in Chapter IV, page 70. This test procedure allows evaluation of heat dissipation to the through-flow of air, which equals the heat generated by the unit since case heat transfer is essentially absent.

Thermal evaluation data determined from tests of the unit are used to define the generalized working plots A, B and D of Figure VI-29. It is not possible to construct plot C of Figure VI-29, necessary for evaluation of case heat transfer under operational conditions, since the case would be insulated in the bench tests. Hence, procedures for determining thermal performance of this type unit are the same as for other types of open units with through-flow of air, except for evaluation of case heat transfer, which may be handled in one of two possible ways. If case heat loss may be expected and only conservative estimates of component temperatures or air rates are desired, the case heat transfer may be neglected so that the heat dissipation rate q_a to the through-flow of air would be equal to q, the heat generation of the equipment. The second approach, which gives more accurate results with case heat loss and which must always be used when case heat gain is expected, is to assume the average case temperature equal to the average of the inlet and discharge air temperatures, and to calculate the heat transfer between the case and its environment by the working charts for free convection and radiation, Figures VI-2 and VI-3. Using the latter approach, case heat transfer by free convection should not be included for any portions of the case surface in the vicinity of the discharging air. A numerical value for the correction factor F_c must be assumed. When case heat loss is expected, F_c should be not greater than 0.9 to assure conservative results. Similarly, when case heat gain is expected, F_c should be taken as 1.1 or greater.

8. Examples

Example VI-12. Reduction of Bench Test Data for Open Unit with Induced Through-Flow of Air to Generalized Form for Use in Predicting Thermal Performance under Operational Conditions

a. Description of Unit, Measurement Procedures, and Test Data

A unit having induced through-flow of air is bench-tested to establish the heat transfer and resistance relationships necessary for thermal evaluation of the unit under operating conditions. The case has closed side panels, multiple openings on one end for intake of air and a single outlet at the opposite end to which the intake of the blower is connected.

The heat dissipation of the unit as tested is 500 watts with the blower motor unit removed under test. Air flow is induced through the unit by an auxiliary air supply apparatus consisting of a blower, metering section and transition duct. The latter is connected to the air discharge cowl of the equipment using a thick rubber gasket so as to reduce heat conduction from the equipment case to the duct and to insure that case heat loss occurs only from the case surface to the surroundings. Immediately following the discharge cowl is a 6-inch straight run of 2-inch diameter duct used as a probing section for determination of pressure and temperature of the air leaving the equipment. Within this section two diametrically opposed static pressure taps and a probing thermocouple are installed. Following this section are straightening vanes, orifice meter and an auxiliary blower-motor unit. The temperature of the air is measured between the straightening vanes and the orifice meter, since in order to meter the air flow, the orifice inlet temperature must be known.

The unit contains 70 components, of which 20 are chosen as representative of the entire group, as they dissipate well over 50 per cent of the total heat generated by the unit. Thermocouples are located on these components in accordance with the methods discussed in Chapter III.

Test conditions for the equipment are:

Ambient temperature	20°C
Ambient pressure	29.1 inches mercury
Environmental wall temperature	20°C
Surface of case	black-painted
Confinement of equipment	large room
Dimensions of equipment case	9 x 9 x 16 inches

The test data given in Table VI-6 are obtained.

Table VI-6. Measured Air Rates, Temperatures and Pressure Drops (Example VI-12)

Run No.	1	2	3	4	5	6
Air rate W, pound per second	0.013	0.02	0.03	0.04	0.05	0.06
Inlet air temperature t_1, °C	20	20	20	20	20	20
Exit air temperature t_2, °C	89	67	52	45	40	37
*Effective temperature t_{ef}, °C	176	136	115	98	93	85
**Case temperature t_c, °C	43	40	38	35	32	29
Pressure drop Δp, inches water	0.155	0.32	0.62	1.05	1.50	1.96

*Average of 20 component temperatures. Individual component temperatures not reported here but would be correlated in terms of the effective temperature t_{ef}.
**Average of 16 thermocouple readings.

b. Reduction of Test Data

Reduction of test data to the generalized forms is illustrated for Run No. 3. The heat dissipated to the through-flow of air is evaluated from equation (VI-24), which gives

$$q_a = 456\, W\, (t_2 - t_1) = 456 \times 0.03(52 - 20) = 438 \text{ watts}$$

The effective component surface temperature for this test is 115°C, giving for the heat transfer parameter

$$(t_{ef} - t_1)/q_a = (115 - 20)/438 = 0.217\ ^\circ\text{C per watt}$$

The case heat loss is found by heat balance as

$$q_c = q - q_a = 500 - 438 = 62 \text{ watts},$$

and the internal temperature differential for this heat loss is

$$t_{ef} - t_c = 115 - 38 = 77^\circ\text{C}$$

The correction factor F_c to be applied to evaluation of case heat transfer by free convection and radiation is next determined. During test of the unit, the proximity of the lower panel of the case to the mounting bench essentially eliminated heat transfer from this surface. Also, free convective heat transfer from the end of the case admitting the through-flow of air must be excluded from case heat transfer calculations. The evaluation of calculated case heat transfer follows the procedure illustrated in Example VI-1, which for Run No. 3 yields a value of 69.6 watts in comparison with the experimentally determined value of 62 watts. Hence, the correction factor is $F_c = 62/69.6 = 0.89$. The corrected pressure drops $\sigma_1 \Delta p$ of the through-flowing air are obtained by multiplying the measured pressure drops with the inlet air density ratio, determined for 20°C and 29.1 inches mercury in Figure V-8 as $\sigma_1 = 0.97$. The reduced data for all the tests are obtained in similar manner and are summarized in Table VI-7.

Table VI-7. Reduced Test Data (Example VI-12)

Run No.	1	2	3	4	5	6
W, pound per second	0.013	0.03	0.03	0.04	0.05	0.06
$(t_{ef} - t_1)$, °C	156	116	95	78	73	65
q_a, watts	401	429	438	456	456	465
$(t_{ef} - t_1)/q_a$, °C per watt	0.389	0.270	0.217	0.171	0.160	0.140
$(t_{ef} - t_c)$, °C	133	96	77	63	61	56
q_c, watts	99	71	62	44	44	35
F_c	1.09	0.90	0.89	0.77	1.01	1.06
$\sigma_1 \Delta p$, inches water	0.15	0.31	0.60	1.02	1.45	1.90

The correction factor F_c is seen to vary among the various runs. For use in evaluation of operational conditions, the average value of 0.95 would be used.

The recommended working charts obtained from these data for use in evaluation of thermal performance at other than bench test conditions are shown in Figures VI-30, -31 and -32. An additional plot showing individual component temperature rise $(t_{cp}-t_1)$ versus the effective temperature rise $(t_{ef}-t_1)$ is required, but is not included in this example.

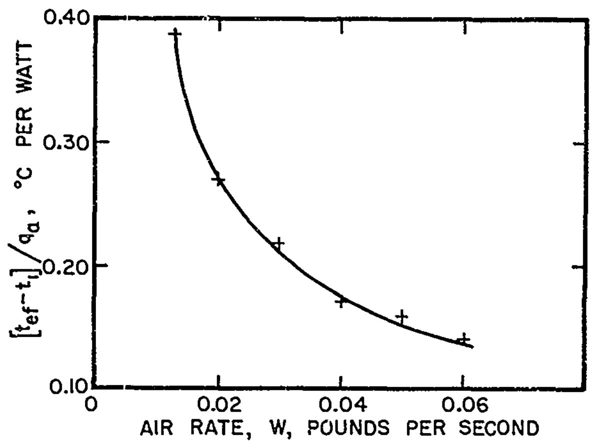
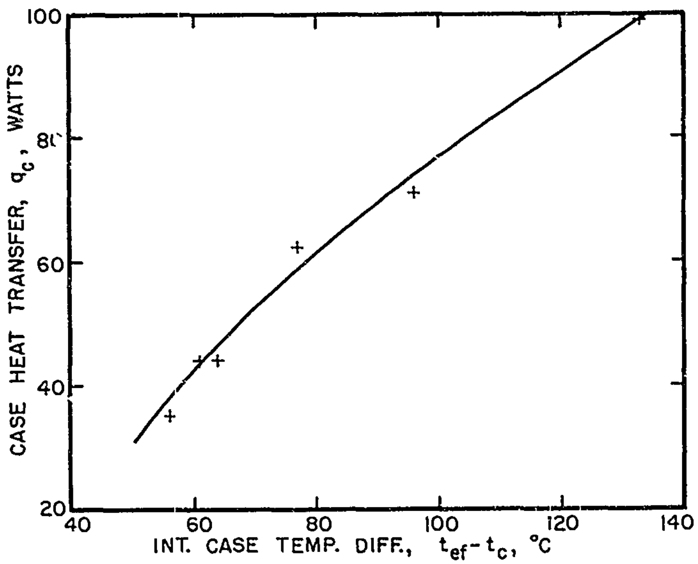


Figure VI-30. Heat Transfer Parameter of Unit (Example VI-12)



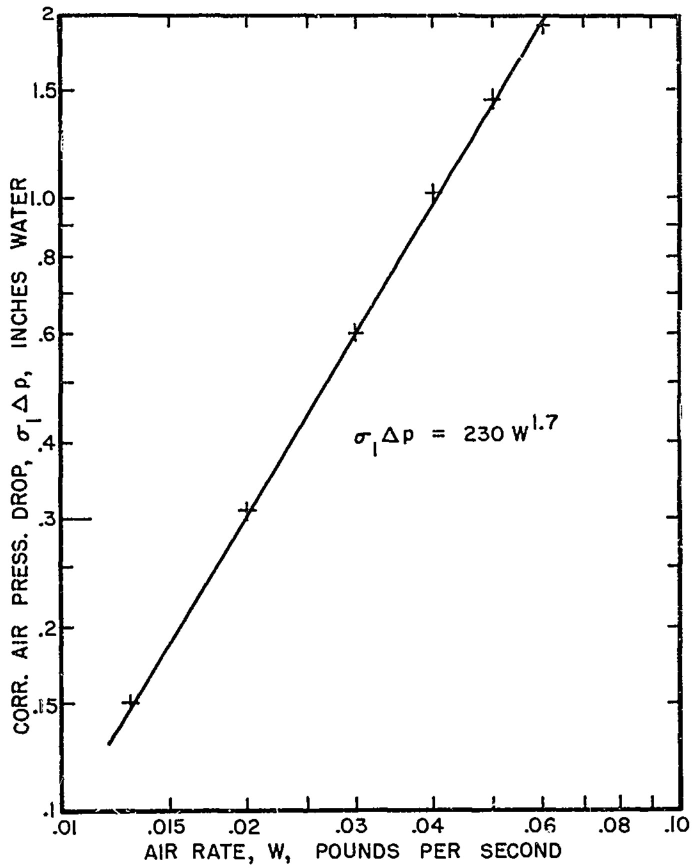


Figure VI-32 (above). Air Flow Resistance Characteristics of Unit (Example VI-12)

Figure VI-31 (left). Case Heat Loss Characteristics of Unit (Example VI-12)

Example VI-13. Determination of an Individual Component Temperature of an Open Unit with Induced Through-Flow of Atmospheric Air at Altitude Operating Conditions

The following operating condition is specified for the equipment described in the previous Example VI-12:

Ambient pressure	5.6 inches mercury (40,000 feet altitude)
Ambient temperature	$10^{\circ}C$
Compartment wall temperature	$10^{\circ}C$
Radiation-receiving surface	Dural
Confinement of equipment	Intermediate
Heat generation of equipment	500 watts

A blower of known performance provides the through-flow of air. It is desired to determine the temperature of a component being part of this equipment on basis of the generalized bench test data presented in Figures VI-30, -31, and -32 in Example VI-12. The trend curve for the temperature rise of this component is given in Figure VI-33.

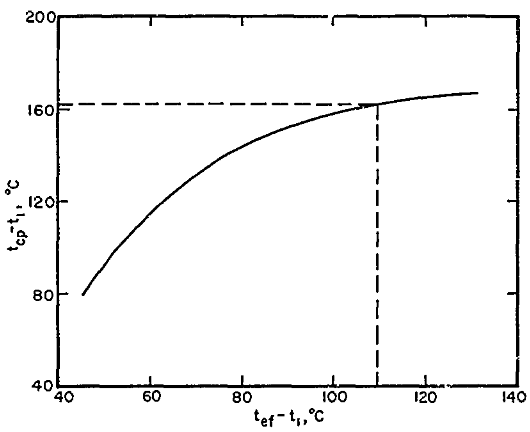


Figure VI-33. Temperature Rise Trend Curve for Component (Example VI-13)

Using the performance data of the blower, the equipment resistance relationship shown in Figure VI-32, and the calculational procedures discussed in Chapter V, it is determined that the blower induces air flow through the equipment at the rate of 0.027 pound per second.

The calculation process requires trial-and-error. If the case heat transfer rate q_c is assumed equal to be 64 watts, then by heat balance,

$$q_a = q - q_c = 500 - 64 = 436 \text{ watts.}$$

From Figure VI-30, for an air rate of 0.027 pound per second the heat transfer parameter $(t_{ef}-t_1)/q_a = 0.25$ °C per watt. Hence,

$$t_{ef} = t_1 + 0.25 \times 436 = 10 + 109 = 119^{\circ}C.$$

The internal temperature differential for case heat transfer is found from Figure VI-31 for $q_c = 64$ watts as $(t_{ef}-t_c) = 83.5^{\circ}C$. Hence,

$$t_c = t_{ef} - 83.5 = 119 - 83.5 = 35.5^{\circ}C.$$

Should this be the correct value of the case temperature, the heat loss to the environment from the case surface will agree with the assumed value of 64 watts. Using the working charts in Figures VI-2b and -3b, the average correction factor $F_c = 0.95$, and $\phi_1 = 0.63$ from Table VI-1, the free convective and radiant heat transfer from the case surface is found to be 65.2 watts for the calculated case temperature of $35.5^{\circ}C$. Since this value agrees essentially with the assumed value of 64 watts, the thermal condition of the equipment may be considered defined. The final step is to define the component temperature from the calculated effective temperature of $119^{\circ}C$ and the correlation plot of Figure VI-33. For an effective temperature rise of $(t_{ef}-t_1) = 109^{\circ}C$, the component temperature rise $(t_{cp}-t_1)$ is found to be $162^{\circ}C$, giving a component temperature of $172^{\circ}C$.

Example VI-14. Determination of Required Blower Performance for Satisfactory Thermal Performance of an Open Unit with Induced Through-Flow of Atmospheric Air

The calculated data of Example VI-13 may be used to illustrate this type problem, supposing it is decided that satisfactory thermal operation of the equipment is obtained with the effective component surface temperature of $119^{\circ}C$, and the operational conditions are the same as specified in Example VI-13, i.e., 40,000 feet pressure altitude, $10^{\circ}C$ compartment temperature, etc.

The calculational procedure of Example VI-13 need not be repeated to determine the case heat loss q_c which must be known. From Example VI-13, $q_c = 64$ watts. Then, since the values of $t_{ef} = 119^{\circ}C$, $t_1 = 10^{\circ}C$ and $q_a = q - q_c = 436$ watts are known, the required air flow rate would be determined directly from Figure VI-30. Since the conditions are the same as in Example VI-13, $W = 0.027$ pound per second. The corresponding corrected pressure drop is, from Figure VI-32, $\sigma_1 \Delta p = 0.5$ inch water.

The density ratio σ_1 is determined from Figure V-8 for 5.6 inches mercury and $10^{\circ}C$ as $\sigma_1 = 0.192$. Hence, the actual pressure drop of the air flowing through the unit is $0.5/0.192 = 2.6$ inches water. The pressure of the air at the exit of the unit, and entrance to the blower, is $5.6 - 2.6/13.55 = 5.41$ inches mercury. The temperature of the air at entrance to the blower is determined by heat balance. Since q_a was found equal to 436 watts (from Example VI-13), then

$$t_2 = t_1 + q_a/(456\ W) = 10 + 436/(456 \times 0.027) = 45.4^{\circ}C$$

The density ratio at entrance to the blower, obtained from Figure V-8 for

5.41 inches mercury and 45.4°C, is $\sigma_i = 0.165$. The blower must produce a static pressure rise of 2.6 inches water, which is equivalent to producing 2.6/0.165 = 15.8 inches water under standard conditions. The volumetric capacity of the blower based on intake conditions is defined by equation (V-1) as

$$Q = 784\ W/\sigma_1 = 784 \times 0.027/0.165 = 128 \text{ cubic feet per minute.}$$

The required blower power input, assuming a blower efficiency of 60 per cent, is

$$hp = Q\ \Delta p/(6350\ \eta_s) = 128 \times 2.6/(6350 \times 0.6) = 0.0874$$

This is equivalent to a 65-watt output required of the drive motor.

CHAPTER VII

USE OF ALTITUDE CHAMBER TEST DATA FOR DETERMINATION OF OPERATIONAL THERMAL CONDITIONS IN THE STEADY STATE

Altitude chamber tests of electronic equipment should be conducted whenever it is not feasible to predict the operational thermal conditions from data secured during bench test. The general classifications of units requiring test in an altitude chamber are those which are vented, those which have blower-motor units of unknown performance, and those whose thermal performance should be verified by actual test for a range of ambient air pressure and temperature. As pointed out in Chapter IV, altitude chamber tests of pressurized and sealed units cooled by free convection and radiation should not be made since the results are likely to be misleading. Types and procedures of altitude chamber tests and measurements to be made in such tests are discussed in Chapter IV for the various classifications of electronic equipments.

Although environmental temperature and pressure are controlled in altitude chambers, they usually do not adequately simulate installation environments. Ambient air pressure and temperature, the state of motion of the ambient air and the type and temperature of surrounding surfaces are basic factors defining the character of a unit's environment. Change in any one of these factors could significantly alter the thermal state of the unit, which defines the operating temperature levels of the thermally critical components. Units tested in an altitude chamber are subjected to environments characterized, in most instances, by temperatures of the radiation-receiving surfaces substantially equal to the chamber air temperature, and by appreciable movement of air over the external surfaces of the unit. On the other hand, units installed in aircraft compartments may be exposed to environments having temperatures of the radiation-receiving surfaces 20° to 40°C above the ambient air temperature, and essentially still air throughout the compartment. Consequently, the thermal state of the unit as described by altitude chamber test would not give a representative indication of the thermal state of the unit as installed in an aircraft compartment.

The purpose of this chapter is to present practical working procedures for evaluating, from data secured during steady-state test in an altitude chamber, the thermal condition of electronic equipment when operated in an installation environment. The working procedures are intended to be corrective and are not applicable to situations where deviations of ambient pressure and temperature from those maintained during test in the altitude chamber are large.

Pressurized and sealed units designed to be cooled externally by free convection and radiation are greatly affected by their environment. Since the case temperature is the principal significant variable, operation of such units under installation conditions can be predicted readily from bench test data, making altitude chamber tests, and therefore also corrective procedures for them, entirely unnecessary. For such units the analytical methods of Chapter VI are sufficient to predict operating thermal conditions from bench test data.

Pressurized and sealed units having a circumferential baffle or a case-envelope heat exchanger are affected by their environment because heat transfer to or from the external baffle surfaces may be caused. Units of this type having low heat concentration are more susceptible to environmental changes, since a fairly large percentage of the heat generated could normally be transferred to the environment, rather than to the cooling air. When the heat concentration of the unit is high, the required cooling air rate would be large in proportion to the external surface and the effect of any change in the environment on the thermal condition of the unit would be less important.

The thermal condition of pressurized units employing an integrated or separate heat exchanger would be affected by change in the environment. The internal air flow rate is normally quite high and produces relatively high case temperatures. The latter are conducive to appreciable heat transfer between the case and its environment.

Closed vented units designed to be cooled externally by free convection and radiation are greatly affected by any change in the environment. All the heat generated by the unit must be dissipated to the environment by the surface of the case, so that environmental conditions differing from those maintained during test in the altitude chamber would have a direct bearing on the case temperature and with it on component temperatures of the unit.

Thermal conditions of open units relying on heat dissipation by external radiation and natural through-flow of atmospheric air are directly affected by the environmental conditions. However, their heat transfer processes are not amenable to corrective analytical procedures. Environmental conditions for this type of unit must be simulated as well as possible during test in the altitude chamber.

Thermal conditions of open units relying on heat dissipation principally to forced through-flow of atmospheric air are somewhat affected by change in environmental conditions. Corrections of altitude chamber test data for this type of unit, to take into account deviations in environmental conditions, are only approximate.

In the following, corrective working procedures are presented for those types of units which would be affected by deviations in environmental conditions from altitude chamber test conditions, as discussed above, and whose heat transfer processes lend themselves most readily to analysis.

<u>Pressurized and Sealed Units with Case Cooled by Forced Convection</u>

Units of this type have a plain case surrounded with a circumferential baffle to provide an air flow passage, or have a case-envelope heat exchanger.

Air which is induced or forced by a blower through the flow passages formed by the baffle or the heat exchanger receives, by the process of forced convective heat transfer, heat generated by the unit. Part of the heat generated by the unit may also be dissipated by the external surface of the baffle or heat exchanger to the unit's environment. Altitude chamber tests recommended for this type of unit are discussed in Chapter IV, page 65.

1. Use of Test Data for Determination of Air Rate and Component Temperatures

The flow rate of cooling air supplied to the heat transfer passages is determined from altitude chamber test data by heat balance. The test procedures outlined in Chapter IV require that the external surface of the unit be insulated. Thus, all heat generated by the unit is dissipated in the heat transfer passages to the cooling air. The heat balance equation for the process is

$$q = 456\, W\,(t_2 - t_1), \qquad \text{(VII-1)}$$

where q represents the total rate of heat generation of the unit expressed in watts, W the air rate in pounds per second, and t_2 and t_1 the mean temperatures of the air at exit and entrance to the unit, respectively.

When the mean air temperatures, t_1 and t_2, are determined during test of the unit in an altitude chamber and the rate of heat generation is known, the air rate provided by the blower may be calculated directly from equation (VII-1). When the blower induces air flow through the heat transfer passages of the unit, the inlet temperature of the air t_1 would be the temperature of the air within the altitude chamber. With forced flow of air through the passages, the mean air temperature t_1 is above the chamber air temperature because of the energy imparted to the air by the blower. The temperature rise of the air across the blower is ordinarily negligible, but may be significant for units designed to operate at high altitude. Two evaluation procedures might be followed for determination of the air rate when forced flow is used, with the choice of the procedure depending upon the temperatures recorded during test. In one procedure the air rate may be evaluated directly from equation (VII-1) when the mean air temperature at discharge from the blower is measured. In the other procedure, if the air temperature at discharge from the blower has not been measured, but the electrical input to the drive motor of the blower is known, it would be reasonable to assume as a good approximation that the total rate of heat dissipated to the air is the sum of that generated by the unit and the electrical input to the drive motor. The inlet temperature t_1 to be used in equation (VII-1) for this method of evaluation is the temperature of the air within the altitude chamber. If the cooling air for the unit is not utilized to cool the blower's drive motor, a more accurate evaluation of the air flow rate would be obtained by estimating the power output of the motor and using the heat dissipation in equation (VII-1) as the sum of this output, which must be converted to heat, and the heat generated by the unit.

The test of a unit in an altitude chamber may be conducted for evaluation of the air rate at only one combination of chamber pressure and temperature, or for a variety of pressures and temperatures, depending upon the range of environmental conditions to be studied. The typical variation of air flow with chamber pressure and temperature for a unit having a blower without speed regulation is illustrated in Figure VII-1. The weight rate of air flow handled by an uncontrolled blower will decrease appreciably with reduced chamber pressure. With normal blower and drive motor performance characteristics, the air rate will vary with change in chamber temperature at a rate somewhat less than the proportional change in density of the air within the chamber. However, it is recommended that for temperature changes not exceeding 20° to 30°C from the value maintained during test in the altitude chamber, the air rate be corrected in direct proportion to the air density, or its equivalent, the inverse ratio of the absolute temperatures. This method of correction produces conservative values of predicted air rate when the temperature increases, since the actual reduction in air flow would be slightly less than predicted. The predicted values of air rate would be somewhat high when the air temperature drops. Any inaccuracies introduced by this method of correction would be of negligible importance, except for drive motors having appreciable variation in speed with change in torque.

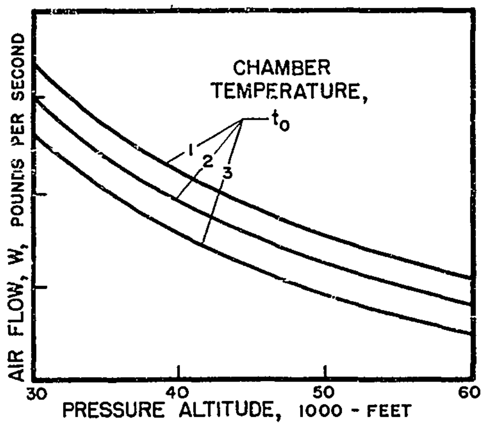


Figure VII-1. Typical Air Rate Variation with Altitude Pressure and Temperature for an Equipment Employing an Uncontrolled Blower as the Source of Cooling Air

Correction of air rate to ambient pressures different from the pressure existing during test in the altitude chamber would be conducted in the same manner as recommended for change in air temperature, i.e., in direct proportion to the air density. The corrective procedure should be limited to pressure changes not exceeding 10 to 15 per cent of the absolute air pressure during test. Consequently, it can only be employed for small changes in altitude; roughly, several thousand feet change at low altitudes and about one thousand feet for high altitudes.

The complete thermal evaluation of a unit tested in an altitude chamber to determine only the air flow rate would require use of additional data obtained from bench test and the thermal evaluation procedures described in Chapter VI. Use of altitude chamber and bench test data would permit evaluation of the case and internal component temperatures of the unit for all combinations of ambient air pressure and temperature to which the unit was subjected during the altitude chamber tests. Calculated case and component temperatures would be plotted as a function of the equivalent pressure altitude and temperature of the ambient air, as illustrated in Figure VII-2. Those combinations of ambient air pressure and temperature resulting in overheating of the unit may be defined readily.

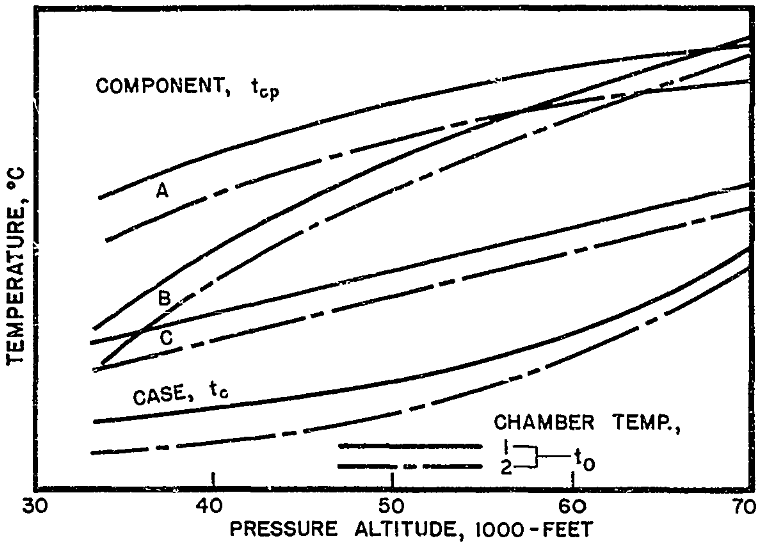


Figure VII-2. Illustration of Effects of Ambient Air Pressure and Temperature on Case and Component Temperatures

Air flow data obtained from altitude chamber tests, as illustrated in Figure VII-1, would be extrapolated to lower air pressure, i.e., higher altitude, and higher air temperature when it is desired to predict the limiting conditions of operation and it is found that the range of pressure and temperature covered during test in the altitude chamber has not resulted in thermally critical operation of the unit. Reasonably accurate prediction of case and component temperatures somewhat beyond the test range would be expected, permitting extension of the temperature curves illustrated in Figure VII-2.

2. Use of Test Data for Determination of Component Temperatures from Measured Case Temperature

Certain units of this type would be tested in an altitude chamber to measure the variation in case temperature with altitude chamber air pressure and temperature. The test procedures recommended in Chapter IV, page 65, require test runs with and without insulation on the external surface of the baffle or heat exchanger. The measured value of the case temperature with insulation over the external surface of the unit defines the maximum temperature of the case for any particular combination of environmental air pressure and temperature in an actual installation, except when heat gain to the unit from its environment occurs. The case temperature measured in an altitude chamber test in which the external surface of the unit is bare would be about as low as could be expected under any installation conditions.

Measured values of the case temperature would be plotted as a function of the chamber air pressure and temperature, as illustrated in Figure VII-3, for the external surface of the unit insulated and bare. The case temperatures used in conjunction with component temperatures measured during bench test of the unit would be used to define precise operational limits of the unit. However, knowledge of the case temperature alone is usually a sufficiently good index of the thermal state of the unit.

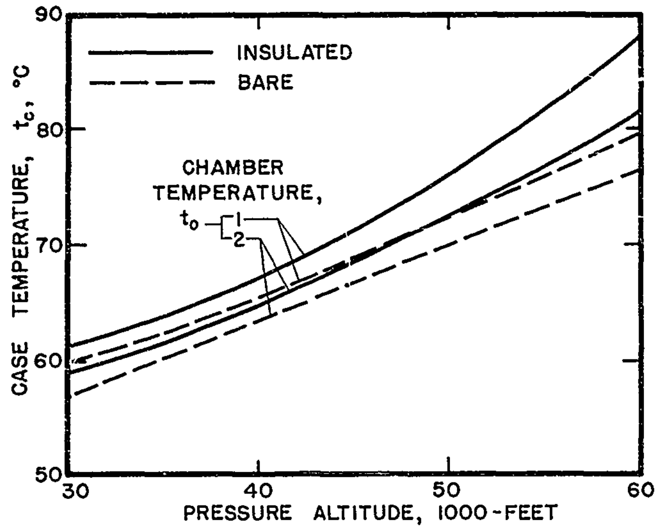


Figure VII-3. Typical Variation of Case Temperature with Ambient Air Pressure and Temperature as Determined from Altitude Chamber Test

3. Use of Test Data for Determination of Heat Transfer Characteristics of Unit

Data obtained from an altitude chamber test of a pressurized or sealed unit may be used as a basis for establishing the heat transfer characteristics of the cooling air passages and the effect of heat transfer between the external surface of the unit and its environment. The heat transfer characteristics so established would be useful for predicting the case surface temperature of the unit under actual installation conditions in an aircraft compartment, where the environmental conditions may differ appreciably from those during test in the altitude chamber. In other words, once the unit has been tested in an altitude chamber at a specified combination, or combinations, of air pressure and temperature, the steady-state thermal condition of the unit may be evaluated when subjected to any type environment having the same combination, or combinations, of ambient air pressure and temperature.

Measurements and test runs in the altitude chamber recommended in Chapter IV, page 65, define case temperature, mean air temperature at inlet and exit to the heat transfer passages and the temperature of the external surface of the unit for each combination of air pressure and temperature within the altitude chamber.

a. Units with Circumferential Baffle

The heat generated by the unit is that dissipated by the case as q_c to the cooling air flowing through the annular passage formed by the case and the baffle, and indirectly to the environment of the unit through the external surface of the baffle. Thus, a heat balance applied to the unit yields

$$q_c = q_a + q_e \tag{VII-2}$$

where q_a and q_e represent the rate of heat dissipation to the cooling air and from the external surface of the baffle, respectively. Heat gain of the unit from the environment would be negative in sign with relation to equation (VII-2), so that the heat to be dissipated to the cooling air is the sum of that generated by the unit and the external heat gain.

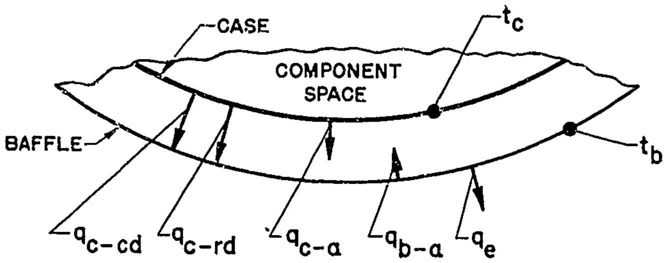


Figure VII-4. Contributing Modes of Heat Transfer for a Unit with Circumferential Baffle and Forced Air Cooling

Within the annular flow passage formed by the case and baffle, the cooling action may be reduced to the contributing modes of heat transfer. This is illustrated in Figure VII-4. The heat generated by the unit q_c is dissipated from the surface of the case by radiant and conductive heat transfer to the baffle, q_{c-rd} and q_{c-cd}, and by convective heat transfer to the cooling air q_{c-a}. A heat balance applied at the surface of the equipment is

$$q_c = q_{c-rd} + q_{c-cd} + q_{c-a} \tag{VII-3}$$

The radiant and conductive heat transfer from the case to the baffle, q_{c-rd} and q_{c-cd}, are dissipated to the cooling air by forced convective heat transfer on the inside surface of the baffle, represented by q_{b-a}, and by transfer through the baffle to the unit's environment, represented by q_e. A heat balance applied at the inside surface of the baffle is

$$q_{c-rd} + q_{c-cd} = q_{b-a} + q_e \tag{VII-4}$$

Also, since the heat dissipated to the cooling air is the result of forced convective heat transfer over the inner surface of the baffle and the outer surface of the case,

$$q_a = q_{c-a} + q_{b-a} = 456\ W\,(t_2 - t_1), \tag{VII-5}$$

where W represents the air rate in pounds per second, and t_2 and t_1 is the exit and entrance temperature, respectively, of the cooling air flowing through the passage formed by the baffle and case.

For each combination of air pressure and temperature within the altitude chamber, a test run on the unit with the baffle insulated is required. Hence, the heat dissipation to the cooling air q_a is equal to the heat generated by the unit q_c, and the air rate W may be evaluated from equation (VII-1). Once the air rate has been evaluated, the heat dissipation to the cooling air q_a when the baffle surface is bare would be evaluated from equation (VII-1) using the measured values of the inlet and exit cooling air temperatures, t_1 and t_2. The heat dissipation to the environment from the baffle q_e may then be evaluated from equation (VII-2). When the blower induces the air flow through the equipment, the air rate for the test run with bare external surfaces is not exactly equal to that when the surfaces are insulated, since the air temperatures leaving the unit and entering the blower, and correspondingly the air densities, are not equal in both tests. Within the usual range of temperature change, the blower will handle a constant volume rate of flow and the weight rate will vary in direct proportion to the air density at the blower's inlet. A correction to the air rate determined from the insulated test should be made if an exact procedure is desired. Normally, however, the change in air rate is less than two per cent and would be ignored in the evaluation procedure.

The radiant heat transfer from the case to the baffle q_{c-rd} would be calculated from equation (VI-4), using the measured values of case and baffle temperature and the external area of the case opposite the baffle as the radiating surface. The term q_{c-cd} representing the heat conducted from the case to the baffle through struts or other supporting members for the baffle, may frequently be neglected since forced convective heat transfer dissipates an appreciable portion of this heat to the cooling air before reaching the baffle. A rough estimate of the conductive heat transfer rate through the struts to the baffle in watts is obtained from

$$q_{c-cd} = \left[kb\ (t_c - t_b)\right] / \left[10.9 - 3.26 \log_e A + 41\ bx/A\right] \qquad \text{(VII-6)}$$

where A is the cross-sectional area of all struts in square inches, x the mean radial length of the struts from case to baffle in inches, b the thickness of the baffle in inches, and k the thermal conductivity of the material of the struts and baffle in Btu per hour-foot-°F (118 for aluminum). Equation (VII-6) should be employed to include q_{c-cd} in the heat balance when the term is more than 20 per cent of q_{c-rd}.

The heat transfer by forced convection from the inner surface of the baffle to the cooling air q_{b-a} may be evaluated from equation (VII-4), and that from the case surface to the cooling air q_{c-a} from equation (VII-3 or -5). The average temperature of the cooling air t_a, which is defined as

$$t_a = (t_1 + t_2)/2 \qquad \text{(VII-7)}$$

would be used to define values of the effective heat transfer coefficients for the case and baffle surfaces. These are

$$H_{c-a} = q_{c-a}/(t_c - t_a) \qquad \text{(VII-8)}$$

and

$$H_{b-a} = q_{b-a}/(t_b - t_a) \qquad \text{(VII-9)}$$

and may be assumed to remain constant for any one combination of air pressure and temperature within the altitude chamber.

Once the effective heat transfer coefficients H_{c-a} and H_{b-a} evaluated from altitude chamber test data are known, it is possible to establish a working method permitting evaluation of the surface temperature of the case for any environmental conditions to which the unit may be subjected, providing that the ambient air pressure and temperature in the actual environment are the same as during test in the altitude chamber. The working method is obtained by combining equations (VII-5, -8, and -9) to give an equation for the case temperature

$$t_c = q_a/H_{c-a} - (H_{b-a}/H_{c-a})(t_b - t_a) + t_a \qquad \text{(VII-10)}$$

In addition, by combination of equations (VII-4 and -9), the sum of radiant and conductive heat transfer from the case to the baffle is given by

$$q_{c-rd} + q_{c-cd} = q_e + H_{b-a}(t_b - t_a) \qquad \text{(VII-11)}$$

Lastly, the temperature of the surface of the case t_c would again be calculated by use of equation (VI-4), the charts in Figure VI-3, the value of the radiant and conductive heat transfer calculated from equation (VII-11 and -6), the baffle temperature t_b and the case surface area.

A trial-and-error solution to the set of equations (VII-6, -10, and -11) and (VI-4) is required to define the case temperature. It is recommended that the trial-and-error process of calculation be conducted by first assuming a value for the temperature of the baffle t_b. With this value of t_b, the known environmental conditions and the surface area of the case, a value of the external heat transfer q_e would be calculated. When the external heat transfer to or from the environment occurs by radiation and free convection, as may be the situation in many installations, the procedures outlined in Chapter VI would be followed. Next, the heat dissipation to the cooling air q_a is defined by equation (VII-2), the exit cooling air temperature t_2 from equation (VII-5), and the average cooling air temperature t_a from equation (VII-7). A value of the case temperature t_c may now be calculated from equation (VII-10). The radiant heat transfer q_{c-rd} is evaluated by equation (VII-11), after deducting the conductive heat transfer q_{c-cd} as calculated from equation (VII-6). This then permits evaluation of a second value of the case temperature t_c by equation (VI-4) and the charts in Figure VI-3. A solution to the trial-and-error procedure is obtained when the two values of case temperature are in agreement. If the two values do not agree, it is necessary to assume another value for the external heat transfer q_e and repeat the calculation process. The procedure of calculation is illustrated in Example VII-1, page 235.

The preceding procedure is specifically applicable to any evaluation in which the air pressure and temperature is the same as in the altitude chamber test. More generally applicable data result when a unit is tested in an altitude chamber for a range of air pressure and temperature, and the case, baffle and cooling air temperatures are measured. On the basis of these test data, it is possible to construct the general heat transfer characteristics of the unit. The reduction of test data for each combination of air pressure and temperature would be accomplished by following the previously outlined procedure for determination of H_{c-a} and H_{b-a} in an exactly similar manner. The values of H_{c-a} and H_{b-a} are primarily functions of the air pressure and secondarily of the chamber air temperature, the latter effect being of negligible importance with most units. Values of H_{c-a} and H_{b-a} for all combinations of air pressure and temperature would be plotted as function of the air rate W, giving curves illustrated in plots A and B of Figure VII-5. The air rate variation with air pressure and temperature would be included as a third working chart, as shown in plot C of Figure VII-5. The procedures for use of the generalized data of Figure VII-5 would be the same as discussed above for the unit tested at a single combination of air pressure and temperature.

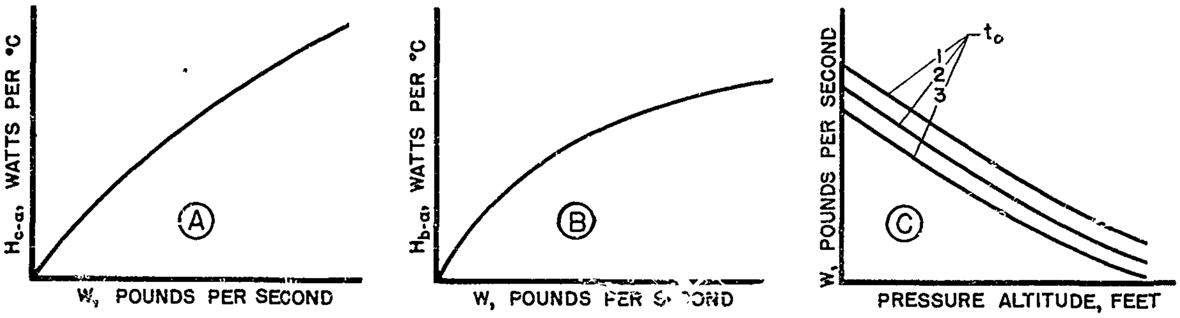


Figure VII-5. Generalized Heat Transfer Characteristics of Unit Determined from Test at Various Air Pressures and Temperatures

Generalized data as illustrated in Figure VII-5 would be particularily useful in estimating the thermal conditions of a unit operating in an environment having air pressure and temperature different from those covered during test in the altitude chamber. The procedure for evaluation under such conditions would be by interpolation or extrapolation of the data used to construct the plots illustrated in Figure VII-5. The curves in plots A and B have ordinate intercepts of zero, so that extrapolation of the data to reduced values of air rate W should be of fairly good accuracy. Extrapolation of the curves for plots A and B to higher values of the air rate W would be of questionable accuracy, but this range is less important since it represents a region of higher air flow where thermally critical operation of the unit would not be likely to occur, unless the air temperature is greatly increased. The air rate as a function of air pressure and temperature, illustrated in plot C, would also require interpolation or extrapolation. A useful guide to the extrapolation of the air rate curve would be obtained by plotting the ratio of the air rate to air density as a function of the air density. The trend curve so established could more accurately be extrapolated than the air rate curve of plot C, Figure VII-5.

If the unit has been tested in the altitude chamber at only one combination of air pressure and temperature, the characteristic curves shown in Figure VII-5 cannot be constructed. However, corrections for small variations in air pressure and temperature from those occurring during test in the altitude chamber could be made by evaluating the change in air rate according to the procedures recommended on page 225, and then correcting the effective heat transfer coefficients H_{c-a} and H_{b-a} in direct proportion to the air rate to the 0.8 power. This procedure would be limited to temperature changes of 20° to 30°C from the test air temperature and pressure changes of 10 to 15 per cent of the absolute test pressure.

b. Units with Case-Envelope Heat Exchanger

The heat generated by the unit is dissipated to the cooling air flowing through the heat exchanger surrounding the case or in part to the environment of the unit through the external surface of the heat exchanger. A heat balance applied to this unit is expressed by equation (VII-2), which is the same as for the unit having a circumferential baffle.

The case-envelope heat exchanger differs from the circumferential baffle in that greater heat transfer surface is provided by the addition of fins, tubes or corrugated sheet metal. Forced convective heat transfer to the cooling air is increased because of the extended surface, but radiant heat transfer from the case to the external shell of the exchanger is practically eliminated since the intermediate surfaces act as radiation shields. Conductive heat transfer from the surface of the case through the internal surfaces of the exchanger plays an important role in transferring heat to the cooling air and to the external surface of the exchanger. The greater portion of the heat transferred from the case by conduction is absorbed by the cooling air, with the remaining portion being transferred to the external surface of the exchanger. At the external surface it is transferred back to the cooling air or partially to the unit's environment. Any heat gain by the unit from its environment is conducted to the internal surfaces of the heat exchanger and absorbed by the cooling air.

It is not possible to calculate the conductive heat transfer directly, like radiant and conductive heat transfer with the unit having a circumferential baffle, because of the complex interaction with the forced convective heat transfer. Hence, corrective procedures applied to altitude chamber tests for this type of unit are somewhat more approximate and, in particular, less reliable when the environment differs from test conditions to the extent that extrapolation of the data would be required. The corrective procedures described in the following are, however, recommended as satisfactory for units subjected to change in environment resulting from variations in the radiant and convective heat transfer external of the unit and for small variations of the ambient temperature and pressure from values occurring during test.

The procedures for test of this unit in the altitude chamber at any specified combination of air pressure and temperature require one run with the external surface insulated, as discussed in Chapter IV, page 66. The mean air temperature at entrance and exit of the heat exchanger, t_1 and t_2, the case surface temperature t_c and the external surface temperature of the heat exchanger t_e are to be measured. For this test run the external heat loss is zero and the air rate W may be evaluated from equation (VII-1). Thus, the temperature difference $(t_c - t_e)$ measured during this test corresponds to zero external heat transfer.

A second test run in the altitude chamber is required with the external surfaces of the unit bare. Here again, the mean air temperatures, t_1 and t_2, the case temperature t_c and the external surface temperature t_e are to be measured. The air rate W is known from the insulated test, so that the heat dissipation to the cooling air q_a may be evaluated by equation (VII-1). The external heat loss q_e is defined by heat balance according to equation (VII-2). The measured values of t_c and t_e define a temperature differential across the heat exchanger $(t_c - t_e)$ which corresponds to the defined value of the external heat transfer q_e.

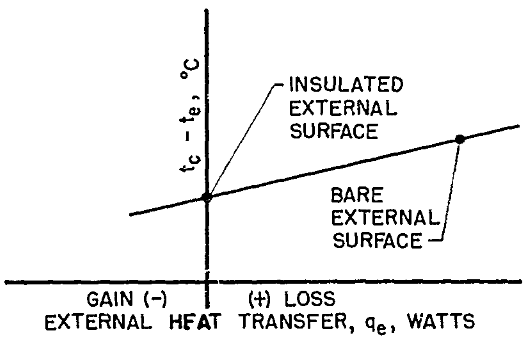


Figure VII-6. Working Plot for Evaluation of External Heat Transfer of Unit with Case-Envelope Heat Exchanger

The two values each of q_e and $(t_c - t_e)$ determined from these tests, permit construction of a working plot as illustrated in Figure VII-6. A straight line would be drawn through the two test points as a reasonably accurate approximation to the exact variation. Intermediate test points could be obtained from test of the unit having varying thickness insulation over the external surface. Extrapolation of the straight line shown in Figure VII-6 far into the region of heat gain is not recommended.

The heat transfer characteristics of the forced convective cooling occurring within the heat exchanger would be defined by an effective internal heat transfer coefficient H_{ex}. Using as the average temperature of the exchanger surfaces

$$t_{ex} = (t_c + t_e)/2, \qquad \text{(VII-12)}$$

and as an average temperature of the cooling air t_a the definition given by equation (VII-7), an average temperature differential for the forced convective heat transfer within the heat exchanger ($t_{ex} - t_a$) is defined and may be evaluated from the test data. The value of the effective heat transfer coefficient H_{ex} would be determined from the heat dissipation to the cooling air q_a and the temperature differential by

$$H_{ex} = q_a/(t_{ex} - t_a) \tag{VII-13}$$

An average value of H_{ex} for the two test runs would be used when applying corrective procedures to altitude chamber test data.

When the unit is operated in an environment different from that existing during test in the altitude chamber in respect to surrounding surface temperatures, but at the same ambient air pressure and temperature, the magnitude of the case temperature t_c would be determined by use of the working plot illustrated in Figure VII-6 and the effective heat transfer coefficient H_{ex}. The recommended procedure of evaluation is as follows. By combining equations (VII-1, -7, -12, and -13), the case temperature t_c is given by

$$t_c = \left[2/H_{ex} + 1/(456\ W)\right]q_a + 2t_1 - t_e \tag{VII-14}$$

Also, by use of the working plot illustrated in Figure VII-6, a value of the case temperature t_c may be determined from known values of q_e and t_e. Then, the procedure would be to assume a value of the external surface temperature t_e and proceed to calculate the external heat transfer q_e from the known heat transfer characteristics of the environment. The case temperature t_c would be calculated in two ways, from equation (VII-14) and by use of the working plot, Figure VII-6. The correct value of the case temperature would be obtained when the two calculated values of the case temperature agree.

The effect of change in ambient air pressure and temperature on the thermal state of the unit may be predicted approximately by use of the working procedure presented for this type unit. Any change in the flow rate of cooling air resulting from temperature or pressure change would be predicted on the basis of the change in air density, as discussed on page 225. The heat transfer coefficient H_{ex}, defined by equation (VII-13), would be corrected in direct proportion to the air rate to the 0.8-power, and the case temperature would then be evaluated by the prescribed procedure using Figure VII-6 and equation (VII-14). The method is limited to relatively small changes in air pressure and temperature since the relation between external heat transfer and the temperature differential ($t_c - t_e$), as shown in Figure VII-6, is not unique, but would be affected by the air rate in a manner impossible to predict from the test data. When the change in air pressure and temperature results in a reduction of air rate, the calculated case temperature will be on the conservative side because the external heat transfer would be underestimated. The case temperature will be predicted somewhat below its actual value when the air rate is increased. The limit to the use of this corrective procedure depends, of course, upon the desired accuracy in the predicted value of the case temperature, but, in general, would not be used if temperatures vary more than 20°C and pressures more than 10 per cent from the test values.

4. Examples

Example VII-1. Use of Altitude Chamber Test Data for Prediction of the Thermal State of a Sealed Unit when Installed in an Aircraft Compartment

A sealed unit having a circumferential baffle is tested in an altitude chamber to determine (1) rate of cooling air flow induced by the blower, and (2) the heat transfer characteristics of the unit. The tests are conducted at a chamber pressure of 2.13 inches mercury absolute, corresponding to an altitude of 60,000 feet, and a chamber temperature of 0°C.

The rate of heat generation by the unit is 500 watts. The case of the unit is cylindrical, 10 inches in diameter and 18 inches in length, and the baffle is separated from the case by a distance of 0.2 inches. All surfaces of the unit are painted black. The surface area of the case is 722 square inches. The external surface of the entire unit is 775 square inches, of which 588 square inches correspond to the surface of the cylindrical portion of the baffle.

a. Test Data. The first test of the unit is conducted with the external surfaces insulated. Steady-state values of the cooling air temperature at entrance to the unit and entrance to the blower for this test run are 0° and 25.5°C, respectively. A second test run is conducted with the external surfaces bare. Steady-state temperatures measured during this test are 0°C for the cooling air at entrance to the unit, 21.5°C for the baffle and 52.5°C for the case.

b. Reduction of Test Data

For the first test run the heat dissipated to the cooling air is equal to that generated by the unit. Hence, by use of equation (VII-1)

$$W = 500/456(25.5 - 0) = 0.043 \text{ pound per second}$$

the rate of cooling air supplied by the blower.

The heat dissipated to the cooling air during the second test run is

$$q_a = 456 \times 0.043(21.5 - 0) = 421.5 \text{ watts}$$

which means that the heat dissipated through the external surfaces of the unit is, by equation (VII-2)

$$q_e = 500 - 421.5 = 78.5 \text{ watts}$$

The conductive heat transfer between the case and the baffle is not considered in this example. The radiant heat transfer from the case to the baffle is defined by the case and baffle temperatures of 52.5° and 16°C, respectively, and the entire surface area of the case, 722 square inches.

From Figure VI-3b, $\Phi_2 = 0.162$. From Table VI-1 for black-painted surfaces and close confinement, $\Phi_1 = 0.90$. Hence, by equation (VI-4)

$$q_{c-rd} = 0.90 \times 0.162 \times 722 = 105.3 \text{ watts}$$

Thus, by equation (VII-3), the forced convective heat transfer from the case surface to the cooling air is

$$q_{c-a} = 500 - 105.3 = 394.7 \text{ watts}$$

and, by equation (VII-4), the forced convective heat transfer from the baffle surface to the cooling air is

$$q_{b-a} = 105.3 - 78.5 = 26.8 \text{ watts}$$

The average temperature of the cooling air is

$$t_a = (0 + 21.5)/2 = 10.75°C,$$

and the effective heat transfer coefficients for the cooling action between the case and air, and the baffle and air are, using equations (VII-8 and -9),

$$H_{c-a} = 394.7/(52.5 - 10.75) = 9.45 \text{ watts per } °C,$$

and

$$H_{b-a} = 26.8/(16 - 10.75) = 5.1 \text{ watts per } °C.$$

c. Use of Test Data

As an illustration of the use of the reduced test data, suppose the unit tested in the altitude chamber is installed in a compartment of an aircraft having the same air pressure and temperature as the altitude chamber, 2.13 inches of mercury absolute and 0°C. The air within the compartment is essentially motionless, so that heat transfer from the surface of the baffle is by free convection and radiation. The confinement of the unit is intermediate and the average temperature of the surrounding surfaces is 50°C, with the surfaces painted black.

Since air flows into one end of the unit and the blower is attached to the opposite end, it is assumed that free convective heat transfer occurs only from the cylindrical portion of the baffle, a surface area of 588 square inches. Radiant heat transfer is assumed to occur over the entire surface area of the unit, 775 square inches. Conduction is neglected. Because of the high temperature of the surrounding surfaces, heat will be radiated to the external surfaces of the unit, rather than from the surfaces to the environment. Free convective heat transfer will occur from the surfaces to the environmental air at 0°C, but because of the low air pressure the radiant heat transfer will prevail and result in a net heat gain by the unit through its external surfaces.

Following the evaluation procedure outlined on page 230, the baffle temperature t_b is assumed equal to 35°C. The free convective heat transfer from the baffle to the air is evaluated from Figure VII-2b, using a surface temperature of 35°C, an air temperature of 0°C, a pressure of 2.13 inches of mercury, and a horizontal cylinder with a significant dimension of 10.4 inches, the outside diameter of the baffle. The free convective heat transfer is 0.0225 watt per square inch, or a total of 0.023 x 588 = 13.5 watts. The radiant heat transfer from the surrounding surfaces to the external surface of the unit is evaluated from equation (VI-4), using Figure VI-3b and Table VI-1. From Table VI-1 for intermediate confinement and black painted surfaces, $\phi_1 = 0.93$. From Figure VI-3b for surface temperatures of 50° and 35°C, $\phi_2 = 0.069$. Hence, radiant heat transfer to the unit is 0.93 x x 0.069 x 775 = 49.7 watts. The net heat gain by the external surfaces is, then, -49.7 + 13.5 = -36.2 watts.

The heat dissipation to the cooling air is defined by equation (VII-2),

$$q_a = 500 - (-36.2) = 536.2 \text{ watts}$$

The average temperature of the cooling air is defined from

$$t_2 = t_1 + q_a/(456\ W) = 0 + 536.2/(456 \times 0.043) = 27.4°C$$

$$t_a = (t_1 + t_2)/2 = 13.7°C$$

The case temperature is calculated from equation (VII-10).

$$t_c = 536.2/9.45 - (5.1/9.54)(35 - 13.7) + 13.7 = 59.0°C$$

Next, the radiant heat transfer from the case to the baffle is determined from equation (VII-11).

$$q_{c-rd} = -36.2 + 5.1(35 - 13.7) = 72.4 \text{ watts,}$$

and since the surface area of the case is 722 square inches and the radiation factor $\phi_1 = 0.90$,

$$\phi_2 = 72.4/(0.90 \times 722) = 0.1115 \text{ watt per square inch}$$

Using this value of ϕ_2, the assumed baffle temperature of 35°C and Figure VI-3b, a second value of case temperature t_c of 59°C is determined which is the correct value since it agrees with the value calculated by equation (VII-10).

The results indicate that the temperature of the case increases from 52.5°C during test in the altitude chamber to 59°C when installed under the specified environmental conditions within the aircraft compartment. The effect of compartment environmental conditions being different from those in the altitude chamber is not too severe for a unit of this type since the heat concentration is rather high, about 610 watts per cubic foot of case, and requires, therefore, an appreciable quantity of cooling air to maintain permissible values for the temperature of the case surface.

For units having lower heat concentration, around 250 watts per cubic foot, the effect of environmental conditions is considerably more severe, as indicated by the following example. A unit identical to the one described in the above example but having a heat generation rate of 200 watts, rather than 500, requires 0.0108 pound of cooling air per second to maintain the case temperature at 55°C when the baffle is insulated. The unit described in the above example requires 0.043 pound of air per second to maintain a case temperature of 55°C with the baffle insulated. However, when the unit with 200 watts heat generation is tested in the altitude chamber, with the case bare, the case temperature drops to 47.5°C in comparison with 52.5°C for the unit generating 500 watts. The heat loss through the baffle during test in the altitude chamber is about 38 per cent of the generated heat, for the 200-watt unit and only about 16 per cent for the 500-watt unit. When the 200-watt unit is subjected to the compartment environmental conditions specified in the above example, the case temperature increases from 47.5° to 62.6°C, while the case temperature of the 500-watt unit was shown to increase only from 52.5° to 59°C.

Example VII-2. Evaluation of the Thermal State of a Sealed Unit when the Ambient Air Temperature Differs from the Altitude Chamber Temperature

The unit and altitude chamber test data described in Example VII-1 are used here to illustrate the method for predicting the case temperature when the air temperature differs from the value maintained during test. It is assumed that the unit is placed in an aircraft compartment having the same environment as described in part (c) of Example VII-1, with the exception that the ambient air temperature is 30°C rather than the test value of 0°C. The air pressure is maintained equal to the test value of 2.13 inches of mercury absolute.

The cooling air supplied by the blower at an ambient temperature of 30°C is determined according to the recommended procedure discussed on page 225. Since the air pressure remains constant, the density is inversely proportional to the absolute temperature, and the air rate at 30°C is

$$W = 0.043(273 + 0)/(273 + 30) = 0.03875 \text{ pound per second,}$$

where 0.043 represents the air rate determined from test in the altitude chamber at an air temperature of 0°C.

The effective heat transfer coefficients H_{c-a} and H_{b-a} are corrected for this change in air rate according to

$$H_{c-a} = 9.45(0.03875/0.043)^{0.8} = 8.69 \text{ watts per } ^{\circ}\text{C}$$

and

$$H_{b-a} = 5.1(0.03875/0.043)^{0.8} = 4.69 \text{ watts per } ^{\circ}\text{C,}$$

where 9.45 and 5.1 define their values at the test temperature of 0°C.

Since the ambient air temperature is quite high, the case and baffle temperatures would be expected to increase considerably over their values at $0^{o}C$. Hence, the baffle temperature t_b is assumed equal to $60^{o}C$ as a first approximation. Following the procedure outlined in part (c) of Example VII-1, the external heat transfer q_b is found equal to 48 watts. Hence, $q_a = 500 - 48 = 452$ watts. By equations (VII-5 and -7),

$$t_a = 30 + 452/(2 \times 456 \times 0.03875) = 42.8^{o}C,$$

and by equation (VII-10)

$$t_c = 452/8.69 - (4.69/8.69)(60 - 42.8) + 42.8 = 85.5^{o}C.$$

The radiant heat transfer from the case to the baffle is defined by equation (VII-11),

$$q_{c-rd} = 48 + 4.69(60 - 42.8) = 128.6 \text{ watts}$$

which from equation (VI-4) and Figure VI-3b defines a second value for the case temperature equal to $91^{o}C$. Repeating the process yields a final temperature for the case of $86.5^{o}C$ and for the baffle of $58.5^{o}C$.

Pressurized and Sealed Units with Integral or Separate Heat Exchanger

Units of this type dissipate the greater portion of the generated heat to cooling air flowing external of the pressurized or sealed case, by employing a heat exchanger which would either be integral with the unit's case or separate and located remote from the case. Any portion of the heat generated by the unit which is not dissipated through the heat exchanger is transferred through the surface of the case to the environment. General details regarding arrangement and heat transfer performance of this type unit are given in Chapter II, pages 9 to 11 and in Chapter VI, pages 178 to 197.

Altitude chamber tests for this type unit would be conducted for reasons similar to those for the pressurized and sealed units having a circumferential baffle or case-envelope heat exchanger. The rate of cooling air supplied by a blower of unknown performance may be determined by heat balance for any specified combination of air pressure and temperature within the altitude chamber. Also, since heat transfer through the case of this type of unit generally represents a significant portion of the total heat generated, it is necessary to correct the thermal performance determined during test in the altitude chamber to the environmental conditions which could be encountered in an actual installation.

The procedures and types of test recommended for this unit in Chapter IV, page 66, define the temperature of the internal air at entrance and exit of the heat exchanger, the temperature of the external cooling air at entrance and exit of the heat exchanger, the temperature of the case, and the temperature of the outer surface of any insulation used to cover the case.

1. Use of Test Data for Determination of Air Rate and Component Temperatures

The altitude chamber test procedure to determine the flow rate of external cooling air requires total insulation of all external surfaces of the unit. Thus, the total heat generated by the unit is dissipated in the heat exchanger to the external cooling air. The flow rate of external cooling air would be determined by heat balance, using equation (VII-1) and the measured values of the external cooling air temperature at inlet and exit of the heat exchanger. The discussion relating to the determination of air rate for units having a circumferential baffle or a case-envelope heat exchanger, given on pages 224 to 226 also applies to a unit of this type. Reference should be made to this material for additional details regarding evaluation of air rate from altitude chamber tests of this type.

The external cooling air rate determined from test of the unit in an altitude chamber at a specified combination of air pressure and temperature would be used in conjunction with bench-test data to determine the temperatures of various components within the case. The method for evaluating component temperatures from bench-test data is discussed in Chapter VI. Once the air rate has been established, the procedure would be to evaluate the internal air temperature t_{it-1} from bench-test data such as illustrated in plot C of Figure VI-18, after which the average temperature of the air within the component space t_{it-av} may be defined from equations (VI-15 and -17). The value of the internal air temperature t_{it-av} would then be used to define the various component temperatures, as illustrated by plot A of Figure VI-18. Component temperatures so determined define their maximum values for the specified combination of ambient air pressure and temperature, since the unit is insulated during test and there occurs, therefore, no heat transfer through external surfaces of the unit to its environment. An exception to this condition exists whenever the unit is subjected to an environment resulting in heat gain of the unit through the external surfaces. The problem of heat gain from the environment is discussed in the subsequent paragraphs dealing with corrective procedures for heat transfer through the external surface.

2. Use of Test Data for Determination of Heat Transfer Characteristics of Unit

As with the pressurized and sealed units having a circumferential baffle or case-envelope heat exchanger, it is necessary to construct from altitude chamber test data corrective procedures for units having integrated or separate heat exchangers which will permit evaluation of the thermal state of the unit when subjected to environments having the same ambient air pressure and temperature but different radiation and convection heat transfer characteristics.

Test procedures recommended in Chapter IV designed to secure test data for evaluation of the heat transfer characteristics of this type unit require one test run with the external surfaces of the case insulated, one test run with the surfaces bare, and one or several test runs with different thicknesses of insulation. Temperatures of the external cooling air at entrance and exit of the heat exchanger, t_{e-1} and t_{e-2}, temperatures of the internal cooling air at entrance and exit of the heat exchanger, t_{it-1} and t_{it-2}, and the temperature of the case of the unit are measured.

The corrective procedure for a unit having an integrated heat exchanger is obtained in the following manner. The external air rate W_e and the internal air rate W_{it} would be evaluated from the first test run where the unit is insulated, since the heat dissipated in the heat exchanger equals the heat generated by the unit. By heat balance

$$W_e = (q/456)(t_{e-1} - t_{e-2}) \qquad \text{(VII-15)}$$

and

$$W_{it} = (q/456)(t_{it-1} - t_{it-2}) \qquad \text{(VII-16)}$$

For each subsequent test run in which the external surface of the unit is bare or only partially insulated, the heat dissipated in the heat exchanger q_{ex} would be evaluated from equation (VII-15) using the measured values of inlet and exit cooling air temperature, t_{e-1} and t_{e-2}, and the previously established value of the external air rate. The difference between the heat generated q and the heat dissipated in the exchanger q_{ex} defines the heat transfer through the case of the unit q_c. Also, for each test run the average internal air temperature t_{it-av} would be defined from the internal air temperatures t_{it-1} and t_{it-2}, and the heat transfer parameter for the heat exchanger,

$$H_{ex} = q_{ex}/(t_{it-1} - t_{e-1}) \qquad \text{(VII-17)}$$

from the calculated values of q_{ex} and the measured values of the temperatures t_{it-1} and t_{e-1}. The parameter H_{ex} should remain constant for all tests conducted at constant values of the altitude chamber air pressure and temperature. An average value of this parameter would be employed when the values are found to vary for different tests at the same air pressure and temperature.

With some units it is entirely possible that accurate values of the case heat transfer would not be obtained by heat balance procedure, because of difficulties in measuring a true mean temperature of the cooling air. A check on the case heat loss is possible when the outer surface temperature of the insulation has been measured, by calculating the heat transfer through the insulation from a known value of its thermal conductance.

A working method for corrective procedure would be established by plotting the difference between the average internal air temperature and the case temperature $(t_{it-av} - t_c)$ as a function of the case heat transfer q_c. This is illustrated in Figure VII-7, where it is indicated that four test runs have been conducted in the altitude chamber.

The procedure for use of the working method when the unit is subjected to an environment different from that in the altitude chamber would be to assume a value for the temperature of the case t_c, to calculate the

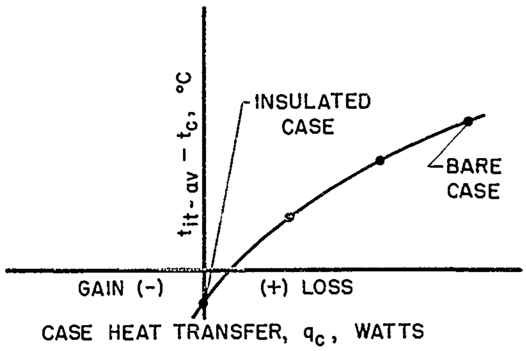


Figure VII-7. Working Plot for Evaluation of Case Heat Transfer of Unit with Integrated or Separate Heat Exchanger

case heat transfer q_c from the known heat transfer characteristics of the environment, and then to define a value of the average internal air temperature t_{it-av} from the working plot illustrated in Figure VII-7. Next, by combination of equations (VII-15, -16 and -17),

$$t_{it-av} = t_{e-1} + \left[1/H_{ex} - 1/(912\, W_{it})\right] q_{ex} \qquad \text{(VII-18)}$$

so that a second value for the average internal air temperature may be determined by this equation from the known value of H_{ex} and W_{it} and the heat dissipated in the heat exchanger q_{ex}, the latter being equal to the difference between the heat generated by the unit and the calculated value of the case heat transfer q_c. The correct value of the average internal air temperature is obtained upon agreement of the two calculated values. Component temperatures would be determined from the average internal air temperature and bench-test data.

When a separate heat exchanger is employed and the air ducts interconnecting the equipment and the heat exchanger are of appreciable length, it would be necessary to account for heat transfer which may occur between the surface of the ducts and its environment. A reasonably accurate method for including this heat transfer would be to include the surface of any ducts as part of the surface of the case.

The prediction of the thermal state of the unit when subjected to ambient air pressures and temperatures differing from those maintained during test in the altitude chamber may be conducted by approximate methods. When the ambient air temperature differs by less than 20°C from the altitude chamber value, and the environmental temperatures of the unit are known to be low relative to its case temperature, a simple rule may be applied stating that the average internal air temperature t_{it-av} will change by an amount approximately equal to the change in the ambient air temperature. This correction procedure is of fair accuracy whenever it is estimated that the heat dissipation in the heat exchanger represents more than 75 per cent of the heat generated by the unit. In general, it is not applicable to units exposed to environments having relatively high temperatures where heat gain through the external surfaces is likely to occur.

Whenever the air pressure and temperature of the actual environment differ from those maintained in the altitude chamber so as to produce a change in the external cooling air rate greater than 10 per cent of the test value, it is recommended that the corrective procedure be conducted along the following general lines. The external cooling air rate would be evaluated by assuming it to vary in direct proportion to the ambient air density, as discussed on page 225. The heat transfer parameter H_{ex} must then be adjusted to the new air rate, an adjustment which can only be evaluated accurately when bench-test data of the unit or manufacturer's performance data for the heat exchanger are available. Otherwise, approximate methods must be employed. If the heat exchanger has cross flow, is of tubular construction with external cooling air flow through the tubes, and the physical details of the tube bundle are known, the heat transfer parameter H_{ex} would be evaluated by the equation

$$H_{ex} = H_{ex\text{-}test} \left\{ \left[0.4\, A_e'^{0.8} / W_{e\text{-}test}^{0.8} + d_{it}'^{0.2}\, A_{it}'^{0.6} / W_{it}^{0.6} \right] \div \right.$$
$$\left. \div \left[0.4\, A_e'^{0.8} / W_e^{0.8} + d_{it}'^{0.2}\, A_{it}'^{0.6} / W_{it}^{0.6} \right] \right\} \qquad \text{(VII-19)}$$

where d_{it}' is the inside diameter of the tubes in feet, A_{it}' the cross-sectional area for flow through the tubes in square feet, A_e' the minimum cross-sectional area for flow over the outside of the tubes in square feet, and W_e and W_{it} the cooling air and internal air rates in pounds per second, respectively. Equation (VII-19) is quite accurate whenever the external air rate is nearly equal to or higher than the internal air rate. It should not be employed when a large change in the external air rate occurs, particularly when the external air rate is small in comparison with the internal air rate. The values of H_{ex} remain on the conservative side when the air rate is increased. When the heat exchanger is not of tubular construction so that equation (VII-19) would not apply, and neither bench-test nor manufacturer's data are available, the value of H_{ex} would be predicted by assuming it proportional to the air rate raised to the one-third power, a method yielding approximately correct results whenever the external air rate is from 75 to 125 per cent of the internal air rate, and the change in the external air rate is not greater than roughly 15 per cent of the original value.

Once the new values of external air rate W_e and the heat transfer parameter H_{ex} have been defined for the new combination of ambient air pressure and temperature, the evaluation procedure for determining the average temperature of the air within the component space of the unit is identical to that outlined previously on page 241 for this type unit.

3. Examples

Example VII-3. Use of Altitude Chamber Test Data for the Prediction of the Thermal State of a Pressurized Unit having an Integrated Heat Exchanger when Installed in an Aircraft Compartment

The unit is tested in an altitude chamber to determine its heat transfer characteristics. The problem is to determine from these heat transfer characteristics the thermal state of the unit as installed in an aircraft compartment having environmental conditions different than those within the altitude chamber.

The unit is tested in the altitude chamber at an air temperature of 30°C and an air pressure of 3.44 inches of mercury absolute, corresponding to an altitude of 50,000 feet. When installed in the aircraft compartment, the air temperature and pressure are predicted to be 35°C and 3.0 inches of mercury absolute, respectively, with the average temperature of the surfaces of surrounding units estimated at 50°C.

The case of the unit is cylindrical, having a length and diameter of 12 inches each, and the surface is painted black. The external surface area is 680 square inches. The integrated heat exchanger is of tubular construction, similar to that described in Example VI-7. A blower induces cooling air flow through the exchanger. The normal rate of heat generation of the unit is 800 watts.

a. Test Data

Two test runs are conducted in the altitude chamber at 800 watts heat generation rate, one with the case insulated and the other with the case bare. Average equilibrium temperatures measured during test are:

Temperature, °C	Case Insulated	Case Bare
External cooling air at inlet to exchanger, t_{e-1}	30	30
External cooling air at exit from exchanger, t_{e-2}	62	56.5
Internal cooling air at inlet to exchanger, t_{it-1}	109.5	95
Internal cooling air at exit from exchanger, t_{it-2}	80.3	72
Surface of case, t_c	103.5	56

b. Reduction of Test Data

Using the data of the test with the case insulated, by equation (VII-5),

$$W_e = 800/456(62 - 30) = 0.0548 \text{ pound per second}$$

and

$$W_{it} = 800/456(109.5 - 80.3) = 0.0601 \text{ pound per second}$$

From equation (VII-17),

$$H_{ex} = 800/(109.5 - 30) = 10.06 \text{ watts per } ^{\circ}C$$

Also,

$$t_{it-av} = (t_{it-1} + t_{it-2})/2 = (109.5 + 80.3)/2 = 94.9^{\circ}C$$

and

$$t_{it-av} - t_c = 94.9 - 103.5 = -8.6^{\circ}C$$

From the data secured during test with the case bare and by use of equation (VII-5),

$$q_{ex} = 456 \times 0.0548\ (56.5 - 30) = 662 \text{ watts}$$

$$q_{ex} = 456 \times 0.0601\ (95 - 72) = 630 \text{ watts}$$

Hence, a value of 650 watts is selected as the heat dissipated in the heat exchanger. The case heat loss is, then

$$q_c = 800 - 650 = 150 \text{ watts}$$

From equation (VII-17)

$$H_{ex} = 650/(95 - 30) = 10 \text{ watts per } ^{\circ}C$$

which agrees closely with the previously calculated value. Also,

$$t_{it-av} = (95 + 72)/2 = 83.5^{\circ}C$$

and

$$t_{it-av} - t_c = 83.5 - 56 = 27.5^{\circ}C$$

A working plot for evaluation of case heat transfer, as illustrated in Figure VII-7, is constructed from the values of q_c equal to zero and 150 watts and the corresponding values of $(t_{it-a} - t_c)$ equal to -8.6° and $27.5^{\circ}C$ by drawing a straight line between the two test points.

c. Use of Reduced Data

The environmental air pressure and temperature within the compartment are $35^{\circ}C$ and 3.0 inches of mercury absolute, respectively, in comparison with the test values of $30^{\circ}C$ and 3.44 inches of mercury absolute. The ratio of the air density in the compartment to that during test is, as shown by equation (A-I-1),

$$\rho_a/\rho_{a-test} = (3.0/3.44)\left[(273 + 30)/(273 + 35)\right] = 0.853$$

Assuming the external air rate to vary in direct proportion to the air density,

$$W_e = 0.858\, W_{e\text{-}test} = 0.858 \times 0.0548 = 0.047 \text{ pound per second}$$

The heat transfer parameter H_{ex} is corrected to the new value of the air rate by

$$H_{ex} = H_{ex\text{-}test}\, (W_e/W_{e\text{-}test})^{1/3} = 10(0.858)^{1/3} = 9.5 \text{ watts per } ^{\circ}\text{C},$$

as discussed on page 243.

The case temperature t_c is assumed equal to 75°C as a first approximation. Using the environmental radiation temperature of 50°C, the surface area of 680 square inches and ϕ_1 from Table VI-1 equal to 0.93, the radiant heat transfer from the case is determined from Figure VI-3b and equation (VI-4) equal to 90 watts. The free convective heat transfer from the case to the environmental air at 35°C is determined from Figure VI-2b for a surface temperature of 75°C, first for the cylindrical portion of the case and then the end plates. The free convective heat transfer is evaluated as 26 watts. Hence, the heat transfer from the case is 90 + 26 = 116 watts, or

$$q_{ex} = 800 - 116 = 684 \text{ watts.}$$

By use of equation (VII-18)

$$t_{it\text{-}av} = 35 + \left[1/9.5 - 1/(912 \times 0.0601)\right] (684) = 94.4^{\circ}\text{C}.$$

Then,

$$t_{it\text{-}av} - t_c = 94.4 - 75 = 19.4^{\circ}\text{C},$$

and from the working plot for case heat transfer constructed from the test data, at $(t_{it\text{-}av} - t_c) = 19.4$, $q_c = 117$ watts, which is in near agreement with the previously calculated value of 116 watts.

The average internal air temperature $t_{it\text{-}av}$ has increased from 83.5°C during test in the altitude chamber to 94.4°C when installed in the compartment. The rise in internal air temperature results from the higher ambient temperature, lower ambient pressure, and higher environmental surface temperatures than under test in the altitude chamber. Reduction of the internal air temperature to the altitude chamber value of 83.5°C could be obtained by lowering the ambient air temperature in the compartment. Calculations indicate the required ambient temperature to be about 28°C.

Vented Units with Closed Case Cooled by Free Convection and Radiation

Tests conducted in an altitude chamber on units of this type provide data permitting more accurate evaluation of the thermal state of a unit under actual operational conditions than do data obtained from bench test. This type of unit, being vented, maintains equal pressure inside and outside of the case, but is closed from the environment to the extent that no cooling air flows into or out of the unit.

Procedures of test for this type unit in an altitude chamber require measurement of component temperatures and the case temperature for various combinations of chamber air pressure and temperature. The tests should be conducted at air pressures and temperatures known to exist under actual installation conditions whenever it is intended to simulate such conditions in the altitude chamber. Otherwise, altitude chamber tests of the unit should be conducted over a range of air pressure and temperature sufficiently great to cover anticipated requirements.

Altitude chamber test data would be correlated by constructing a plot of measured temperatures of the various components as a function of the measured case temperature with ambient air pressure as a parameter, since the heat transfer process inside the case is affected by the pressure level of the internal air.

Altitude chamber test data correlated by this method would be used to predict the thermal state of the unit when subjected to installation environments different than those maintained during test. The temperature of the case would be evaluated from the known environmental conditions, the heat generation of the unit and an assumed value of the correction factor F_c, by use of the convection and radiation charts presented in Figures VI-2 and -3, following the procedure outlined in Example VI-1. The correction factor F_c applied to evaluation of heat transfer by free convection and radiation from external surfaces of a unit, as defined and discussed in Chapter VI, pages 137 to 138, would be assumed equal to unity whenever bench-test data on the unit permitting evaluation of its exact value are not available. Component temperatures are then defined by the correlation plot of component temperature as a function of case temperature and ambient air pressure as derived from the altitude chamber tests.

Example VII-4. Use of Altitude Chamber Test Data for the Evaluation of the Thermal Conditions of a Vented Unit with Closed Case when Installed in an Aircraft Compartment

A vented unit having a closed case is tested in an altitude chamber to determine its thermal condition when operated at an ambient air pressure of 2.13 inches of mercury absolute, 60,000 feet, and ambient air temperatures of 0°, 10°, 30° and 50°C.

The unit has a heat generation of 100 watts. The case of the unit is painted black and is rectangular, having a width of 10 inches, a height of 8 inches, and a length of 14 inches. The case does not seal tightly at the junction with the chassis, and, as a result, pressure equalization of the internal and external atmospheres is maintained, although no perceptible air flow occurs. Twenty-five thermocouples are located on the exterior surfaces of the case, five on each side panel and five on the top panel. The temperatures of all thermally critical components are also measured.

For the air pressure of 2.13 inches mercury absolute, average case temperatures of 20°, 28.5°, 47.5° and 66.5°C are measured at ambient air temperatures of 0°, 10°, 30° and 50°C, respectively. Component temperatures (not presented in this example) are plotted as functions of the case temperature. The resulting plots are valid for use in predicting component temperatures when the unit is operated under any environment at or near the test pressure of 2.13 inches of mercury absolute.

As an example of the use of these data, suppose the unit is operated in an aircraft compartment in still air with the surrounding surfaces at the same temperature as that of the air. The unit will be assumed small in comparison with the compartment, assuming essentially no obstruction to radiation, with the receiving surfaces being Dural. Heat is dissipated from the case surface to the environment by free convection and radiation, and is evaluated by use of Figures VI-2b and -3b and equation (VI-4), and an assumed value of the correction factor F_c equal to unity. This procedure is illustrated in detail in Example VI-1.

The average case temperatures are calculated to be 40.7°, 48.5°, 64° and 81°C for the ambient temperatures of 0°, 10°, 30° and 50°C, respectively, showing increases over the altitude chamber values of 20.7°, 20°, 16.5° and 14.5°C, respectively. In many installation compartments, close confinement created by other units having about the same case temperature would increase the calculated case temperatures by as much as 40° to 60°C, illustrating the important effect of the environment on this type unit. Component temperatures are defined from the correlation plots of component temperature versus case temperature at the specified ambient pressure.

<u>Vented Units with Open Case and Through-Flow of Atmospheric Air by Natural Convection</u>

As discussed in Chapter IV in reference to test procedures for this type unit, it is not practical to utilize analytical methods for the prediction of thermal conditions when environmental conditions differ from those maintained during test in the altitude chamber. Hence, installation conditions are to be simulated as nearly as possible during test in the altitude chamber.

A simple rule may be applied to units of this type for prediction of change in component temperatures resulting from change in ambient air temperature. Although it is only approximately correct, it may be assumed that the component temperatures would be changed in amount equal to the change in ambient air temperature. This assumption is conservative in considering an increase of ambient temperature, but is somewhat optimistic in considering a decrease.

<u>Vented Units with Open Case and Forced Through-Flow of Atmospheric Air</u>

Corrective procedures are required for interpretation of data obtained during altitude chamber test of open units having forced through-flow of atmospheric air. The corrective procedures differ in accordance with the method by which air is admitted to and discharged from the case.

1. Units Admitting and Discharging Air Through Opposite Ends of the Case

Most units relying on direct through-flow of air for dissipation of heat generated by the components admit the air in one end of the case and discharge it through the opposite end. The air may be admitted and discharged through a single inlet and a single outlet with either forced or induced flow created by a blower. Also, with a blower inducing air flow through the unit, the air may be admitted through multiple openings in one end of the case and discharged through a single outlet in the opposite end. With a forced flow arrangement, the air may be admitted through a single inlet and discharged through multiple openings. Whatever the arrangement of inlet or outlet of cooling air through opposite ends of the case of the unit may be, procedures can be established based upon altitude chamber test data permitting fairly accurate prediction of the thermal state of the unit when subjected to installation environments different than those maintained within the altitude chamber.

Test procedures in the altitude chamber recommended in Chapter IV for each combination of air pressure and temperature require a test run with the case insulated and one with the case bare. Temperatures of the air at exit of the case, inlet to the case and of the case surface would be measured for each test run.

The air rate would be defined from the test run with the unit insulated by use of equation (VII-1). When the unit employs multiple openings at inlet or discharge in one end of the case, it is not possible to prevent radiant heat exchange between that end of the case and its environment. Hence, although usually of minor importance, the radiant heat transfer from the end of the case should be evaluated and be subtracted from the heat generated by the unit in order to define a correct value of the heat dissipated to the cooling air. The radiant heat transfer would be evaluated by use of equation (VI-4) and the chart given in Figure VI-3b.

Once the air rate is defined from test of the insulated unit, the case heat exchange with the environment is defined by heat balance and by the data obtained during test with the external surfaces bare. The procedure is identical to that outlined previously in this Chapter for other units, with the exception that when induced flow of the cooling air is employed the temperature, and correspondingly the density, of the air entering the blower may be different for the insulated and bare test runs to the extent that correction of the air rate obtained in the insulated test is required. The air rate obtained when the external surfaces are bare would be evaluated by multiplying the air rate defined from the insulated test with the ratio of the air density entering the blower when the surfaces are bare to the air density at the same point when the surfaces are insulated. Neglecting a minor change in pressure drop of the air across the unit,

$$W = W_{ins}\,(273 + t_{2\text{-}ins})/(273 + t_2) \qquad \text{(VII-20)}$$

where W and t_2 represent values of air rate and exit temperature of the cooling air with the surfaces bare.

The procedure for correlating the heat transfer data would follow that outlined for this type unit in Chapter VI, pages 207 to 221. The effective temperature of the components t_{ef} would be defined for the evaluated air rate, either from a correlation obtained from bench tests of the unit or from component temperatures measured during test in the altitude chamber. The heat transfer parameter $q_a/(t_{ef} - t_1)$ may then be defined, as illustrated in plot B of Figure VI-29. Case heat transfer q_c would be correlated as a function of the difference between effective temperature and average case temperature, as illustrated in plot C of Figure VI-29. Since only two test points would normally be available from altitude chamber tests, it is recommended that the variation of q_c with $(t_{ef} - t_c)$ be established by a straight line between the two points.

The use of these data to correct the thermal state of a unit defined by test in the altitude chamber to any other environment would be conducted by the procedure outlined in Chapter VI. It is recommended that the corrective procedure be not applied to environments having ambient air pressure and temperature greatly different from those maintained during test in the altitude chamber. Minor changes may be accounted for by correcting the air rate in proportion to the change in air density, and the heat transfer parameter $q_a/(t_{ef} - t_1)$ proportional to the air rate raised to roughly the 0.7 or 0.8 power.

2. Units Admitting and Discharging Air Through Ends and Side Panels of the Case

A few units relying on cooling by direct through-flow of atmospheric air admit the air through louvered openings in the end and side panels of the case with induced flow, or discharge the air through similar openings with forced flow. Corrective procedures recommended for units of this type should be recognized as being only approximately correct.

Since it is not possible to insulate the case without changing the air flow resistance, the unit must be tested in the auxiliary test box described in Chapter IV, page 70. The temperatures measured at inlet and exit of the box would be used in conjunction with equation (VII-1) to define the air rate supplied by the unit's blower. Heat dissipation from the external surfaces of the unit would be assumed to take place only by radiation. If the average temperature of the case has not been measured during test, it would be assumed equal to the average of the inlet and exit temperatures of the cooling air. Component temperatures would be correlated directly with the chamber air pressure and temperature.

Data secured during test in the altitude chamber would be used to predict the thermal conditions in other radiation environments by evaluating the effect of case heat transfer on the average temperature of the cooling air. The case heat transfer would be evaluated by assuming the case surface at the average temperature of the cooling air, and assuming that radiant heat transfer occurs from the entire surface of the case, but free convective heat transfer occurs only from those portions of the case which have no louvered openings and are not in the path of the discharging air. A trial-and-error process of calculation is required. The component temperatures would be assumed to change by the same amount as the average temperature of the cooling air. The corrective procedure is limited to compensation for effects of different radiation environment and is not recommended for evaluation of the effect of change in ambient air pressure and temperature from those maintained during test in the altitude chamber.

CHAPTER VIII

TEST AND EVALUATION METHODS FOR DETERMINATION OF THERMAL CONDITIONS DURING NON-STEADY STATE OPERATION

All airborne electronic equipments undergo a period of non-steady state thermal performance after operation is initiated. During this period the temperatures of the case and of the various components, the heat dissipated from the components to the internal atmosphere of the unit, the heat dissipated or received by the external surfaces of the unit, etc., vary with time of operation. This period of non-steady state thermal performance may last as long as a few hours under fixed environmental conditions before all temperatures and heat transfer rates become invariant with time and the unit is said to have reached thermal equilibrium with its environment. In thermal equilibrium, the heat generated within the unit in addition to any which might be received from sources external of the unit equals the heat dissipated by the unit to its environment from case and component surfaces and by all possible modes of heat transfer. Environmental limits for satisfactory thermal performance in the steady state can be defined by the test and evaluation procedures discussed in Chapters IV, VI and VII.

In some applications, the environmental conditions during the transient thermal state are more severe than those permitted for satisfactory thermal operation in the steady state. This condition may be brought about by the flight speed of the aircraft or by extremely confined installation conditions. As a unit continues to operate in such an environment, the specified limiting component temperatures would be exceeded before thermal equilibrium would be reached. However, if the unit must operate for a limited time only, either because of intermittent use or short flight duration, limiting temperatures may not be reached. Therefore, evaluation procedures which would permit the determination of temperature-time histories of all types of units and their basic components, operating under any specified environmental condition, are of great importance in equipment application. Utilizing such procedures it should be feasible to ascertain for every application of specified operating time, the most compact physical configuration and the necessary provisions for heat dissipation of least complexity which would permit operation up to the maximum allowable temperatures. Necessary modifications would become apparent from the results of analysis. They may point out the feasibility of reducing over-all dimensions of a system installation and/or of reducing ventilation or even sealing of installation compartments if temperature limits are not reached. Oppositely, if they are exceeded, it may be indicated that additional ventilation, or some means of insulation on compartment walls or equipment cases would be required.

The ultimate objective in the analysis of the transient thermal state of an electronic unit is to predict the temperature-time history of all critical basic components, contained in the unit. Attainment of this objective depends largely on the ability to predict the temperature-time history of the unit's case and associated heat transfer elements, as well as the transient performance of blowers and motors which may be used for the supply of cooling air. Purely analytical prediction of temperature-time histories can give approximately correct results when the units are of simple arrangement and geometric form, and subjected to environments remaining constant with time. However, in most instances the analytical tools must be supplemented with experimental test data of the units to determine reference transient characteristics or appropriate physical constants. It is to be noted that apart from the complexities resulting from unknown distribution of heat loss or gain by the various modes of heat transfer which to a considerable extent defy exact analytical solution in the steady state, a purely analytical approach for non-steady state evaluation is made particularly difficult because of the time variable involved. Furthermore, with variable environmental conditions which will often occur during non-steady state operation, the cooling conditions defined by the ambient air pressure and temperature, the temperature of radiation-receiving surfaces, or the air rate delivered by a blower will vary with time of operation.

The procedures set forth in this chapter are suggested as practical methods for predicting the thermal state of a unit during non-steady state operation. They should be recognized as not being exact, but designed to produce results having fair accuracy and meeting the need of pre-operational knowledge of the non-steady state temperatures.

In the subsequent presentation of methods for determination of the transient thermal performance of specific types of units, references to test procedures and analytical procedures for each are treated together. This is done since it is felt that the analyst would be required to assume the principal responsibility. Therefore, he would find it necessary to guide experimental personnel in the procurement of test data which are needed to supplement the analytical procedures.

In defining the characteristics of the various types of units analyzed, features which could be incorporated in designing specifically for non-steady state operation, e.g., insulation, are not included. It is intended to treat such features and their effect on thermal evaluation in a future supplement of this manual dealing with equipment modifications. Thus, the analytical methods presented in this chapter are principally intended to be applicable to existing equipment or units designed for short-duration application but having essentially the same characteristics.

Basic Theory of Non-Steady State Operation

For purposes of predicting non-steady state temperatures of various components of an electronic equipment, the unit would be idealized by considering it to have the following characteristic parts: (1) a heat source generating heat equal to the difference of the electrical input and the electrical output, (2) a heat-absorbing body or bulk having a constant thermal capacity K, and (3) an external surface through which heat is rejected to or received from the environment by radiation, free or forced convection, or combinations thereof.

The principle of conservation of thermal energy is a law which applies to any unit, irrespective of whether the unit is operating in the transient or steady state. It permits keeping a record of the sources and sinks for all heat involved in the over-all process at any instant during operation, whether heat is generated by the unit, received by the unit from an external source, rejected by the unit to an external sink, or remains in the unit by being absorbed by the thermal capacity of its bulk. The unit's bulk may store or absorb heat, or it may give up heat, but is herein generally referred to as a positive quantity, meaning that heat is absorbed. Hence, a general energy balance for a unit at any instant during operation is a statement that the rate of heat generation q plus the rate of heat received q_{rc} minus the rate of heat dissipated q_{ds} equals the rate of heat absorbed q_{ab}. The interrelation of these various energy transfers is described by the following equation.

$$q + q_{rc} - q_{ds} = q_{ab} \qquad \text{(VIII-1)}$$

During steady-state operation the rate of heat absorption q_{ab} is zero since the temperature remain constant with time, and all heat generated or received by the unit must be dissipated to the available heat sinks. In the transient thermal state, however, the temperatures of the various components and the structure of the unit are varying, and heat is being absorbed or given up by the unit's bulk, depending upon whether the average temperature of the bulk is increasing or decreasing. A condition in which the unit gives up heat, i.e., q_{ab} is negative, and the bulk temperature is decreasing, is sometimes encountered because of rapid change in the state of the environment, or the initiation of operation of a cooling system after the unit has operated for a period of time, or because of intermittent operation.

Non-steady state operation creating a transient thermal state of the unit can exist under fixed environmental conditions, but would always exist under varying environmental conditions. A transient thermal state may be created in a fixed environment by the addition of a heat source within the unit, i.e., by initiation of operation. The transient state would continue only until the unit's average temperature reaches a value permitting dissipation of the heat generated and received by the unit to the available sinks. On the other hand, operation in a changing environment creates a transient thermal state lasting always somewhat longer than the period during which the environmental conditions are varying. Then, the internal heat generation may or may not be the principal factor in establishing the time period during which a transient thermal state occurs. Equipment operating in aircraft having substantially constant flight conditions, or operating on the ground, would have transient thermal states created principally by initiation of their operation. Equipments operating in aircraft having widely varying flight conditions would have transient thermal states created by both their internal heat sources and changing environmental conditions.

The amount of heat absorbed or given up when the bulk temperature of a unit increases or decreases, respectively, is a function of its thermal capacity. The thermal capacity of a body is defined as the product of its specific heat and weight, and defines the heat required to create a temperature change of one degree of the body. Thermal capacity is a principal factor in effecting the rate of temperature change of an electronic equipment. When heat is absorbed by a unit, for example, the various temperatures rise at a rate inversely proportional to the thermal capacity, i.e., the higher the value of thermal capacity the lower the rate of temperature increase, and vice versa. If a unit has a very high thermal capacity the temperatures will appear to have remained essentially unaltered shortly after operation is initiated, since the bulk would be capable of absorbing considerable heat with very little change in temperature. On the other hand, if the thermal capacity of a unit is very low the temperatures will change very rapidly, and would reach their equilibrium or steady-state values shortly after operation of the unit is initiated. In the latter case, the heat generated would be dissipated almost immediately through the heat transfer surfaces of the unit to the environment.

The rate of heat absorption q_{ab} of a unit undergoing transient thermal operation is related to the thermal capacity K_{cp} and temperature t_{cp} of the various components and structural elements by the equation

$$q_{ab} = d/d\tau \; \Sigma \, (K_{cp} t_{cp}) \qquad \text{(VIII-2)}$$

This expression cannot generally be solved because of the many different variations of temperature for the various components in a unit resulting from the extremely complex interaction of heat transfer by conduction, convection and radiation.

A practical approach to the evaluation of non-steady state performance of any unit is the calculation of the time history of a representative temperature which is the basis for evaluating the rate of heat dissipation q_{ds} and the rate of receiving heat q_{rc}. Since the rate of heat generation q is usually independently defined, knowledge of q_{ds} and q_{rc} permits solving equation (VIII-1) for the rate of heat absorption q_{ab}. Thus, in order to make equation (VIII-2) dependent on a single temperature, it is convenient to express the unit's thermal capacity not in terms of a true value, but as an equivalent value K_{ev} corresponding to the representative temperature t_{rp}. The interrelationships of actual and assumed terms is defined by the equation

$$K_{ev} \, t_{rp} = \Sigma \, (K_{cp} \, t_{cp}) \qquad \text{(VIII-3)}$$

Then, equation (VIII-2) becomes

$$q_{ab} = d/d\tau \, (K_{ev} \, t_{rp}) = K_{ev} \, (dt_{rp}/d\tau) \qquad \text{(VIII-4)}$$

This assumes that the equivalent thermal capacity is constant and independent of temperature. This assumption is fairly accurate for the types of materials used and the relative temperature levels of the components and the representative surfaces in the range of operating conditions normally to be considered. In operations where the increase of representative temperature would exceed 200°C the assumption would become inaccurate and would produce slightly erroneous results in the direction of lower representative temperature.

The representative temperature t_{rp} to be used for each type of unit must be suitable not only for definition of q_{ds} and q_{rc} but, most important, must provide means for the determination of component temperatures. As pointed out above, the complexity of the heat transfer mechanism makes a direct solution for changes in individual component temperatures with time practically impossible. However, the proper choice of the representative temperature t_{rp} makes it possible to determine component temperatures on the basis of supplementary experimental data. The case temperature is best used for pressurized or sealed units cooled by radiation and free convection, or forced convection, and for closed vented units. The average internal air temperature is appropriate as a representative temperature for pressurized or sealed units with integrated or separate heat exchanger. The effective component temperature, being the average of several component surface temperatures, is most significant for non-steady state analysis of vented units with natural or forced through-flow of atmospheric cooling air.

Determination of the equivalent thermal capacity for all types of units is an experimental process which may be incorporated readily into other tests necessary to provide the data on component-to-representative temperature correlation, as described subsequently for each type of unit. Additional measurements of variation of power input, power output, flow rates, and environmental conditions with time must be made.

In general, the procedure consists in determining the total amount of heat absorbed by the unit in reaching an equilibrium thermal condition while operating in a fixed environment with which it was originally, before operation was initiated, also in equilibrium. Since all temperatures are transient during this equilibrium period after operation is initiated, the rates of heat absorption and heat dissipation are also transient. The total quantity of heat E absorbed during the period may be calculated by mechanically integrating the curve of instantaneous heat absorption determined from the difference of the rates of measured total heat generation and calculated heat dissipation. Thus,

$$E = \Sigma q_{ab-av} \Delta \tau \qquad \text{(VIII-5)}$$

where q_{ab-av} is the average heat absorption rate taken as the average of the value at the beginning and end of each time interval. The choice of the length of the time intervals must be made on basis of the variation in slope of the heat absorption curve to avoid too great an error in the direction of increased E.

The equivalent thermal capacity K_{ev} is determined from this heat absorption and the total rise of representative temperature up to equilibrium by

$$K_{ev} = E/(t_{rp-eq} - t_{rp-0}) \qquad \text{(VIII-6)}$$

Methods applicable to the determination of K_{ev} and examples illustrating their use are discussed in subsequent sections dealing with the analysis of transient thermal performance of specific types of units.

Pressurized and Sealed Units Cooled by Free Convection and Radiation

The temperature-time history of a pressurized and sealed unit operating in an environment such that free convective and radiant heat transfer occurs over its external surfaces may be predicted by relatively direct procedures having fairly good accuracy. They may be applied to units operating in fixed or variable environments with constant or variable heat generation. Thus, the transient thermal state may be evaluated when created by initiation of operation of the unit only, by variation in the environment resulting from change in the operational conditions of the aircraft, by variation of the heat generation due to change in function or by combinations of the three.

1. Test Procedures

Bench tests of the unit are required to provide basic temperature data needed for evaluation of non-steady state operation. However, the test data may be secured when the unit is bench tested for determination of the thermal characteristics during steady state operation, as discussed in Chapter IV, so that in effect no additional tests of the unit are required. Temperature-time histories of the case and all thermally critical components should be determined by actual measurements taken from the time when operation is initiated and all parts of the unit are at the same temperature as the environment, until thermal equilibrium is again attained while operation continues. The environment should preferably be maintained at constant temperature. However, corrections can be made in the interpretation of the data for small changes in environmental conditions during test.

It is to be pointed out that, although the theoretical time to thermal equilibrium is infinite, the time to equilibrium for practical interpretation of non-steady state operation is defined by the time required for a case or component to attain a constant temperature within the accuracy limitations of the measuring instrument. The case of a unit operating at a temperature within one degree of its measured equilibrium temperature has, for all practical purposes, reached the steady state. Exact definition of component equilibrium time by bench test of the unit is quite difficult. Therefore, when needed in order to more specifically define equilibrium time from bench test data, the analyst may desire to select the time corresponding to a component temperature rise of 98 or 99 per cent of the equilibrium component temperature rise, the latter having been determined from a steady-state test in the same environment. Most of the evaluation methods presented in the subsequent paragraphs do not require knowledge of equilibrium time for the various components.

Case and component temperatures should be measured according to the methods recommended in Chapters III and IV. It is particularly important that the case temperature be measured at a number of uniformly spaced positions over the case surface, in order that a representative value of the average case temperature may be determined.

To provide data for the determination of the unit's equivalent thermal capacity, as outlined on page 255, the variation of power input and output with time must also be determined, as accurately as practical, by measurements.

2. Evaluation of Equivalent Thermal Capacity

In accordance with the principles discussed on pages 255 and 256, being the basis for evaluation of equivalent thermal capacity by test, it is necessary to reduce the test data taken in the transient bench test in such a manner that the rate of heat absorption of the unit at any time during the bench test can be determined by calculation. Since the rate of heat generation is known by the difference of measured power input and output, equation (VIII-1) would give the solution for instantaneous rates of heat absorption, if the instantaneous rates of heat dissipation are calculated. In bench test, q_{rc} in equation (VIII-1) would normally be zero, if the test has been performed with the necessary care of screening the test area from air currents and from surfaces at temperatures substantially higher or lower than the air temperature, the rate of heat dissipation from the case, $q_{ds} = q_c$, can be calculated by using the free convection charts of Figure VI-2 and the radiation equation (VI-4). The necessary correction factor F_c to compensate for the deviation of actual from calculated conditions is determined on basis of the end conditions of the bench test at which equilibrium must exist, the rate of heat absorption must be zero, and the rate of heat dissipation by free convection and radiation must equal the rate of heat generation. The correction factor F_c so determined can be assumed with good accuracy applicable to the heat dissipation from the case during the entire test period from initiation of operation to thermal equilibrium.

From a plot of the measured case temperature t_c versus time τ, the corresponding variation in the rate of heat dissipation with time is calculated, by means of Figure VI-2, equation (VI-4) and the value of F_c previously determined. The calculation of instantaneous rates of heat dissipation must be performed for a sufficient number of conditions during the test to establish a reliable curve of q_c versus τ. Since q versus τ is available from test measurements, the variation of rate of heat absorption q_{ab} would be determined by equation (VIII-1). The total heat absorption E during the test is then determined by means of integrating the q_{ab}-variation mechanically, using equation (VIII-5). The curve of q_{ab} versus τ may also be integrated graphically by use of a planimeter. The equivalent thermal capacity K_{ev} is determined from equation (VIII-6) based on the difference of the case temperature, which is the representative temperature ($t_c = t_{rp}$), at the end (equilibrium) and beginning of the bench test.

A numerical illustration of the procedure described above is contained in Example VIII-1, page 266.

3. Methods of Evaluating Component Temperature Variation in Non-Steady State Operation

In this type of unit, component temperatures at any time of operation may be predicted from the momentary average case temperature, in one of two ways. The first method would be used whenever transient temperature data are available for the construction of temperature-time curves. The second method is more approximate, but does not require as extensive bench test data as the first. It simplifies considerably the evaluation procedures, especially when the unit is subjected to a transient environment. Both methods are, however, subject to certain limitations on accuracy whenever operational temperature conditions of the unit differ greatly from those maintained during bench test.

a. Evaluation on Basis of Transient Temperature Curves

Characteristic temperature patterns of the various thermally critical components are defined from the temperature-time histories obtained during bench test by plotting for each component

$$(t_{cp} - t_{cp-\underline{0}})/(t_{cp-eq} - t_{cp-\underline{0}}) \text{ versus } (t_c - t_{c-\underline{0}})/(t_{c-eq} - t_{c-\underline{0}})$$

The first parameter is referred to as the component temperature parameter and defines the temperature rise of a component at any time over its temperature at time zero as a fraction of the component temperature rise under conditions of thermal equilibrium. The second parameter, referred to as the case temperature parameter, defines, similarly, the temperature rise of the case at any time over its temperature at time zero as a fraction of the case temperature rise under conditions of thermal equilibrium. Since data for evaluation of these parameters are obtained from bench test of the unit, where prior to operation the component temperatures are equal to the case temperature, the initial values of $t_{cp-\underline{0}}$ and $t_{c-\underline{0}}$, i.e., at time $\tau = 0$, must be taken as being equal for general application. This means, therefore, that for use of these data to evaluate other conditions, it must be assumed that the unit is at a uniform temperature just before operation is initiated. The characteristic temperature patterns defined by the parameters would not be applicable for evaluation of component temperature variation when the unit is not operating but is subjected to a variable environment which might result in heat gain or loss by the bulk of the unit. They are applicable to fixed or variable environments, but require uniform temperatures throughout when operation of the unit is initiated.

Characteristic temperature patterns established by plotting the component temperature parameter as a function of the case temperature parameter are illustrated in Figure VIII-1. Both parameters have values of zero at the time operation is initiated and of unity at the time thermal equilibrium is reached. A component whose temperature varies with time in a manner identical to the variation of the case temperature has a characteristic temperature pattern corresponding to the dashed line. A component having relatively low thermal capacity but operating at relatively high surface temperatures would have a characteristic temperature pattern similar to components

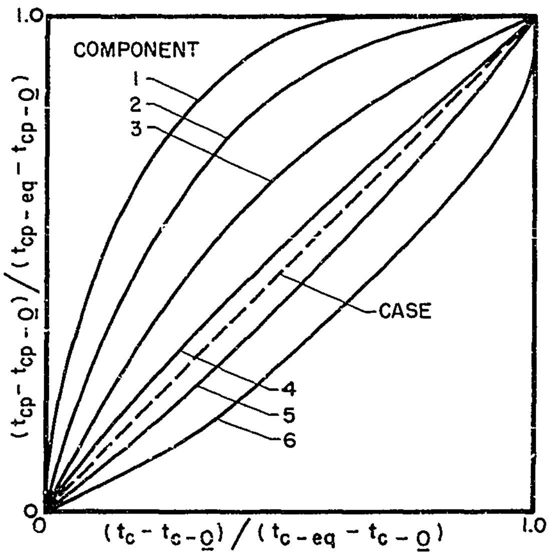


Figure VIII-1. Typical Characteristic Temperature Patterns for Various Components in a Pressurized and Sealed Unit

(1) and (2) of Figure VIII-1. These patterns correspond to those commonly found for vacuum tubes, since their ratios of heat dissipation per unit surface area to thermal capacity are normally quite high. Consequently, their surface temperatures during the early stages of operation increase at a rate greater than that for the temperature of the case. The time of operation required for a tube to attain a temperature closely approaching that at conditions of thermal equilibrium is somewhat less than that for the case of the unit, since a tube's thermal capacity per unit heat generation is low and heat is dissipated from the tube's surface to a considerable extent by radiation. Therefore, once the surface temperature becomes quite high, any change in the case temperature produces only a small change in the tube's surface temperature. Tubes having larger thermal capacity per unit heat generation and smaller surface area have characteristic temperature patterns approaching in shape that illustrated for component (3) in Figure VIII-1.

The characteristic temperature pattern of component (3) in Figure VIII-1 also illustrates that to be expected for some resistors, or similar components. The thermal capacity per unit heat generation is greater than that for a tube. Therefore, the component temperature during the early part of operation would be lower. However, since the heat generation per unit surface would usually be greater, the equilibrium temperature would exceed that of a tube. Heat dissipation from this type of component would be by convection, conduction and radiation, with the resulting surface temperature variation being more like that for the unit's case. It would normally rise above that for the case, as noted by comparing the line for component (3) in Figure VIII-1 with the dashed line, since its thermal capacity is relatively small. The time to equilibrium for this type component would be comparable to that for the case.

A non-heat producing component having relatively small thermal capacity would be expected to have a characteristic temperature pattern similar to that for component (4) or (5) in Figure VIII-1. If the component is exposed to high-temperature surfaces, such as several nearby vacuum tubes, considerable heat will be received by the component during the early stages of operation because of radiant heat transfer, and the surface temperature would initially increase at a rate somewhat greater than for the case. Should a similar component be shielded from receiving direct radiation from other components, the characteristic temperature pattern may resemble more nearly that for component (5) in Figure VIII-1. The rate of temperature rise of this type component is somewhat lower than for the case, because its gain in heat is principally by convection and conduction, whereas the case receives heat by convection, conduction and radiation. The actual shape of characteristic temperature patterns for non-heat producing components depends greatly upon their position in the unit, type of mounting, number and size of thermal buses connecting the component to other elements of the unit, and their ratio of surface area to thermal capacity. The time required for a non-heat producing component to reach thermal equilibrium is on the same order of magnitude as for the case or bulk of the unit.

Components such as transformers, having high thermal capacity relative to their internal heat generation, would be expected to have a characteristic temperature pattern similar to that shown for component (6) in Figure VIII-1. The large thermal capacity results in a lag of the component temperature relative to the case temperature. Also, the time to reach a certain degree of equilibrium is for this type component usually greater than for the case or bulk of the unit. The equilibrium temperature of the component would be higher than the equilibrium temperature of the case because of the temperature difference between the component and the case required to dissipate its internal heat generation during steady-state operation.

The use of the characteristic temperature patterns to determine component temperatures during non-steady state operation requires only knowledge of the equilibrium component and case temperatures and the variation of the case temperature with time of operation. Equilibrium component and case temperatures must be obtained by calculation. They are temperatures that would be obtained under given environmental conditions if the unit would be allowed to reach equilibrium conditions, like in the bench test from which the basic data are evolved. The methods presented in Chapters VI and VII are applicable to the determination of equilibrium temperatures. The determination of case temperature variation is discussed in subsequent paragraphs.

Using the above procedure, it would be assumed that the characteristic temperature patterns illustrated in Figure VIII-1 remain the same as determined from bench test for all environmental conditions to which a pressurized and sealed unit may be subjected. Actually, they would vary to some degree as the average temperature level of the unit is changed, since the internal heat transfer coefficients of radiation and free convection vary with temperature level. However, any variations of heat transfer mechanism within the unit resulting from environments usually encountered, are of minor importance to the prediction of component temperatures on the basis of case temperature.

b. Evaluation on Basis of Equilibrium Temperature Rise Data

The second method of predicting component temperatures during transient operation requires only knowledge of the equilibrium temperatures of the case and components and the temperature-time variation of the case. As previously pointed out, this method is of more approximate nature but does not require bench test determination of the characteristic temperature patterns. First, the case temperature variation with time would be defined. Then, it is assumed that the difference in temperature between any component and the case at equilibrium remains constant at all other times of operation. A temperature-time curve would be constructed for each component parallel to the temperature-time curve of the case and at a higher temperature by the amount the equilibrium temperature of the component exceeds the equilibrium temperature of the case. However, since it is known that the temperature of all components equals that of the case at time zero, a second curve originating from the initial case temperature would be drawn to fair in with the first component temperature curve which runs parallel to the case temperature curve. This construction is illustrated in Figure VIII-2 by the dashed lines for two types of components. The solid lines depict the actual temper-

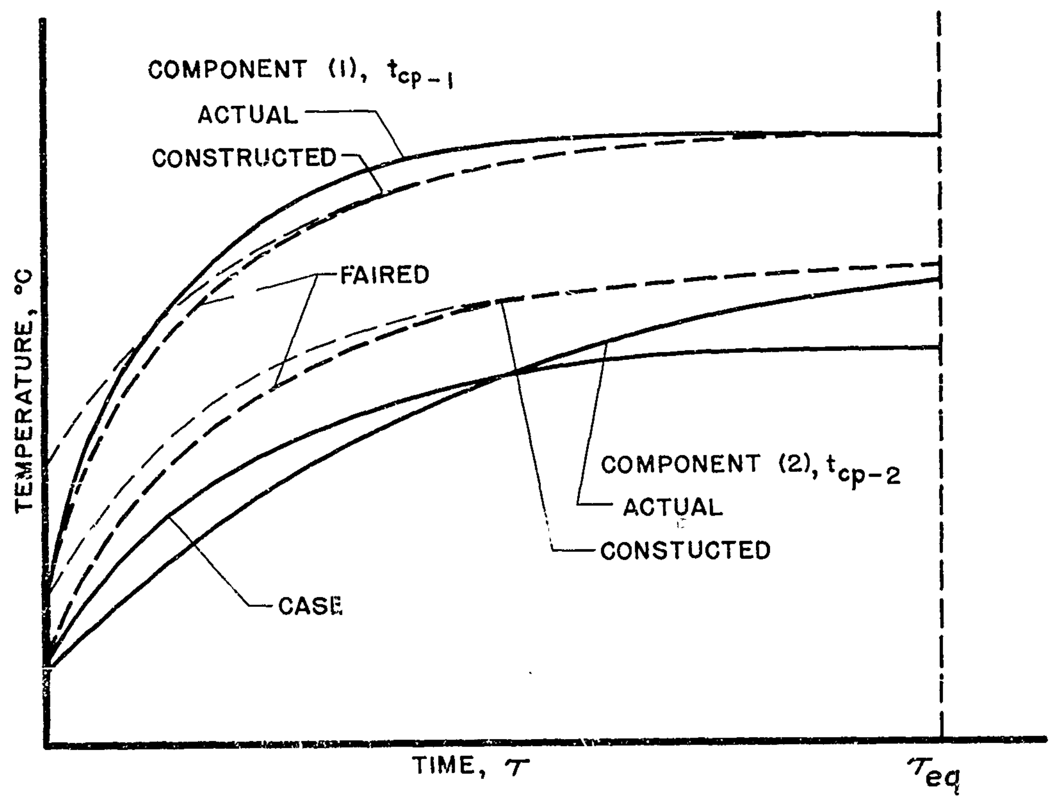


Figure VIII-2. Prediction of Component Temperatures during Non-Steady State Operation by Approximate Method ature-time variation for the case and for components (1) and (2). Component (1) is taken to represent a high-heat producing element having relatively low thermal capacity such as a tube, while component (2) is intended to represent an element having high thermal capacity, such as a transformer. Comparison of the actual temperature variation with the predicted variation shows that for a component similar to component (1), this method may yield temperatures lower than actual and, therefore, on the unsafe side, since the difference between the component and case temperatures may at some time of the operation actually be greater than at thermal equilibrium. This greater difference in temperature is attributable to the fact that the radiant heat transfer coefficient increases with increasing temperature level. On the other hand, for a component similar to component (2), such as a transformer, the predicted temperatures would always be lower than actual and on the safe side, in some instances even by a considerable margin. Also, the exact equilibrium time for a component such as (2) in Figure VIII-2, can not be defined. But, here again, the method yields conservative results. Fairly accurate results could be expected by this method for non- or low-heat producing components.

Limitations on the accuracy of this second method for predicting component temperature-time histories are seen to exist. The method is, nevertheless, reasonably safe for use for a wide variety of components. Its use is recommended whenever characteristic temperature patterns are not known, or the unit operates in a variable environment and the more detailed and laborious computations required for the first method cannot be justified.

Whichever method is employed to predict the temperature-time histories of the components, it is necessary to evaluate the temperature-time history of the case. It is obtained by solving the basic equation (VIII-4), where the representative temperature t_{rp} for a pressurized and sealed unit is the case temperature t_c. Equation (VIII-4) would be integrated by graphical or mechanical procedures, which are discussed in the subsequent section dealing with procedures of evaluation for this type unit. When the environmental conditions remain constant with time, the equilibrium temperatures of the case and components may be predicted by the methods presented in Chapter VI. Then, from the temperature-time history of the case and these equilibrium temperatures, the temperature-time histories of the various components would be predicted by use of one of the two previously recommended methods. The process is considerably more complicated when the environmental conditions vary with time of operation. Integration of equation (VIII-4) requires a greater number of computations and the temperatures of the case and components must be computed at several times of operation, rather than just for the equilibrium time of operation, if the method illustrated in Figure VIII-1 is used to predict the temperature-time histories of the components. Variable environmental conditions complicate the evaluation procedures only in the integration of equation (VIII-4) when the component temperatures are predicted by the more approximate method illustrated in Figure VIII-2. The procedures are illustrated numerically in Example VIII-2, page 269.

4. Procedures for Evaluation of Operational Conditions

The evaluation procedures for determination of the transient thermal state of pressurized and sealed units cooled by free convection and radiation are discussed first for environments remaining fixed with respect to the time of operation, and then for environments varying with time of operation as created by change in the operational conditions of the aircraft or the equipment.

a. Constant Environment and Heat Generation

The heat transfer between the case surface and the environment by free convection and radiation would be calculated for a range of selected case temperatures according to the procedures outlined in Chapter VI, pages 131 to 138. The correction factor F_c needed for these calculations would be assumed as constant and equal to the value established during bench test for operation at thermal equilibrium. The calculations define the heat dissipation rate from the case surface q_{ds} and the rate of any heat received by the surface q_{rc}. Then, by use of the heat balance equation (VIII-1), with q being constant, the rate of heat absorption of the unit q_{ab} for each selected case temperature may be defined. These data would be plotted as a function of the case temperature as illustrated in plot A of Figure VIII-3. The case

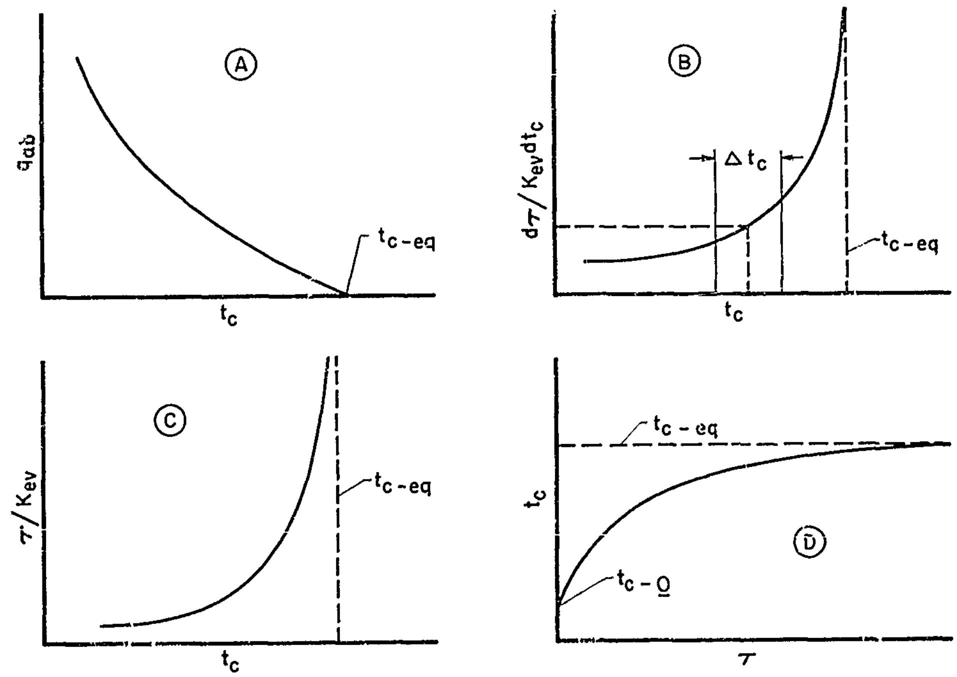


Figure VIII-3. Illustration of Working Plots Used to Predict the Temperature-Time History of the Case of a Pressurized and Sealed Unit Cooled by Free Convection and Radiation. Environmental Conditions Constant temperature resulting in a heat absorption rate q_{ab} equal to zero defines the equilibrium case temperature for the specified environmental conditions. The case temperature equal to the ambient air temperature would be the lower limit for evaluation of q_{ab} since lower case temperatures are not likely to occur. The reciprocal of the curve illustrated in plot A would again be plotted as a function of the case temperature, as illustrated in plot B of Figure VIII-3. Since, according to equation (VIII-4), the reciprocal of the rate of heat absorption q_{ab} is equal to $d\tau/K_{ev}dt_c$, this curve would be integrated mechanically to yield the general temperature-time plot illustrated in plot C. The mechanical integration may be conducted by selecting temperature intervals Δt_c, defining an average value of the ordinate for each interval, and then forming the sum of the products of the average values of the ordinates and their respective temperature intervals. The magnitude of the temperature interval Δt_c would be reduced as the slope of the curve increases, or whenever it is desired to increase the accuracy of evaluation. The actual temperature-time history of the case, illustrated in plot D, is obtained from plot C by introducing the initial case temperature t_{c-0} corresponding to time τ of zero and multiplying the ordinate by the equivalent thermal capacity of the unit K_{ev}. The temperature-time histories of the various components within the unit are then defined by one of the two methods previously outlined. The entire evaluation procedure for a pressurized and sealed unit operating in a fixed environment is given in Example VIII-2.

b. Variable Environment and Variable Heat Generation

The evaluation procedure for units of this type subjected to operating conditions which vary with time is a step-by-step trial-and-error process. It is necessary to employ this procedure to define transient thermal conditions when a unit operating in the steady state is subsequently subjected to environmental change, or changes its electrical function and thereby its rate of heat generation. For example, a unit could reach thermal equilibrium during pre-flight operation and then be subjected to a variable environment as the aircraft pursues its flight plan. In addition, its function may be altered during flight. The procedure would also be applicable to operation analysis in an environment which is appreciably affected by the heat dissipation of the unit. For example, if operation is initiated in a closed aircraft compartment, the compartment temperature would rise, even under constant flight conditions, until the rate of heat dissipation from the compartment can equal, by virtue of increase in the compartment's surface temperature, the heat generation rate of the electronic equipment.

To permit use of the procedure, the variations of ambient air pressure and temperature, surrounding wall temperatures, and rate of heat generation are plotted as functions of time of operation. This establishes the working plots needed for this type transient analysis.

The calculation process is conducted by first selecting a time interval $\Delta\tau$. Next, the temperature rise of the case Δt_c corresponding to the selected time interval would be assumed, thereby defining an average temperature of the case t_{c-av} over the time interval $\Delta\tau$. Average values of the ambient air pressure and temperature and of the radiation-receiving surfaces would be defined from the working plot showing their variation with time. The heat transfer between the environment and the surface of the case is then evaluated according to the procedures outlined in Chapter VI, using the value for the correction factor F_c established during bench test at thermal equilibrium. Equation (VIII-1) would then be used to evaluate the average rate of heat absorption q_{ab-av} for the chosen time interval $\Delta\tau$, after which the corresponding temperature rise of the case Δt_c is defined by equation (VIII-4), modified to the form

$$\Delta t_c = q_{ab-av} \Delta\tau / K_{ev}. \qquad \text{(VIII-7)}$$

When the value of Δt_c calculated by this equation equals the value of Δt_c originally assumed, the case temperature t_c existing at the operating time corresponding to the end of the selected time interval is defined.

When characteristic temperature patterns are employed to define the temperature-time histories of the unit's components, as illustrated in Figure VIII-1, the next step in the evaluation procedure is to calculate the would-be equilibrium temperature of the case t_{c-eq} corresponding to the environmental conditions existing at the end of the selected time interval. This equilibrium temperature would occur when the heat absorbed q_{ab} would be zero, so that the heat generated by the unit q, plus any heat which might be received q_{rc}, would be dissipated by the case surface, q_{ds}. The surface temperature of the case required to dissipate this amount of heat is evaluated by the same procedures previously employed to evaluate the average heat dissipation over the selected time interval. Corresponding would-be equilibrium component temperatures are then defined from the equilibrium case temperature, and lastly, the various component temperatures existing at the end of the time interval are evaluated from the characteristic temperature curves such as those of Figure VIII-1, and these would-be equilibrium temperatures. This step-by-step evaluation procedure is repeated until the final desired time of operation is reached. The procedure is illustrated by part (b) of Example VIII-2. The same procedure to define the temperature-time history of the case would be employed when component temperatures are evaluated by the more approximate method, illustrated in Figure VIII-2. It is apparent that considerable labor would be eliminated by use of the approximate method for determination of component temperatures.

5. Examples

Example VIII-1. Evaluation of Equivalent Thermal Capacity from Bench Test Data of a Pressurized Unit Cooled by Radiation and Free Convection

The following test data are reported for a unit having a cylindrical case, 12 inches in diameter and 14 inches long, 528 square inches cylindrical surface, 226 square inches end surfaces, painted black.

Environment	
temperature, t_o	23°C
pressure, p_o	29.9 inches mercury
wall temperature, t_w	23°C
confinement	large room
Test duration (to near equilibrium)	180 minutes
Heat generation, electrical input minus output (constant)	250 watts

The variation of case temperature with time, determined from the average of readings of 16 thermocouples, is given in Figure VIII-4. It is apparent from the shape of the curve that the test time of 180 minutes was sufficiently long since the rate of case temperature increase at that time was only on the order of 0.02°C per minute. Thus the equilibrium temperature of the case t_{c-eq} = 70°C.

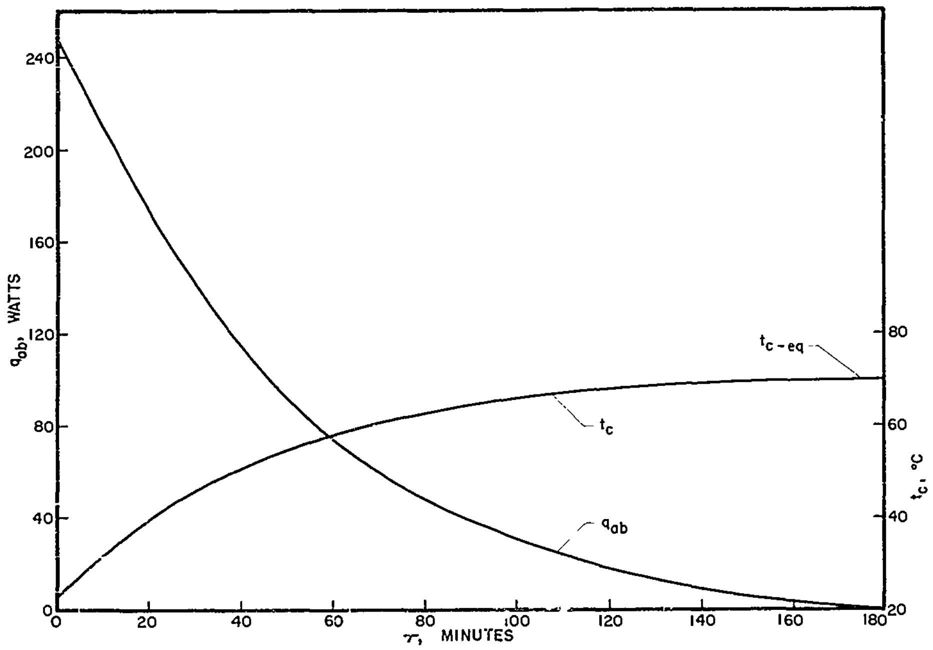


Figure VIII-4. Temperature-Time History for Case of Pressurized Unit, Obtained in Bench Test, and Calculated Variation of Rate of Heat Absorption (Example VIII-1)

For this case temperature and the given environmental conditions, the unit heat dissipation at equilibrium, by free convection, is found from Figure VI-2b, as 0.122 watt per square inch for the cylindrical surface, and 0.161 watt per square inch for the vertical ends. Hence,

$$q_{ds-cv(calculated)} = (0.122 \times 528) + (0.161 \times 226) = 100.8 \text{ watts}$$

The radiant heat transfer is calculated from the value found in Table VI-1 for $\phi_1 = 0.95$ and the value found in Figure VI-3b, based on $t_o = 23°C$ and $t_c = 70°C$, for $\phi_2 = 0.225$ watt per square inch. Hence,

$$q_{ds-rd(calculated)} = 0.95 \times 0.225\ (528 + 226) = 161 \text{ watts,}$$

and the total calculated rate of heat dissipation is 100.8 + 161 = 261.8 watts. Comparing this to the rate of heat dissipation measured electrically defines $F_c = 250/261.8 = 0.955$.

Values of case temperature are read from the faired curve in Figure VIII-4 at time intervals of 20 minutes and for each case temperature the heat dissipation by free convection and radiation is calculated as given above. The actual combined values are defined by multiplication of each calculated value with $F_c = 0.955$. By difference between the known rate of heat generation q = 250 watts, the rate of heat absorption at each temperature is defined from equation (VIII-1). In this manner the data in Table VIII-1 are obtained.

Table VIII-1. Calculated Transient Bench Test Data (Example VIII-1)

τ, minutes	t_c, °C	$q_{ds} = q_c$, watts	q_{ab}, watts
0	23	0	250
20	39.5	72.1	177.9
40	51	135.6	114.4
60	58	176.6	73.4
80	63	201.8	48.2
100	66	218.5	31.5
120	68	231.8	18.2
140	69	242	8
160	69.6	244	6
180	70	250	0

The corresponding curve q_{ab} versus τ is also shown in Figure VIII-4. Based on the above data, the calculated total heat absorption E, using equation (VIII-5) and values for q_{ab-av} based on equal 20-minute intervals, is

$$\begin{aligned} E &= \left[(250 + 177.9)/2 + (177.9 + 114.4)/2 + \text{etc.}\right] \times 20 = \\ &= \left[125 + 177.9 + 114.4 + 73.4 + 48.2 + 31.5 + 18.2 + 8 + 6\right]20 = \\ &= 602.6 \times 20 = 12{,}052 \text{ watt-minutes.} \end{aligned}$$

Then, the equivalent thermal capacity K_{ev} is determined from equation (VIII-6) which is here defined as

$$K_{ev} = E/(t_{c-eq} - t_o) = 12,052/(70 - 23) = 257 \text{ watt-minutes per } ^oC$$

Example VIII-2. Prediction of the Transient Thermal State of a Pressurized Unit Cooled by Free Convection and Radiation

It is desired to determine the transient thermal performance of the pressurized unit described in Example VIII-1 when operated in (1) a fixed environment, and (2) an environment varying with the time of operation. The unit generates heat at the rate of 250 watts and is cooled by free convection and radiation. The case of the unit is painted black and is cylindrical, 12 inches in diameter and 14 inches in length.

a. Fixed Environmental Conditions

The unit which has an initial uniform temperature of 20°C is subjected to an environment having an air pressure of 21 inches mercury absolute and equal air and wall temperatures of 119°C, when operation of the unit is initiated. The surrounding walls have Dural surfaces and the unit is large compared to its enclosure, so that the confinement is close. The environmental conditions are summarized by: t_o = 119°C, p_o = 21 inches mercury absolute, t_w = 119°C, close confinement, Dural surfaces, ϕ_1 = 0.30 (from Table VI-1). Following the procedure recommended on page 263, the heat transfer rate between the case and the environment is determined for a range of case temperatures. The radiant and free convective heat transfer is evaluated by use of Figures VI-2 and VI-3, following the procedure illustrated in Example VI-1. The correction factor F_c for this unit is 0.955, as determined from bench test data and the analysis given in Example VIII-1. Calculated values of the case heat transfer rate are tabulated below. A negative value indicates that heat is received by the case surface. The heat transfer between the case surface and the environment is zero at the case temperature of 119°C, and the equilibrium temperature is seen to be very nearly equal to 180°C since the heat dissipated by the case at this temperature is within 0.5 watt of the heat generation of the unit.

t_c, °C	20	40	60	80	100	119	140	160	180
$q_c = q_{ds} - q_{rc}$, watts	-330	-252	-194	-118	-59	0	68	157	250.5

The difference between the heat generation rate q and the heat transfer rate between the environment and the case defines the heat absorption rate q_{ab} of the unit, according to equation (VIII-1). These data are then plotted as a function of the case temperature t_c, as illustrated in plot A of Figure VIII-3, from which the plot of the reciprocal of the heat absorption rate $1/q_{ab}$ versus the case temperature t_c is obtained. This plot is shown in Figure VIII-5. Integration of the curve in Figure VIII-5 yields the relationship of τ/K_{ev} versus t_c. The integration is performed by dividing the case-temperature scale into 10-degree intervals and noting values of the ordinate at the end of each interval. Thus,

$$\Delta\tau/K_{ev} = (1/q_{ab})_{av}\,(\Delta t_c) = 10\,(1/q_{ab})_{av}$$

or, for $t_c = 30^{\circ}C$,

$$(\tau/K_{ev})_{30} = \Delta\tau/K_{ev} = 10\,(0.00175 + 0.00180)/2 = 0.0177$$

and, for $t_c = 40^{\circ}C$,

$$(\tau/K_{ev})_{40} = (\tau/K_{ev})_{30} + \Delta\tau/K_{ev} - 0.0177 + 10\,(0.00180 + 0.00195)/2$$

$$(\tau/K_{ev})_{40} = 0.0364$$

This process is continued for the entire case temperature range shown in Figure VIII-5 to give the variation of τ/K_{ev} as a function of case temperature. Since K_{ev} equals 257 watt-minutes per $^{\circ}C$, as determined from analysis of the bench test data in Example VIII-1, and the case temperature is $20^{\circ}C$ at zero-time of operation, the actual temperature-time curve for the case may be directly evaluated by finding for known values of (τ/K_{ev}) at known t_c the corresponding values of τ. The curve is shown in Figure VIII-6. The equilibrium case temperature, although not shown on this plot, is $180^{\circ}C$.

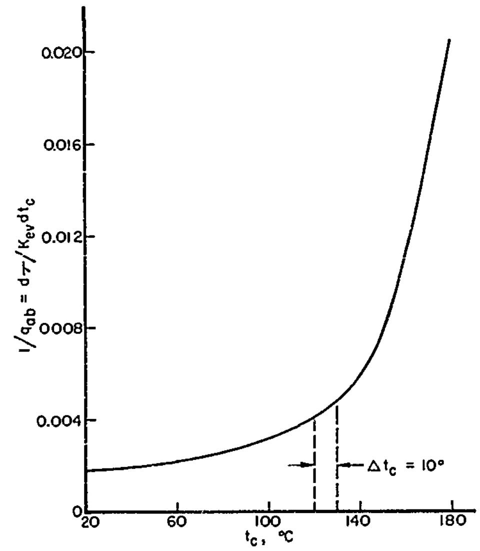


Figure VIII-5. Calculated Plot of Reciprocal of Rate of Heat Absorption (Example VIII-2)

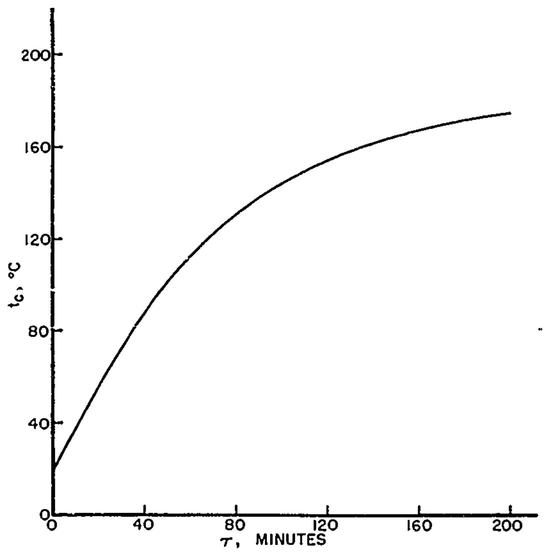


Figure VIII-6. Calculated Temperature-Time History of Case Operating in a Fixed Environment (Example VIII-2)

Characteristic temperature patterns determined from bench test of the unit for three thermally critical components are shown in Figure VIII-7.

The difference between the equilibrium component temperature and the equilibrium case temperature, $(t_{cp-eq} - t_{c-eq})$, as determined from bench test of the unit, is 110°, 40° and 50°C for components (1), (2) and (3), respectively. Hence, for the equilibrium case temperature, as determined above, i.e., $t_{c-eq} = 180°C$, the equilibrium component temperatures are, by addition of the bench test gradients,

$$t_{cp-eq-1} = 290°C \qquad t_{cp-eq-2} = 220°C \qquad t_{cp-eq-3} = 230°C$$

The initial case temperature t_{c-0} and all initial component temperatures t_{cp-0} are 20°C. The actual temperature-time histories of the three components can then be determined from these temperatures and the above equilibrium temperatures. Corresponding values on the abscissa and ordinate of Figure VIII-7 are used to determine values of component temperatures corresponding to the case temperature at known times during the operation being analyzed. For example, to find the temperature of component (2) 60 minutes after operation is initiated, first the case temperature is found from Figure VIII-6 at 60 minutes to be $t_c = 114°C$. Thus all values are known to determine

$$(t_c - t_{c-0})/(t_{c-eq} - t_{c-0}) = (114 - 20)/(180 - 20) = 0.587$$

From Figure VIII-7 is found the corresponding value of $(t_{cp} - t_{cp-0})/(t_{cp-eq} - t_{cp-0})_2 = 0.665$ which is solved for t_{cp-2} by substitution of the known values, giving $t_{cp-2} = 0.665\,(220 - 20) + 20 = 153°C$ at $\tau = 60$ minutes.

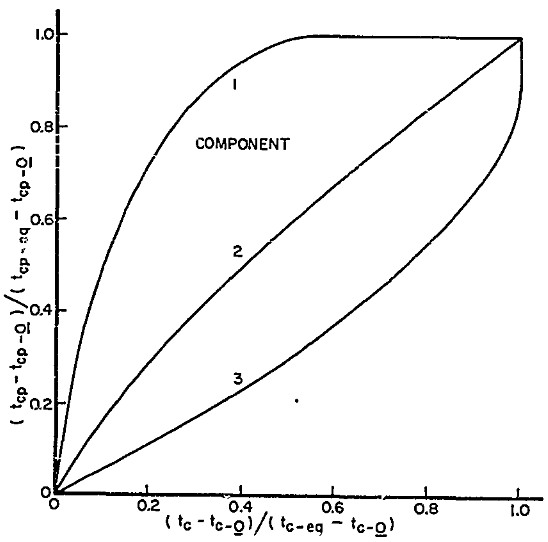


Figure VIII-7. Characteristic Component Temperature Patterns (Example VIII-2)

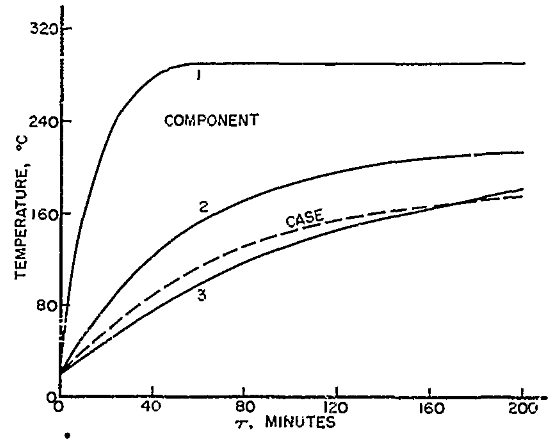


Figure VIII-8. Calculated Temperature-Time History of Critical Components (Example VIII-2)

The temperature-time curves so obtained are shown in Figure VIII-8. They can be used to determine allowable operating time of the unit.

Suppose, for example, satisfactory operation of component (3) is limited to a maximum surface temperature of 150°C. The maximum permissible operating time for the unit is then from Figure VIII-8 equal to 129 minutes, or roughly 2 hours. It should be pointed out that the values of $(t_{cp-eq} - t_{c-eq})$ were obtained from bench test of the unit when the equilibrium case temperature was about 90°C rather than 180°C, as in this example. The values of $(t_{cp-eq} - t_{c-eq})$ for a case temperature of 180°C would undoubtedly be somewhat lower. Thus, the predicted temperature-time histories as given in Figures VIII-8 are somewhat high and, therefore, on the safe side. It is not to be inferred that this unit would be expected to operate satisfactorily at an equilibrium temperature of 180°C. Rather, 180°C represents the equilibrium temperature for the specified environment and the problem has been to define the maximum safe operating time of the unit on the basis of limiting component temperatures, as illustrated above.

b. Variable Environmental Conditions

The unit is installed in a compartment where the ambient temperature and altitude schedule follow those described in Figure VIII-9. The altitude schedule describes an ambient pressure variation corresponding to the N.A.C.A. standard atmosphere. The temperature of the radiation-receiving walls is assumed equal to the ambient air temperature at all times of operation. Initiation of operation of the unit occurs at time zero in Figure VIII-9; the temperatures of the components and case of the unit prior to operation are 30°C. Also, it is assumed that the rate of heat generation of the unit q is constant with time and equal to 250 watts.

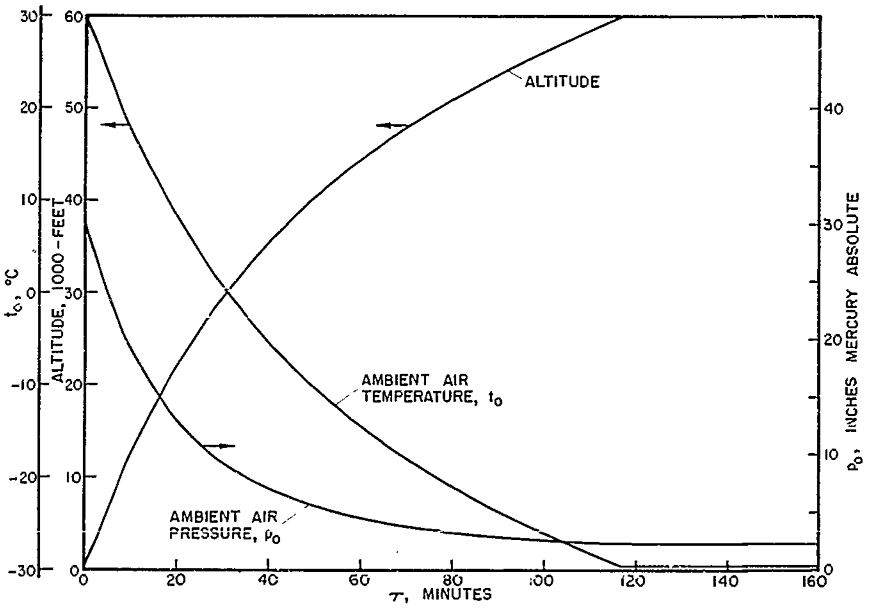


Figure VIII-9. Variable Environmental Conditions (Example VIII-2)

Because of the variable environmental conditions, the temperature-time histories of the case and components must be evaluated using the step-by-step calculation procedure outlined on pages 265 to 267. A time interval of 5 minutes is selected. Hence, the time of operation corresponding to the beginning of the interval is 0 minutes, and to the end of the interval is 5 minutes. Using Figure VIII-9, the ambient air temperature and altitude at the beginning of the interval are 30°C and sea level, respectively, and at the end of the interval 23.7°C and 6600 feet, respectively. Thus, the average air temperature and altitude for the first interval are $(30 + 23.7)/2 = 26.8°C$ and $(0 + 6600)/2 = 3300$ feet, respectively. The air pressure corresponding to the average altitude of 3300 feet is defined by Table A-I-2 and equals 26.5 inches mercury absolute. In summary, for the first time interval of operation

$$t_{o-av} = 26.8°C, \quad t_{w-av} = 26.8°C, \quad p_{o-av} = 26.5 \text{ inches mercury absolute}$$

Next, a temperature change of the case Δt_c is assumed for the selected time interval of 5 minutes. Set $\Delta t_c = 5°C$. Then, the temperature of the case at the end of the interval is $(30 + 5) = 35°C$, or the average case temperature for the interval is $t_{c-av} = 32.5°C$. Heat is dissipated by free convection and radiation from the case at an average temperature of 32.5°C to the air and walls at an average temperature of 26.8°C. The free convection and radiation are evaluated according to the procedures outlined in Example VI-1. The correction factor F_c and the radiation factor Φ_1 are 0.955 and 0.30, respectively (refer to part (a) of this example for source of these data). Accordingly, the heat dissipated by the case to its environment q_{ds} is evaluated to be 12.8 watts. The heat received by the case q_{rc} is zero, so by equation (VIII-1), the average rate of heat absorption during this time interval is $q_{ab} = 250 - 12.8 = 237.2$ watts. Hence by equation (VIII-7), since the equivalent thermal capacity is 257 watt-minutes per °C,

$$\Delta t_c = 237.2 \times 5/257 = 4.6°C$$

whereas the assumed value of Δt_c is 5°C. Therefore, another value of Δt_c must be assumed and the above calculational process must be repeated. A second trial indicates that $\Delta t_c = 4.6°C$ is very close to the correct case temperature rise, with q_{ds} equal to 11.7 watts. In summary, the temperature of the case after 5 minutes of operation is evaluated to be $t_c = 34.6°C$. This step-wise procedure for determination of case temperature must be continued as long as the environmental conditions vary with time, i.e., until = 117.5 minutes. For operation beyond this time, the procedure for constant environment described in part (2) may be used. However, it would usually be found more convenient to continue the step-wise calculations.

The component temperatures after 5 minutes of operation are evaluated from the characteristic temperature patterns given in Figure VIII-7. The values of t_{cp-0} and t_{c-0} used in the parameters of this plot would always be equal to 30°C, but the equilibrium temperatures t_{cp-eq} and t_{c-eq} must be evaluated corresponding to the environmental conditions existing at the end of each time interval. The equilibrium case temperature t_{c-eq} after 5 minutes of operation is determined by the following process. The ambient air temperature and pressure at the end of 5 minutes of operation are 23.7°C and 23.45 inches of mercury (this pressure corresponds to the altitude of 6600 feet). The equilibrium case temperature is defined by that temperature required to dissipate by free convection and radiation, the total rate of heat generation $q = 250$ watts to the environment. It is evaluated by using Figures VI-2b and -3b, the correction factor $F_c = 0.955$ and the radiation factor $\phi_1 = 0.30$. The process is best conducted by assuming various values of the case temperature and then calculating the corresponding values of the heat dissipation. A plot of the heat dissipation versus case temperature defines the case temperature required to dissipate 250 watts. By this process the equilibrium temperature at $\tau = 5$ minutes is 101.2°C, and the abscissa parameter of Figure VIII-7 is equal to

$$(t_c - t_{c-0})/(t_{c-eq} - t_{c-0}) = (34.6 - 30)/(101.2 - 30) = 0.0646$$

Thus, from Figure VIII-7, the ordinate parameter $(t_{cp} - t_{cp-0})/(t_{cp-eq} - t_{cp-0})$ is 0.377, 0.108 and 0.035 for components (1), (2) and (3), respectively. From part (a) of this example, $(t_{cp-eq} - t_{c-eq})$ equals 110°, 40° and 50°C for components (1), (2) and (3), respectively. Hence, t_{cp-eq} is (102.2 + 110) = = 212.2°C, (102.2 + 40) = 142.2°C and (102.2 + 50) = 152.2°C for components (1), (2) and (3), respectively. Substituting these values into the ordinate parameter of Figure VIII-7 yields the component temperatures at the end of 5 minutes of operation. They are:

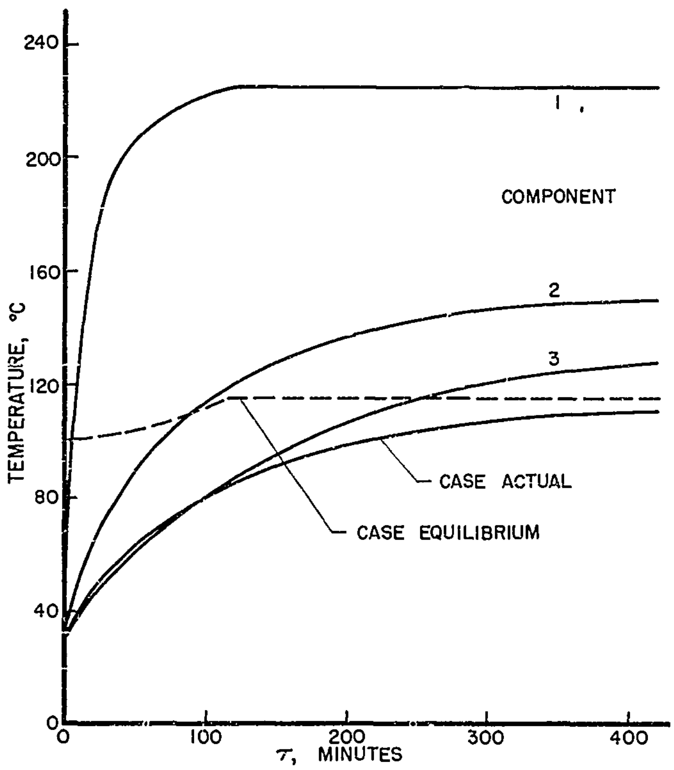


Figure VIII-10. Temperature-Time History of Components and Case for Variable Environmental Conditions (Example VIII-2)

$$t_{cp-1} = 98.7^{\circ}C, \quad t_{cp-2} = 42.1^{\circ}C, \quad t_{cp-3} = 34.3^{\circ}C$$

This procedure for the calculation of instantaneous component temperatures, requiring the calculation of equilibrium case temperatures, corresponding to instantaneous environmental conditions, must be continued as long as the environmental conditions are variable. Thereafter, the same procedure as in part (a) is used since the equilibrium case temperature would remain constant. Resulting case and component temperature-time histories for an extended period of operation are presented in Figure VIII-10. The dashed line illustrates the variation of the instantaneous "would-be" equilibrium temperature of the case, which is required to define the component temperatures.

Pressurized and Sealed Units Cooled by Forced Convection

Prediction of the temperature-time history of a unit of this type, having a case surrounded by a baffle or a case-envelope heat exchanger and cooling air flow created by a blower, can be made with good accuracy on basis of test data and analytical procedures. The methods are more complex than for units cooled by free convection and radiation. This is due to the fact that, while the major means of heat dissipation is forced convection over the case surface, or its extensions forming a heat exchanger core, heat loss or gain may also occur at the external surface of the baffle or heat exchanger. Conditions during the transient thermal state may be evaluated analytically for fixed and variable environments with constant or variable rates of heat generation.

1. Test Procedures

The necessary data for evaluation of non-steady state operating conditions consist of (1) the air flow supplied to the unit by its blower in any environment, (2) the forced convective heat transfer characteristics of the external surfaces, (3) temperature-time histories of the case and all thermally critical components, and (4) the equivalent thermal capacity.

The determination of available flow rates can be performed by analysis, using known characteristics of the equipment-blower-motor combination, determined by test or supplied as specifications. The methods discussed in Chapter V may be used for that purpose. Otherwise, the air rates as function of air pressure and temperature must be determined by tests in an altitude chamber. For these tests the unit must be insulated as well as possible and must reach thermal equilibrium at each operating condition. The range of air pressure and temperature should be sufficiently great to cover the range of environmental conditions expected to be encountered in non-steady state operation. The test procedures are discussed in Chapter IV, page 65 and the calculation of air flow rate from these data in Chapter VII, page 224. It is also indicated in Chapter VII how flow rates for pressures and temperatures slightly different than used in test can be calculated.

To provide data for the determination of heat transfer characteristics of the cooling passage and the external surfaces, bench tests at variable rates of air flow are required. A series of tests with bare external surfaces is sufficient for units with a circumferential baffle. Those with case-envelope heat exchangers require an additional series of tests with well-insulated external surfaces. In all these tests, only measurements taken at equilibrium conditions are important. Not all measurements outlined in Chapter IV, page 52 need be made. Pressure drop data are only necessary if air flow data under other environmental conditions are to be obtained by calculation rather than by altitude chamber test, as indicated above. The principal requirements are for measurement of air flow rate, presumably supplied by an auxiliary apparatus, of surface temperatures on the case and all other surfaces of the heat exchange passage, of the rate of heat generation, and of the air temperatures at inlet and outlet of the cooling passage.

The data required for determination of the temperature-time relationships of the components and the case, and of the equivalent thermal capacity of the unit are obtained in one bench test with bare case, run with the complete unit, including its external cooling blower, but with the addition of an auxiliary air flow apparatus (see Appendix IV). The auxiliary apparatus would serve to measure the air flow and to maintain the air flow constant with time. Thus any natural tendency of the blower to vary the weight flow rate as the temperature level of the unit increases is eliminated. This is necessary for a more reliable determination of equivalent thermal capacity. In this test, operation is initiated from thermal equilibrium, i.e., all parts of the unit are at the same temperature as the environment. Operation is discontinued when all parts have again reached practical thermal equilibrium while operating, i.e., their rates of temperature rise have become imperceptible. During the entire test it is desirable to have a constant environmental temperature. Case, component, external surface, inlet air and outlet air temperatures, air rate, and electrical input and output must be recorded as functions of operating time.

2. Evaluation of Equivalent Thermal Capacity

As discussed on page 258 for the unit cooled by free convection and radiation, the data obtained in bench test must also be reduced for the unit cooled by forced convection to permit calculation of the rate of heat absorption at any time of the transient bench test. Again heat would not be received by the unit externally. Therefore, equation (VIII-1) gives q_{ab} when the rate of heat dissipation q_{ds} can be calculated since the heat generation rate is available from measurement. For this type of unit q_{ds} is the sum of the heat dissipation of the cooling passage(formed by the case and a circumferential baffle or by a heat exchanger case) by forced convection to the cooling air, and of the external surfaces by radiation and convection to the environment. The forced convective heat dissipation rate $q_{ds-\underline{cv}}$ in watts at any time is given by

$$q_{ds-\underline{cv}} = 456\, W\, (t_2 - t_1), \qquad \text{(VIII-8)}$$

where W is the known air flow rate in pounds per second, and t_2 and t_1 the measured temperatures in ^oC at outlet and inlet of the cooling passage, respectively. The part of the external heat transfer rate which consists of the radiant rate q_{c-rd} is calculated from the radiation equation (VI-4) based on the measured instantaneous values of external surface temperature t_e and environmental wall temperature t_w. The convective part of the external heat transfer during test may be attributable to conditions which are essentially those of free convection. However, the discharge configuration of the unit may be such that appreciable air flow is aspirated over the external surface of the unit and the heat transfer conditions would more closely resemble those of forced convection. Different evaluation methods would be applicable to the two conditions. In order to clarify the condition by calculation, rather than by judgement, the external heat dissipation rate by free convection at equilibrium conditions, i.e., at the end of the test, is determined using the charts in Figure VI-2. This value added to the external radiant heat dissipation rate is compared to the measured external heat dissipation under final equilibrium conditions, obtained by heat balance from

$$q_{ds-e-eq} = q_{eq} = 456\ W\ (t_{2-eq} - t_{1-eq}) \qquad \text{(VIII-9)}$$

If the ratio of the measured to the above-calculated external heat dissipation rate is smaller than unity or no more than 10 per cent greater, it may be adapted as a value of the correction factor F_c for the entire test since its magnitude indicates that conditions closely approaching free convection exist. If the value of the ratio is greater than 1.1, conditions more closely resembling forced convection appear to exist. In that event, an external effective convection coefficient H_e in watts per ^oC is found, again on the basis of the equilibrium data at the end of the test. It is defined by the measured value of external heat dissipation at equilibrium, $q_{ds-e-eq}$ from equation (VIII-9), the calculated radiant heat dissipation q_{e-rd} and the temperatures of the external surface t_{e-eq} and of the environment t_o. Thus H_e is defined by

$$H_e = (q_{ds-e-eq} - q_{e-rd})/(t_{e-eq} - t_o). \qquad \text{(VIII-10)}$$

Since the air flow rate W and environmental conditions are constant during the entire evaluation test, it is permissible to assume that the value of H_e would be invariable during the test. Therefore, it provides the necessary means for calculation of the external rate of heat dissipation q_{ds-e} at any time as the sum of its product with the instantaneous temperature difference $(t_e - t_o)$, plus the calculated radiant heat transfer rate q_{e-rd} based on the same temperatures. The effective external heat transfer coefficient H_e is used primarily for the purpose of providing a means for more accurate calculation of the instantaneous rates of heat absorption during bench test. This does not imply its applicability for the subsequent analysis of other operational conditions.

The procedure for evaluation of heat absorption rates during the test is based on plots of t_1, t_2, t_e, W and t_o versus τ. (W and t_o would actually be constant, if the test is also run to determine temperature-time relationships for components and the case, but small variations may occur.)

A suitable number of corresponding instantaneous values are chosen from these plots to calculate instantaneous values of q_{ds-cv} from equation (VIII-8), of q_{e-rd} from equation (VI-4), and of q_{e-cv} using either the charts of Figure VI-2, if it was found that $F_c \lesseqgtr 1.1$, or using the equation

$$q_{e-cv} = H_e (t_e - t_o), \qquad \text{(VIII-11)}$$

if it was found that $F_c > 1.1$. If $F_c \lesseqgtr 1.1$, the instantaneous values of heat absorption rate q_{ab} are calculated from

$$q_{ab} = q - q_{ds-\underline{cv}} - F_c (q_{e-rd} + q_{d-cv}) \qquad \text{(VIII-12a)}$$

If $F_c > 1.1$, as determined from equilibrium conditions,

$$q_{ab} = q - q_{ds-\underline{cv}} - q_{e-rd} - \left[q_{e-cv} = H_e (t_e - t_o)\right] \qquad \text{(VIII-12b)}$$

The total heat absorption E during the test is found by equation (VIII-5) or graphical integration of a plot q_{ab} versus τ. The equivalent thermal capacity K_{ev} is found from equation (VIII-6), where the case temperature t_c is the representative temperature t_{rp}, at the end (t_{c-eq}) and beginning ($t_{c-\underline{O}}$) of the bench test.

The procedure here described is used in Example VIII-3, page 284, for a unit cooled by forced flow.

3. Methods of Evaluating Component Temperature Variation in Non-Steady-State Operation

Since the internal heat transfer mechanism of a pressurized or sealed unit cooled by forced convection is identical to that of a pressurized or sealed unit cooled by free convection and radiation, the methods for predicting temperature-time histories of the components within a unit of this type are identical to those outlined on pages 259 to 263, once the temperature-time history of the case has been evaluated. The method employing characteristic temperature patterns for determination of component temperatures, page 260, is generally recommended. The more approximate method, page 262, would be used only when the characteristic temperature patterns of the components are not known, or the greater computational effort required by the first method cannot be justified.

4. Procedures for Evaluation of Operational Conditions

Evaluation procedures for determination of the transient thermal state of pressurized and sealed units cooled by forced convection require step-by-step analysis whether the operational conditions of the unit are variable or constant with respect to time of operation. The purpose of the analysis would be to define the actual and "would-be" equilibrium temperature-time histories for the case surface of the unit. Evaluation procedures are discussed first for units having a circumferential baffle, and then for those employing a case-envelope heat exchanger. The evaluation procedures permit inclusion of any heat transfer occurring between the external surfaces of the unit and its environment.

a. Units with Circumferential Baffle

The heat transfer characteristics of a unit having a circumferential baffle are defined from data obtained during bench test, where with the external surfaces of the unit bare, equilibrium temperatures of the case, baffle and the air at inlet and outlet of the unit are measured for a range of air rates. The heat dissipated to the cooling air q_{ds-cv} at any air rate would be evaluated from the known values of the inlet and outlet air temperatures by equation (VIII-8). The difference of the heat generated by the unit q and that dissipated to the cooling air q_{ds-cv} defines the external heat transfer q_e between the external surfaces of the unit and its environment. The correction factor F_c would then be evaluated from this value of q_e and the calculated external heat transfer by radiation and free convection. It is then necessary to evaluate the effective heat transfer coefficients H_{c-a} and H_{b-a} for forced convective heat transfer between the case surface and the cooling air and the inner baffle surface and the cooling air, respectively. Procedures for definition of these coefficients as a function of the air rate are identical to those described in detail in Chapter VII, pages 229 to 231. Any heat conducted from the case to the baffle through struts or other supporting members would be evaluated by equation (VII-6) and included in the heat balance according to equation (VII-4). Resulting values of H_{c-a} and H_{b-a} would be plotted as a function of the cooling air rate W to yield a working plot for use with the evaluation procedures subsequently discussed.

When it is required to evaluate the transient thermal state of a unit under variable operating conditions, working plots would be constructed showing the variation of ambient air pressure and temperature, wall temperature, cooling air rate and the unit's heat generation as a function of the time of operation. Ambient air pressure and temperature and the surrounding wall temperature of the compartment in which the unit is installed are affected by the aircraft's flight plan, heat generation within the compartment and the ventilation or refrigeration which may be provided for the compartment. The cooling air rate, as supplied by a blower, is a function of the unit's environmental conditions. Its variation with time of operation would be evaluated using known characteristics of the equipment-blower-motor combination, determined by test or supplied as specifications. The methods discussed in Chapter V may be used for that purpose. Otherwise, the air rate variation with change in ambient air pressure and temperature must be determined from test of the unit in an altitude chamber, as discussed in Chapter VII, page 224, or approximately by the method discussed on page 225.

The evaluation procedure for determination of operational thermal conditions is conducted on a step-by-step basis, and requires trial-and-error solution to determine the average temperatures of the case and baffle for each step in the analysis. The following recommended procedure for analysis would be used for variable or constant operational conditions. At time zero, the unit's case, baffle and component temperatures are defined. The zero-time of operation must be selected as the time at which operation of the unit is initiated, or as the time at which it is initially subjected to variable operational conditions when it has been previously operated in a constant environment for a period of time sufficiently long to permit it to reach a state of thermal equilibrium. For example, a unit under pre-flight operational conditions may attain thermal equilibrium before being subjected to variable operational conditions resulting from the aircraft pursuing its flight plan. The time-zero would then be selected as the time at which the environmental conditions are initially changed. If thermal equilibrium of the unit has not been reached by this time, it would be necessary to refer the time-zero to the time at which operation of the unit was initiated. The analysis to determine the transient thermal state would then be conducted first for the pre-flight operation, during which the environmental conditions remain constant, and then for the flight operational conditions, during which the environmental conditions would normally vary.

Once the zero-time for analysis has been selected and the corresponding thermal state of the unit is defined, a time interval of operation $\Delta\tau$ is selected. Average values of the ambient pressure p_{o-av} and temperature t_{o-av}, wall temperature t_{w-av}, cooling air rate W_{av} and the effective heat transfer coefficients H_{c-a-av} and H_{b-a-av} are then defined from the working plots corresponding to the average time of operation τ_{av} for the selected time interval. The rate at which these variables change with time of operation would affect the choice of the magnitude of the time interval. Next, it is necessary to assume values of the change in case temperature Δt_c and the change in baffle temperature Δt_e. The average case and baffle temperatures for the time interval are defined by

$$t_{c-av} = t_{c-1} + \Delta t_c/2 \qquad \text{(VIII-13)}$$

and

$$t_{e-av} = t_{e-1} + \Delta t_e/2 \qquad \text{(VIII-14)}$$

where t_{c-1} and t_{e-1} represent the case and baffle temperatures, respectively, at the beginning of the selected time interval Δ .

The average temperature rise of the cooling air from inlet to outlet of the unit during the time interval is then evaluated by the equation

$$(t_{2-av} - t_{1-av}) = \left[H_{c-a-av}\,(t_{c-av} - t_{1-av}) + H_{b-a-av}(t_{e-av} - t_{1-av})\right] \Big/ \left[456\, W_{av} + (H_{c-a-av} + H_{b-a-av})/2\right] \qquad \text{(VIII-15)}$$

When the blower forces the air through the heat transfer passage, the inlet air temperature t_{1-av} would differ from the average ambient temperature t_{c-av} by the temperature rise of the air created by the blower. The temperature rise of the cooling air, evaluated by equation (VIII-15), is then used to define the rate of heat dissipation to the cooling air q_{ds} by

$$q_{ds-\underline{cv}-av} = 456\, W_{av}\,(t_{2-av} - t_{1-av}) \qquad \text{(VIII-16)}$$

Next, it is necessary to check the correctness of the assumed value for the baffle temperature. A correct value of the baffle temperature has been selected when the rate of heat transfer through the baffle equals the rate of heat transfer between the external surface of the baffle and its environment. The heat transfer between the baffle surface and its environment is normally by free convection and radiation and would be evaluated from the charts of Figures VI-2 and -3, following the procedure outlined in Example VI-1, using the assumed value of the average temperature of the baffle t_{e-av}, the average values of the ambient air pressure p_{o-av} and temperature t_{o-av}, and the wall temperature t_{w-av}. The correction factor F_c would be taken equal to the value obtained during bench test of the unit. If F_c obtained in bench test is greater than unity, and there is indication that under installation conditions convection may be reduced, a smaller value may be assumed for the non-steady state analysis. The rate of external heat transfer q_e is positive when transferred from the surface to the environment, and negative when received by the surface from the environment. The rate of heat transfer through the baffle is evaluated from the equation

$$q_{e-av} = q_{c-rd-av} + q_{c-cd-av} - q_{b-a-av}, \qquad \text{(VIII-17)}$$

where $q_{c-rd-av}$ represents the average rate of heat transfer by radiation from the case surface to the inner surface of the baffle, evaluated by use of Figure VI-3, and defined by the average temperature of the case t_{c-av} and of the baffle t_{e-av}. The rate of conductive heat transfer $q_{c-cd-av}$ would be evaluated from equation (VII-6), using the average values of the case and baffle temperatures. The average rate of convective heat transfer q_{b-a-av} needed in equation (VIII-17) is defined by

$$q_{b-a-av} = H_{b-a-av} (t_{e-av} - t_{a-av}), \qquad \text{(VIII-18)}$$

where the average temperature of the cooling air t_{a-av} is defined by

$$t_{a-av} = t_{1-av} + (t_{2-av} - t_{1-av})/2 \qquad \text{(VIII-19)}$$

A correct value of baffle temperature is obtained when the rate of external heat transfer q_e evaluated by equation (VIII-17) equals the rate of external heat transfer between the baffle surface and the unit's environment. When the two values of q_e do not agree, it is necessary to assume another value for the temperature rise of the baffle Δt_e during the selected time interval $\Delta\tau$, to calculate the new value for the average temperature of the baffle, and then to repeat the above-indicated process.

After the correct baffle temperature has been determined, it is then necessary to check the accuracy of the assumed temperature rise of the case Δt_c. The average rate of heat transfer from the surface of the case over the selected time interval $\Delta\tau$ is, by heat balance,

$$q_{c-av} = q_{ds-av} + q_{e-av} \qquad \text{(VIII-20)}$$

Hence, the average rate of heat absorption by the unit over the time interval $\Delta\tau$ is

$$q_{ab-av} = q_{av} - q_{c-av} \tag{VIII-21}$$

and the corresponding temperature change of the case is

$$\Delta t_c = q_{ab-av} \Delta\tau / K_{ev}. \tag{VIII-22}$$

All terms in the previous equations are algebraic quantities, therefore the proper sign on any term should be used accordingly. When, for example, heat is received by the unit from its environment, the term q_e is negative, or if heat of the unit's structure is given up, the term q_{ab} will be negative and a drop in the case temperature occurs.

If the temperature change of the case Δt_c evaluated by equation (VIII-22) equals the value originally assumed for the selected time interval, the correct value of the average case temperature for this time interval has been established. Otherwise, it is necessary to repeat the entire evaluation procedure. Since a double trial-and-error procedure is involved in the process, i.e., finding first a correct baffle temperature for any assumed case temperature, and then determining if the case temperature is correct, it is recommended that a check on the correctness of the assumed value of Δt_c be conducted each time the baffle temperature change Δt_e is checked. By this procedure an indication is gained as to the correctness of both assumptions, and, when necessary, new values for Δt_c and Δt_e may be assumed simultaneously for each time through the computational process. This evaluation procedure for a unit operating in a variable environment is illustrated in Example VIII-4. No simple rules can be given to guide the analyst in making the above-mentioned temperature assumptions. By comparison with known conditions, he may make more intelligent estimates, based on differences in air rate, environmental conditions, elapsed time of operation, etc. Here, like in all procedures of this type, experience is the best guide. Relatively small effort is needed to acquire such experience.

Component temperature-time histories are determined from the temperature-time history of the case by procedures identical to those for sealed units cooled by free convection and radiation, as discussed on page

b. Units with Case-Envelope Heat Exchanger

Evaluation procedures needed to determine the transient thermal state for a unit of this type are in principle the same as for units having a circumferential baffle.

The heat transfer characteristics of the exchanger are defined from data obtained from steady-state bench test of the unit with the external surfaces first insulated and then bare. The rate of heat dissipation to the cooling air $q_{ds-\underline{cv}}$ for any air rate would be evaluated from measured values of the inlet and outlet air temperatures by equation (VIII-8). The effective forced convective heat transfer coefficient for the heat exchanger H_{ex} would then be evaluated as

$$H_{ex} = q_{ds-\underline{cv}} / (t_{ex} - t_a), \tag{VIII-23}$$

where t_{ex} and t_a are defined by the measured temperatures of the case t_c, external surface of the exchanger t_e, and the air at inlet and outlet of the unit by the equations

$$t_{ex} = (t_c + t_e)/2 \qquad \text{(VIII-24)}$$

and

$$t_a = (t_1 + t_2)/2 \qquad \text{(VIII-25)}$$

The calculated values of H_{ex} would be plotted as function of the air rate W, giving a generalized working plot for use when evaluating the transient thermal characteristics of the unit. A second working plot would be constructed from the steady-state bench test data to correlate the external heat transfer q_e, which is equal to the difference of the rate of heat generation q and the rate of forced convective heat dissipation q_{ds-cv} evaluated by equation (VIII-8). The external heat transfer q_e would be plotted as a function of the difference of the case and external surface temperatures $(t_c - t_e)$ with the cooling air rate W as a parameter. This is illustrated in Figure VIII-11 where two sets of test points are shown, one set corresponding to the bench tests with the external surfaces insulated, $q_e = 0$, and the other set to the

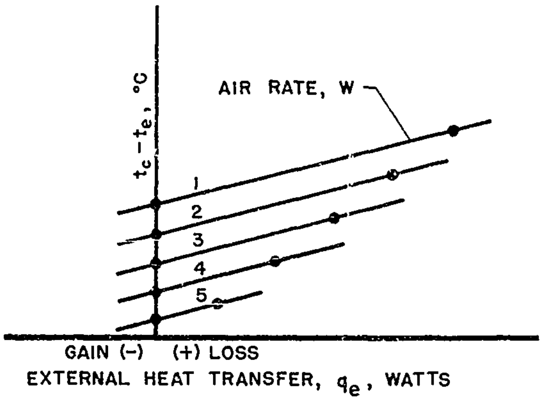


Figure VIII-11. Working Plot for Evaluation of External Heat Transfer

bench tests with the external surfaces bare. Straight lines connecting the test points at corresponding air rates would be drawn on the working plot. By this procedure, a fairly accurate approximation of the actual variation of external heat transfer with the case temperature differential $(t_c - t_e)$ for any value of the air rate W, is obtained. The plot permits evaluation of q_e for any environmental heat transfer condition, providing the external heat transfer coefficient can be defined as a function of ambient pressure, temperature, surface characteristics and air velocity. Extension of the straight lines to the left of the ordinate would permit evaluation of the thermal conditions when heat is gained by the external surfaces of the unit. Extrapolation of the data by this process into the region of high heat gain is not recommended. Extrapolation of the data to higher values of heat loss than existed during bench test with the external surfaces bare, i.e., to values higher than those corresponding to the right-hand set of test points in Figure VIII-11, by extension of the straight lines would normally yield fairly accurate results.

Having established working plots for evaluation of the effective heat transfer coefficient H_{ex} and the external heat transfer rate q_e, the transient thermal state for this type unit may be determined by a step-by-step evaluation procedure, similar to that used in part (a) of this section, for a sealed unit having a circumferential baffle. A time interval of operation $\Delta\tau$ would be selected. Values of the case temperature change Δt_c and the external surface temperature change Δt_e would then be assumed. The average temperatures of the case and external surface, t_{c-av} and t_{e-av}, for the selected time interval of operation would be calculated from their values at time zero of operation and the assumed values of Δt_c and Δt_e. Next, the rate of external heat transfer q_e would be evaluated in two ways. One value would be determined corresponding to the heat transfer between the external surfaces of the unit and its environment by free convection and radiation, using the charts in Figures VI-2 and -3. Average values of the ambient air pressure and temperature and of the surrounding wall temperature for the selected time interval of operation would be used in defining the characteristics of the environment when evaluating this rate of heat transfer. A second value of q_e is defined by the working plot illustrated in Figure VIII-11, using the average case and external surface temperatures and the average air rate W_{av} over the selected time interval. Agreement of the two values of the external heat transfer defines a correct value of t_e for the assumed case temperature change Δt_c.

The next step in the evaluation procedure is to verify the correctness of the assumed case temperature change Δt_c. The average rate of heat dissipation to the cooling air is given by

$$q_{ds-\underline{cv}-av} = (t_{c-av} + t_{e-av} - 2\, t_{1-av}) \Big/ \left[(2/H_{ex-av}) + 1/(456\, W_{av})\right], \quad \text{(VIII-26)}$$

where H_{ex-av} represents the average value of the effective heat transfer coefficient corresponding to the average air rate W_{av} and t_{1-av} defines the average temperature of the cooling air at entrance to the heat transfer passages. The average rate of heat dissipation from the case of the unit q_{c-av} is equal to the algebraic sum of $q_{ds-\underline{cv}-av}$ and q_{e-av} (equation VIII-20), and the average rate of heat absorption $q_{\underline{ab}-av}$ is defined by equation (VIII-21).

Lastly, the temperature change of the case Δt_c is evaluated by equation (VIII-22). Agreement of the calculated and assumed values of Δt_c completes the evaluation process for the selected time interval of operation. The temperature-time history of the case defined by this process would be used to evaluate component temperature-time histories following the procedures outlined on pages

5. Examples

Example VIII-3. Evaluation of Equivalent Thermal Capacity of a Pressurized Unit with Circumferential Baffle, Cooled by Forced Convection

A pressurized unit having a painted cylindrical case 10 inches in diameter and 19 inches long is surrounded by a concentric baffle of 10.5-inch diameter, painted internally and externally, forming with the case a passage for forced air flow. In order to procure the necessary data for the determination of the unit's equivalent thermal capacity, a transient bench test is performed in accordance with the procedure discussed on page 276. During the test, the following constant conditions are determined by measurement:

Environment	
temperature, t_o	20°C
pressure, p_o	29.5 inches mercury absolute
wall temperature, t_w	20°C
confinement	large room (ϕ_1 = 0.95)
Heat generation, electrical input minus output, q	400 watts
Air flow rate, W	0.040 pound per second
Test duration (to near equilibrium)	160 minutes

The variations with time of the case temperature t_c, the baffle temperature t_e, and the air discharge temperature t_2, each determined from the average readings of several thermocouples, are given in Figure VIII-12. It is apparent that after 160 minutes of test time practical equilibrium was obtained. The values of the variables reached at equilibrium are

$$t_c = 61.7^oC, \quad t_{e-eq} = 32^oC, \quad t_{2-eq} = 38.8^oC$$

The actual external heat dissipation at equilibrium is from equation (VIII-9)

$$q_{ds-e-eq} = 400 - 456 \times 0.040\ (38.8 - 20) = 58 \text{ watts}$$

From the values given above, the calculated values of radiant and free convective heat dissipation, according to the methods of Chapter VI, are 30 and 16 watts, respectively. For calculation of radiant heat transfer, the surface of the cylindrical baffle and of one end, equal to 714 square inches, is used. For calculation of convective heat transfer, only the surface of the cylindrical baffle, equal to 627 square inches, is used. Thus, F_c = = 58/(31 + 16) = 1.23 which is greater than 1.1. Therefore, the alternate method is used in which the calculated radiant heat dissipation is assumed correct and an effective external heat transfer coefficient is determined. In accordance with equation (VIII-10),

$$H_e = (58 - 31)/(32 - 20) = 2.25 \text{ watts per } ^oC$$

Then, for any specified time τ, using the temperature test data of Figure VIII-12, the rates of heat dissipation from the case by radiation and convection are calculated by use of equation (VI-4) and (VIII-11), respectively. For example, for τ = 60 minutes, t_e = 30.2°C and t_o = 20°C. Correspondingly, ϕ_2 = 0.0365 watt per square inch. Therefore, q_{e-rd} = 0.95 x 0.0365 x 714 = = 24.8 watts, and q_{e-cv} = 2.25 (30.2 - 20) = 23 watts. At the same time, the air discharge temperature is t_2 = 36.1°C which gives, for the constant air flow rate W = 0.040 pound per second and $t_o - t_1$ = 20°C, according to equation (VIII-8), the heat dissipation rate by forced convection as

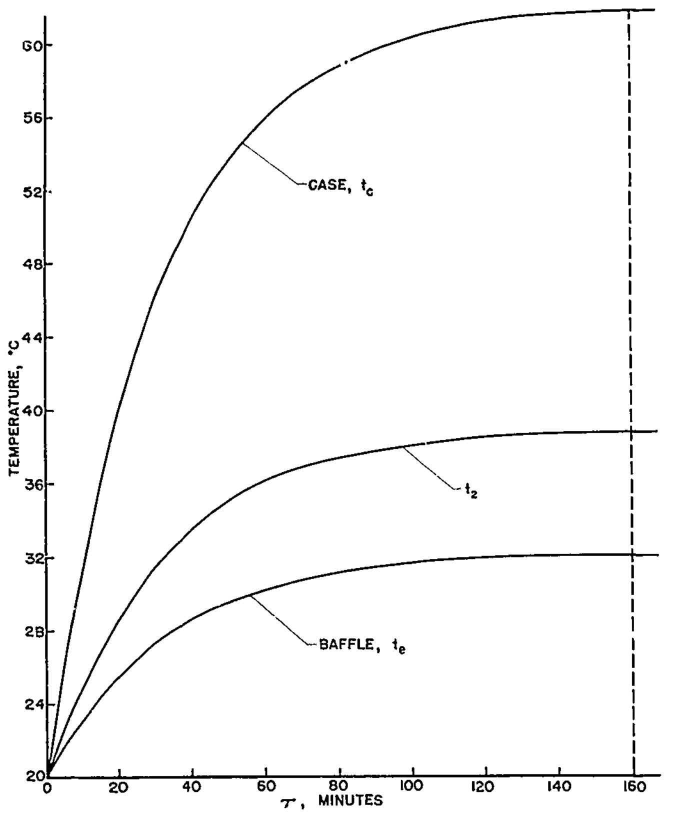


Figure VIII-12. Temperature-Time Data Obtained in Transient Bench Test of Pressurized Unit Cooled by Forced Convection (Example VIII-3)

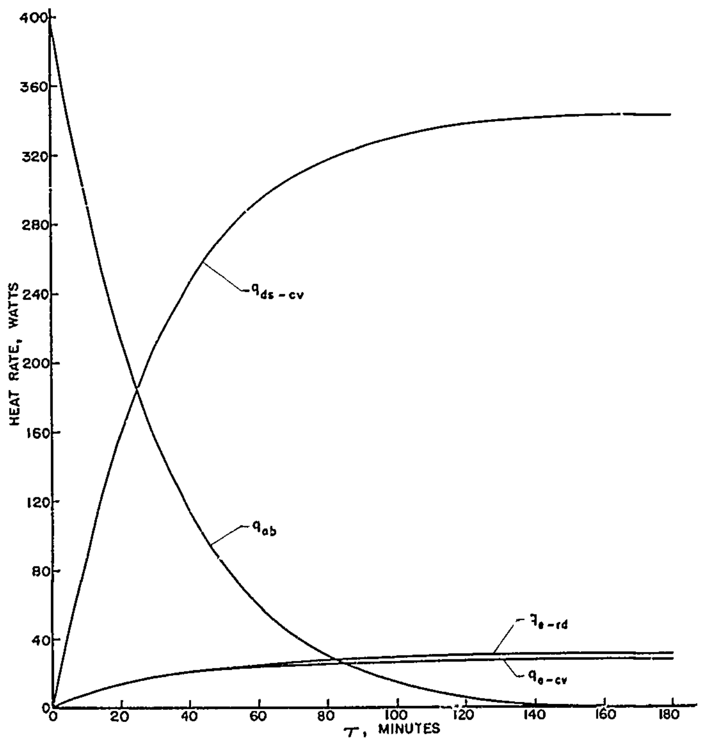


Figure VIII-13. Calculated Time Variations of Heat Dissipation and Absorption Rates in Bench Test of Pressurized Unit Cooled by Forced Convection (Example VIII-3)

$$q_{ds-cv} = 456 \times 0.040\ (36.1 - 20) = 293.5 \text{ watts}$$

Consequently, the rate of heat absorption at $\tau = 60$ minutes is, in accordance with equation (VIII-12b), and for the constant heat generation rate $q = 400$ watts,

$$q_{ab} = 400 - 293.5 - 24.8 - 23 = 58.7 \text{ watts}$$

Performing the above calculations for the conditions at a sufficient number of values for τ provides the data for the curves of q_{e-cv}, q_{e-rd}, q_{ds-cv}, and q_{ab} versus time τ shown in Figure VIII-13. The curve for q_{ab} can be integrated by planimetry or using equation (VIII-5) to obtain a value for the total heat absorbed during the bench test. Using the latter method, the choice of time intervals is best made in accordance with the variation in curvature, using shorter intervals where the curvature is greater. Thus,

$$\begin{aligned} E &= (400/2 + 298 + 216/2)10 + (216/2 + 184 + 157 + 134 + 115 + 98 + \\ &+ 83 + 71 + 59 + 50 + 42/2)5 + (42/2 + 30 + 21 + 14/2)10 + (14/2 + \\ &+ 6 + 2)20 = 606 \times 10 + 1080 \times 5 + 79 \times 10 + 15 \times 20 = \\ &= 12{,}550 \text{ watt-minutes} \end{aligned}$$

The equivalent thermal capacity is calculated from equation (VIII-6) in the form

$$K_{ev} = E/(t_{c-eq} - t_o)$$

The equilibrium case temperature is found from Figure VIII-12 to be 61.7°C. Thus,

$$K_{ev} = 12{,}550/(61.7 - 20) = 302 \text{ watt-minutes per } ^{\circ}C$$

Example VIII-4. Determination of the Transient Thermal State of a Pressurized Unit with Circumferential Baffle, Cooled by Forced Convection

It is desired to determine the thermal conditions of a pressurized electronic unit operating in an aircraft undergoing rapid change in flight speed and altitude. The unit is cooled by air flow induced by a blower over the heat transfer surfaces formed by the case of the unit and a circumferential baffle. It is assumed that the unit has been operated for several hours on the ground so that thermal equilibrium has been attained by the time of take-off. During flight, the compartment air pressure and temperature and the surrounding wall temperature vary with time in the manner indicated in Figure VIII-14. The blower, being uncontrolled, provides a cooling air rate variable with time of flight as shown in Figure VIII-15. The effective heat transfer coefficients H_{c-a} and H_{b-a} vary with the air rate in a manner determined from bench-test of the unit, and are plotted in Figure VIII-15 as a function of the time of flight.

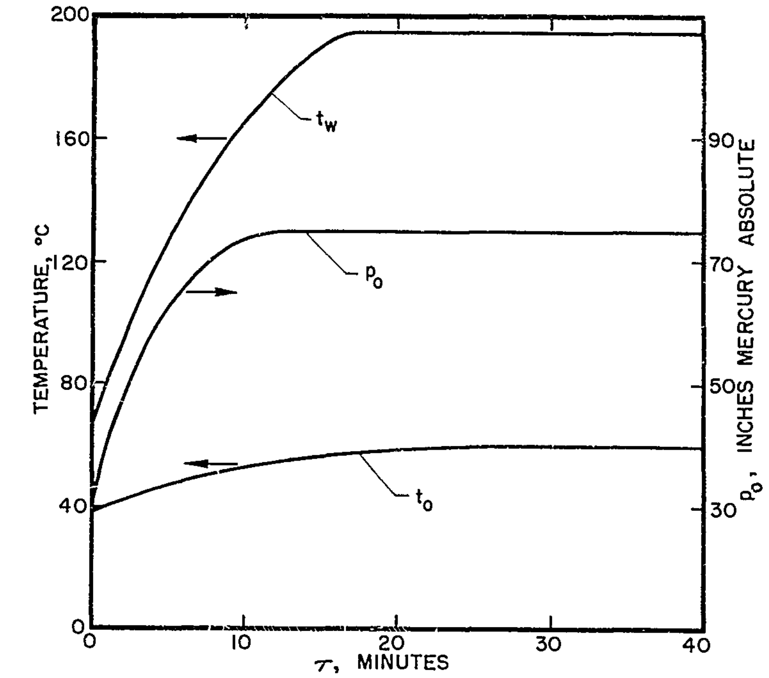


Figure VIII-14. Variation of Environmental Air and Wall Temperatures and Air Pressure with Time of Flight (Example VIII-4)

The construction of the unit is identical to the one described in Example VIII-3. The case is cylindrical, 10 inches in diameter and 19 inches in length, and the circumferential baffle is separated from the case surface by a distance of 1/4 inch. All surfaces of the unit are black painted and the installation conditions are such that the confinement is close. The equivalent thermal capacity of the unit is 302 watt-minutes per ^{o}C, as determined in Example VIII-3, and the rate of heat generation is constant and equal to 400 watts.

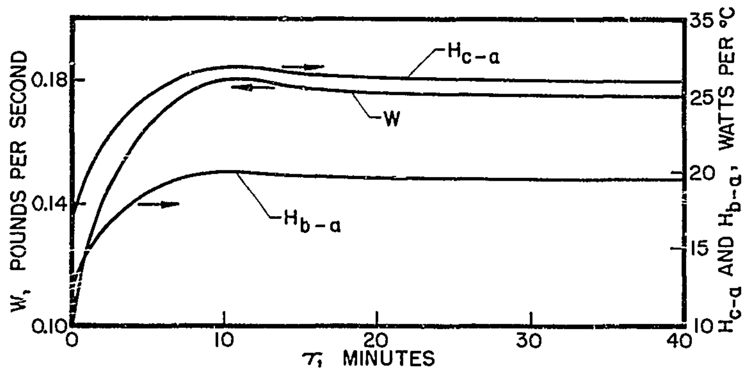


Figure VIII-15. Variation of Cooling Air Rate and Forced Convective Heat Transfer Coefficient with Time of Flight (Example VIII-4)

a. Thermal Conditions of Unit Before Take-Off

The unit is operated in the aircraft prior to take-off for an extended period of time. Hence, at the time of take-off the unit's thermal condition is one of steady-state operation in equilibrium with its environment. The environmental conditions during this period of operation are assumed to remain constant with time and are defined by an ambient air pressure p_o of 30 inches mercury absolute, an ambient temperature of 38°C and a compartment wall temperature of 65°C as shown in Figure VIII-14 for zero-time of operation. The cooling air rate is 0.10 pound per second, as shown in Figure VIII-15.

The equilibrium case and baffle temperatures during ground operation are determined by trial-and-error procedure. Since the wall temperature of 65°C is appreciably above the air temperature of 38°C, a baffle temperature t_{e-eq} is assumed equal to 50°C. The rate of external heat transfer q_e by free convection and radiation is evaluated by use of Figures VI-2 and -3. Free convective heat transfer is assumed to occur only over the baffle surface, which has a surface area of $\pi(10 + 0.5) \times 19 = 627$ square inches. Heat is transferred from the external surface at 50°C to the ambient air at 38°C. From Figure VI-2b, $q_{e-cv-eq} = 16.1$ watts. Radiant heat is transferred from the surrounding walls at 65°C, to the external surface of the unit at an average temperature of 50°C. The external surface receiving radiant heat is taken as the external surface of the baffle and one end of the unit, an area of 714 square inches. From Table VI-1, for close confinement and black-painted surfaces, $\phi_1 = 0.90$. From Figure VI-3b, for surface temperatures of 65° and 50°C, $\phi_2 = -0.082$. Hence, $q_{e-rd-eq} = 0.90 \times 714 \times (-0.082) = -52.7$ watts. The net calculated external heat transfer at equilibrium between the unit and its environment is, then, $q_{e-eq} = 16.1 - 52.7 = -36.6$ watts, indicating heat gain from the environment. The value of q_{e-eq} is taken as calculated, assuming that $F_c = 1$. This is based on a qualitative evaluation of the installation which has indicated that external air flow aspiration effects over the baffle, present in the bench test, would be eliminated.

The unit's rate of heat generation is 400 watts. Thus, the rate of heat dissipation to the cooling air is $q_{ds-cv-eq} = 400 - (-36.6) = 436.6$ watts. The temperature rise of the cooling air at this rate of heat dissipation is, using equation (VIII-8),

$$\Delta t_{a-eq} = q_{ds-cv-eq}/(456\ W) = 436.6/(456 \times 0.1) = 9.57^\circ C$$

The average temperature of the cooling air is, then

$$t_{a-av-eq} = 38 + (9.57/2) = 42.8^\circ C$$

From Figure VIII-15, for the air rate of 0.10 pound per second, the effective heat transfer coefficient H_{b-a} is 12.3 watts per °C. Hence, the forced convective heat transfer from the baffle to the cooling air is

$$q_{b-a-eq} = H_{b-a}(t_e - t_{a-av}) = 12.3\ (50 - 42.8) = 88.6 \text{ watts.}$$

Neglecting any conductive heat transfer from the case to the baffle, the permissible rate of radiant heat transfer from the case to the baffle is, by equation (VII-4)

$$q_{c-rd-eq} = q_{b-a-eq} + q_{e-eq} = 88.6 - 36.6 = 52 \text{ watts}$$

In other words, the case receives heat from the environment at the rate of 36.6 watts and the baffle is capable of dissipating heat to the cooling air at the rate of 88.6 watts, so the rate of radiant heat transfer from the case to the baffle permitted for the assumed baffle temperature of 50°C is 52 watts. If the actual radiant heat transfer equals 52 watts, then a correct value for the baffle temperature has been established.

The actual rate of radiant heat transfer from the case to the baffle is determined in the following manner. The heat dissipated to the cooling air by the case surface is equal to $q_{ds-cv-eq} - q_{b-a-eq}$, or 436.6 - 88.6 = 348 watts. From Figure VIII-15, for an air rate of 0.10 pound per second, the effective heat transfer coefficient H_{c-a} is 16.4 watts per °C. Hence,

$$t_{c-eq} - t_{a-av-eq} = q_{c-a-eq}/H_{c-a} = 348/16.4 = 21.2^{\circ}C$$

or,

$$t_{c-eq} = t_{a-av-eq} + 21.2 = 42.8 + 21.2 = 64^{\circ}C$$

The actual radiant heat transfer from the case to the baffle may now be defined by use of Figure VI-3b, where for a case temperature of 64°C and a baffle temperature of 50°C, $\phi_2 = 0.076$. From Table VI-1, for close confinement and black-painted surfaces, $\phi_1 = 0.90$, and the surface area of the cylindrical portion of the case is $\pi \times 10 \times 19 = 598$ square inches. Hence,

$$q_{c-rd-eq} = 0.90 \times 598 \times 0.076 = 40.9 \text{ watts.}$$

The actual rate of radiant heat transfer from the case to the baffle is 40.9 watts, while heat balance with a baffle temperature of 50°C gave a value of 52 watts. Thus, another value for the baffle temperature would be assumed and the process repeated.

The correct case and baffle temperatures are determined to be 64.5° and 49.6°C, respectively. The heat gain by the unit from the environment is 38.5 watts. These conditions define the steady-state thermal performance of the unit at the time of take-off.

b. Transient Thermal State During Flight

The transient thermal performance for a unit of this type is determined by the procedure outlined in pages 279 to 282. Since thermal equilibrium of the unit exists at the time of take-off for the aircraft, the zero time of operation is selected for this condition. Hence, $t_{c-0} = 64.5^{\circ}C$ and $t_{e-0} = 49.6^{\circ}C$.

The step-by-step analysis is given for the first time interval.

The first time interval $\Delta\tau$ is selected as 1.0 minute; thus, $\tau_1 = 0$, $\tau_2 = 1.0$ and $\tau_{av} = 0.5$ minutes. At this average time of operation, from Figures VIII-14 and -15, $t_{o-av} = 39.0^\circ C$, $p_{o-av} = 35.6$ inches mercury absolute, $t_{w-av} = 73.3^\circ C$, $W_{av} = 0.113$ pound per second, $H_{c-a-av} = 18.0$ watts per $^\circ C$ and $H_{b-a-av} = 13.6$ watts per $^\circ C$. The next step is to assume a value for the case temperature change Δt_c. It is assumed equal to zero for this time interval, since the increase in both cooling air rate and environmental wall temperature would tend to produce a compensating effect on the case temperature. Thus, t_{c-av} for the interval equals the initial temperature of 64.5°C. Next, because of the higher environmental temperatures, the baffle temperature would be expected to increase. Assume $\Delta t_e = 2^\circ C$, $t_{e-av} = 49.6 + 2/2 = 50.6^\circ C$. The temperature rise of the cooling air may now be evaluated by equation (VIII-15).

$$(t_{2-av} - t_{1-av}) = \left[18(64.5 - 39) + 13.6(50.6 - 39)\right] \Big/ \left[456 \times 0.113 + (18 + 13.6)/2\right]$$

$$t_{2-av} - t_{1-av} = 9.15^\circ C$$

By use of equation (VIII-16), the average rate of heat dissipation to the cooling air is

$$q_{ds-\underline{cv}-av} = 456 \times 0.113 \times 915 = 472.5 \text{ watts}$$

The average rate of external heat transfer q_{e-av} is evaluated internally and externally of the baffle to determine the correctness of the assumed value for Δt_e. The average value of q_{e-av} is first evaluated for the free convective and radiant heat transfer between the baffle and its environment. This is defined by $t_{e-av} = 50.6^\circ C$, $t_{o-av} = 39^\circ C$, $t_{w-av} = 73.3^\circ C$ and $p_{o-av} = 35.6$ inches mercury absolute, using Figures VI-2b and -3b. The surface areas and F_c used in this calculation are the same as used in part (a) of this example. By this procedure, q_{e-av} is evaluated as -67 watts, showing that the radiant heat transfer received by the external surface is greater than the free convective heat transfer from the external surface to the ambient air.

A second value of q_{e-av} is determined by equation (VIII-17). The conductive heat transfer between the case and baffle is assumed to be negligible. The average rate of radiant heat transfer from the case to the baffle $q_{c-rd-av}$ is evaluated from the average case and baffle temperatures, 64.5° and 50.6°C, respectively, by use of Figure VI-3b and equation (VI-4), using $\phi_1 = 0.90$. It is determined that $q_{c-rd-av} = 40.9$ watts. The average rate of convective heat transfer from the baffle to the cooling air is defined by equations (VIII-18 and -19).

$$t_{a-av} = 39 + 9.15/2 = 43.6^\circ C$$

$$q_{b-a-av} = 13.6\,(50.6 - 43.6) = 95.2 \text{ watts}$$

Hence, by equation (VIII-17),

$$q_{e-av} = 40.9 - 95.2 = -54.3 \text{ watts}$$

This value of q_{e-av} of -54.3 watts is compared with the previous value of -67 watts, and indicates that the assumed value for the baffle temperature rise Δt_e is not correct. However, since it is reasonably close the correctness of the assumed value for the case temperature change Δt_c should be checked to determine a more accurate value. By equation (VIII-20),

$$q_{c-av} = 472.5 - 54.3 = 418.2 \text{ watts}$$

which is the average rate of heat removal from the case surface. Hence, by equation (VIII-21), the average rate of heat absorption for the selected time interval is

$$q_{ab-av} = 400 - 418.2 = -18.2 \text{ watts}$$

The negative value indicates that heat is given up by the bulk of the unit. The temperature change of the case is, then, by equation (VIII-22) and for $K_{ev} = 302$,

$$\Delta t_c = -18.2 \times 1/302 = -0.06^\circ C$$

The case temperature decreases by a small amount over the time interval of one minute.

The values of Δt_c and Δt_e originally assumed are revised and the process is then repeated. Correct values of Δt_c and Δt_e are determined to be -0.05° and $3.0^\circ C$, respectively. Thus, after one minute of operation the case temperature is $64.5 - 0.05 = 64.45^\circ C$, and the baffle temperature is $49.6 + 3.0 = 52.6^\circ C$. The temperature-time histories of the case and baffle for the specified flight plan determined by repeating this process for various selected time intervals are shown in Figure VIII-16. Temperature-time histories of the components would be defined by one of the procedures outlined on pages 259 to 263.

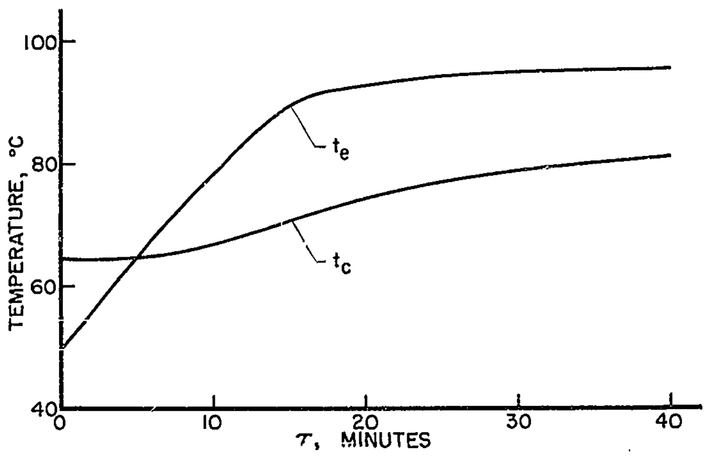


Figure VIII-16. Variation of Case and Baffle Temperatures with Time of Flight (Example VIII-4)

Pressurized and Sealed Units with Integrated or Separate Heat Exchanger

Methods of predicting the temperature-time history of a unit of this type are in many respects similar to those applicable to other units cooled by forced air flow, discussed in the preceding section. Results of comparable accuracy may be obtained by the use of similar test data and analytical procedures. The transient thermal state of such a unit may be evaluated analytically for fixed and variable environments with constant and variable rate of heat generation.

1. Test Procedures

The necessary data for evaluation of non-steady-state operating conditions are the same as those for other pressurized units cooled by forced air flow, as indicated on page 275. The test procedures and required test runs are also the same as those described on pages 275 to 276. The only additional measurements required in all tests are those of the internal air temperature t_{it} entering and leaving the heat exchanger core. Also, if the unit does not have its own external cooling air blower, all tests would be performed using an auxiliary air flow apparatus for the supply and measurement of cooling air rates. If the unit is designed for use with a separate heat exchanger, the characteristics of the ducts would have to be included in the evaluation. Therefore, temperature measurements of the internal circulating air must be made not only at the inlet and outlet of the heat exchanger, but also at the internal inlet and outlet of the unit proper. If the separate heat exchanger is a central unit serving other electronic units of similar design, the whole system should be tested simultaneously. In that instance, it would be desirable to include all the ducting in the laboratory test set-up. However, it would not be practical to determine by altitude chamber tests of the entire system the air flow rates furnished by a cooling blower attached to a central heat exchanger under various environmental conditions. Instead, the heat exchanger alone with the blower can be tested by simulating the heat load supplied by the units to be cooled by means of a heater in the internal air circulation circuit. If the central heat exchanger is designed for ram air cooling, flow rates would be defined by the flight characteristics of the aircraft and the resistance characteristics of the heat exchanger and the air induction system.

2. Evaluation of Equivalent Thermal Capacity

The procedure resembles that required for the other units cooled by forced air flow, as described on pages 276 to 278. For this type of unit, the total rate of heat dissipation at any time is the sum of the heat dissipation in the heat exchanger by forced air flow, and that occurring at the surfaces of the case and of the connecting air ducts, if such are used. In bench test, the external heat dissipation from the case and ducts is likely to occur only by radiation and free convection since air-aspirating effects frequently encountered in other units can be avoided by normal test precautions which would remove air discharge ports from the vicinity of the case surfaces.

The forced convective heat dissipation rate is defined by equation (VIII-8), referred to the external cooling air, in the form

$$q_{ds-cv} = q_{ex} = 456\, W_e\, (t_{e-2} - t_{e-1}) \qquad \text{(VIII-27)}$$

The radiant and free convective heat transfer rates from the case surface are determined by the case temperature t_c and environmental conditions. Like described on page 277, for the other units cooled principally by forced convection, the accuracy of case heat loss calculations are checked by heat balance at equilibrium conditions attained at the end of the test run. Equation (VIII-9) is used in the form

$$q_{ds-c-eq} = q_{eq} - 456\, W_e\, (t_{e-2-eq} - t_{e-1-eq}) \qquad \text{(VIII-28)}$$

and the value of F_c is determined, as described on page 277. It is unlikely that the alternate method of evaluating the convective heat transfer coefficient, as described on page 277, would be required since test precautions should be taken to provide an environment for the unit approaching free convective conditions very closely. However, if it is found that $F_c > 1.1$ the alternate procedure must be used to provide the necessary accuracy for the calculation of instantaneous rates of heat absorption during the bench test. Equation (VIII-10) would then have the form

$$H_c = \left[q_{ds-c-eq} - q_{c-rd}\right] / (t_{c-eq} - t_o) \qquad \text{(VIII-29)}$$

The procedure for evaluation of heat absorption rates during the test is based on plots of t_{e-1}, t_{e-2}, t_c, W_e and t_o versus τ, the latter two variables usually being constant. The methods used are the same as those described on page 278. It is not likely that equation (VIII-12a) would be applicable in the form

$$q_{ab} = q - q_{ex} - F_c\, (q_{c-rd} + q_{c-cv}) \qquad \text{(VIII-30a)}$$

Should it be necessary to use an H_c-factor in evaluating the bench test data, equation (VIII-12b) would be used, transformed in similar manner to

$$q_{ab} = q - q_{ex} - q_{c-rd} - H_c\, (t_c - t_o) \qquad \text{(VIII-30b)}$$

The method as described on page 278 is used for determination of the total heat absorption E during the test. The equivalent thermal capacity K_{ev} is found from equation (VIII-6). For this type of unit, the thermal state is best expressed in terms of the average temperature of the internal circulating air which, as noted on page 293, should be available from measurements of the internal inlet and outlet air temperature. Therefore, equation (VIII-6) as applied to this type unit would have the form

$$K_{ev} = E / (t_{it-av-eq} - t_{it-av-\underline{0}}) \qquad \text{(VIII-31)}$$

The initial average internal air temperature is most likely to be equal to the environmental temperature $t_{o-\underline{0}}$.

3. Evaluation of Component Temperatures During Non-Steady State Operation

Component temperatures during non-steady state operation for a unit of this type would be predicted from characteristic temperature patterns, such as discussed on pages 259 to 261, and illustrated in Figure VIII-1, using the average internal air temperature t_{it-av} as a reference rather than the case temperature. The temperature-time histories of the various components and of the average internal air temperature determined from bench test of the unit would be used to plot the parameters

$$(t_{cp} - t_{cp-\underline{O}})/(t_{cp-eq} - t_{cp-\underline{O}}) \text{ versus } (t_{it-av} - t_{it-av-\underline{O}})/(t_{it-av-eq} - t_{it-av-\underline{O}})$$

for each component under consideration. The average internal air temperature t_{it-av} is evaluated from bench-test data as the average of the internal air temperatures at inlet and exit of the heat exchanger.

The use of the characteristic temperature patterns to define component temperatures during non-steady state operation requires knowledge of the variation of average internal air temperature with time of operation. Procedures for evaluation of the temperature-time history of the internal air are presented in the subsequent section. The more approximate method for predicting temperature-time histories of the components, discussed on pages 262 to 263, may also be applied to a unit of this type by using the constant temperature differential as $(t_{cp-eq} - t_{it-av-eq})$ rather than $(t_{cp-eq} - t_{c-eq})$. It is recommended for use only when the characteristic temperature patterns of the components have not been established, or when only an approximate temperature-time history is desired and the more laborious and time-consuming procedures required for the characteristic temperature patterns cannot be justified. This procedure requires, nevertheless, knowledge of the variation of the average internal air temperature with the time of operation.

4. Evaluation Procedures for Determining Non-Steady State Thermal Conditions

Non-steady state thermal conditions for a unit of this type are evaluated by a step-by-step computational procedure and from generalized bench-test data defining the performance of the heat exchanger, case heat transfer, the characteristic temperature patterns of the components, and the equivalent thermal capacity of the equipment. A trial-and-error computational procedure is required to define the variation of the average internal air temperature with time of operation, after which the component temperature histories may be evaluated directly.

Steady-state bench test data are used to define two working plots for use in non-steady state analysis. The case heat transfer rate q_c, evaluated by heat balance from the bench-test data, would be plotted as a function of the temperature difference between the average internal air and the case surface, as illustrated in Figure VII-7. The effective heat transfer coefficient of the heat exchanger q_{ex}, defined as

$$H_{ex} = q_{ex}/(t_{it-1} - t_{e-1}) \qquad \text{(VIII-32)}$$

and derived from bench-test data, would be plotted as a function of the external cooling air rate W_e. The temperatures t_{it-1} and t_{e-1} used in the definition of H_{ex} are the temperatures of the internal and external air at entrance to the heat exchanger, respectively. The heat dissipated to the cooling air in the heat exchanger q_{ex} would be evaluated from the bench-test data by equation (VIII-27).

For evaluation of the transient thermal performance, a zero-time of operation is selected, which would be when operation of the unit is initiated and the entire equipment is at a uniform initial temperature $t_{\underline{0}}$, or when the unit operating in thermal equilibrium is suddenly subjected to a variable environment, such as the conditions used in Example VIII-4. A time interval of operation $\Delta\tau$ is then selected and the average total time at which operation for the interval would be evaluated, $\tau_{av} = \tau_1 + \Delta\tau/2$. The environmental conditions and the external cooling air corresponding to this average time instant of operation would then be defined, i.e., the average ambient pressure p_{o-av}, average ambient temperature t_{o-av}, average wall temperature t_{w-av} and average air rate W_{e-av}. Next, it is necessary to assume values of the average internal air temperature change Δt_{it} and the case temperature change Δt_c for the selected time interval $\Delta\tau$. The average internal air temperature and case temperature would then be evaluated as

$$t_{it-av} = t_{it-av-\underline{0}} + \Delta t_{it}/2 \qquad \text{(VIII-33)}$$

and

$$t_{c-av} = t_{c-\underline{0}} + \Delta t_c/2 \qquad \text{(VIII-34)}$$

These values permit definition of the average rate of case heat transfer q_{c-av} from the working plot derived from bench-test data giving case heat transfer as a function of the differential between the average internal air temperature and the case temperature.

A second value for the average rate of case heat transfer q_{c-av} would then be calculated from the average case temperature t_{c-av} and the average environmental air pressure and temperature, p_{o-av} and t_{o-av}, and the average wall temperature t_{w-av}, using the free convection and radiation charts in Figures VI-2 and -3. Agreement of this value of case heat transfer with the first value derived from the internal case temperature differential defines the correct average case temperature for the selected time interval of operation. When they do not agree, the value of Δt_c originally assumed would be revised and the process would be repeated until approximate agreement is reached.

The correctness of the assumed value for the change in average internal air temperature Δt_{it} would be determined by evaluating the average rate of heat dissipation in the heat exchanger q_{ex-av} to the cooling air from the equation

$$q_{ex-av} = (t_{it-av} - t_{e-1-av}) \Big/ \left[1/H_{ex-av} - 1/(912\, W_{it})\right], \qquad \text{(VIII-35)}$$

where t_{a-1-av} defines the average temperature of the external cooling air at entrance to the heat exchanger, H_{ex-av} the effective heat transfer coefficient corresponding to the average external cooling air rate W_{e-av}, determined from the working plot derived from bench-test data, and W_{it} the internal air rate which remains constant for all operation conditions and is defined by heat balance from the bench-test data. The average rate of heat absorption by the equipment q_{ab-av} is then defined by heat balance,

$$q_{ab-av} = q_{av} - q_{ex-av} - q_{c-av} \qquad \text{(VIII-36)}$$

from which the temperature change of the internal air is defined by

$$\Delta t_{it} = q_{ab-av} \; \Delta\tau/K_{ev} \qquad \text{(VIII-37)}$$

Agreement between this value of Δt_{it} and the value originally assumed defines the correct temperature change of the internal air, providing the assumed value of the case temperature change agrees with its calculated value.

The trial-and-error evaluation procedure is repeated for each selected time interval of operation, until the temperature-time history of the internal air is defined. Component temperature-time histories are evaluated from the internal air temperature-time history by the procedure indicated in the previous section. The evaluation procedure herein presented is applicable to units having a separate heat exchanger, providing heat transfer between interconnecting ducts and their environment is included in the analysis as a part of the case heat transfer.

Vented Units with Closed Case Cooled by Radiation and Free Convection

Prediction of the temperature-time history of a unit of this type, usually having a relatively low heat generation rate per unit volume, can be made with reasonable accuracy, providing suitable test data are available to define the internal and external heat transfer mechanisms of the unit. The analytical procedures are not complex, but must rely for good accuracy on a prohibitive quantity of test data. However, relatively limited test data provide the basis for estimates of temperature-time histories having a usually acceptable degree of reliability. In general, all types of possible non-steady-state operating conditions can be analyzed.

1. Test Procedures

The necessary data for evaluation of non-steady-state operation of this type of unit consist principally of (1) temperature-time histories of all critical components and the case, (2) heat transfer characteristics of the case, and (3) the equivalent thermal capacity. This unit has the salient feature that all heat dissipated or received must pass through the case surface. However, the temperature relationships between the case and the individual components are not independent from the environment, as for sealed units, but vary with the ambient air pressure which also exists within the unit. Since numerical knowledge on the distribution of component heat dissipation to the case by the three modes of heat transfer, under varying environmental conditions, is practically impossible to obtain, altitude chamber tests must be relied upon extensively to obtain the needed test data.

The relative temperature-time histories of the components and of the case are affected by air pressure and, therefore, at least two, but preferably three or more, transient tests must be run from initiation of operation to thermal equilibrium. One test would be run on the bench, being careful to maintain a free convection environment by suitable screening. The other test, or preferably tests, would be run in an altitude chamber. All tests must be performed without case insulation. Altitude chamber tests should be run at least at the minimum ambient pressure anticipated as operating environment and preferably at two temperature levels defining the extremes of the probable range. However, one chamber temperature, similar to the ambient of the bench test, is sufficient if less accuracy in the subsequent analysis is desired. It is also desirable to run the same tests at one or two intermediate pressure levels, with two chamber temperatures at each to provide complete data. However, these runs may be omitted if best accuracy over the entire probable range of environmental operating conditions is not required.

Since all runs are also needed to provide data on the heat transfer characteristics of the case, complete surface temperature explorations, as discussed in Chapter IV, page 67, are required. In particular, it is necessary to affix a sufficient number of thermocouples to the case to permit the assumption that the average of the measurements would be representative for the entire case. Besides component temperatures, environmental air and surface temperatures must be measured during the entire test. The rate of heat generation must be ascertained by measurement of electrical input and output during the entire test and should, if possible, be maintained constant. It is also desirable to maintain the environmental conditions constant during each test.

2. Evaluation of Equivalent Thermal Capacity

The equivalent thermal capacity of this type unit is not a unique value, but is a function of the environmental air pressure. Since all heat dissipated and received by the unit must pass through the case surface, it is logical to select the average case temperature as the representative temperature on which the equivalent thermal capacity is based. However, the heat transfer mechanism between the components and the case is affected by air pressure. For example, at lower pressure the temperature difference between components and case is increased and, therefore, more heat is absorbed in the unit so that the equivalent thermal capacity would be greater.

The extent to which the equivalent thermal capacity increases with reduced pressure depends on how important the contribution of free convection is to heat transfer from components to the case at ground level. Thus, it may occur that the equivalent thermal capacity may increase 30 per cent, or less, between ground level and 50,000 feet altitude. In that event, it is sufficiently accurate to assume a linear variation of the equivalent thermal

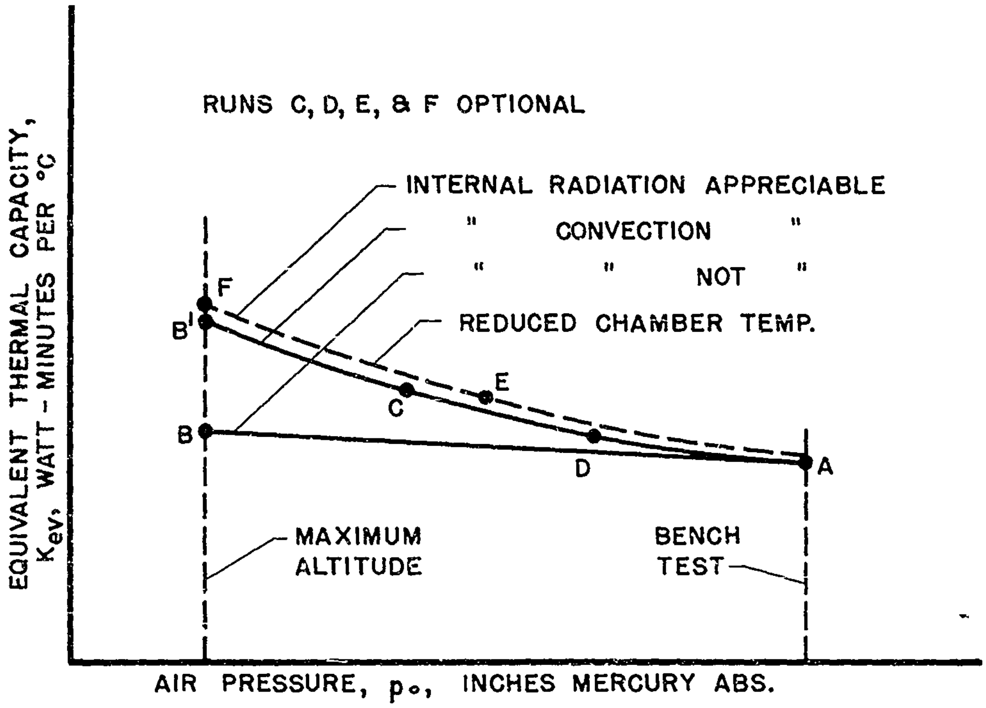


Figure VIII-17. Interpretation of Variable Equivalent Thermal Capacity and Required Test Points for Closed Vented Unit

capacity with ambient pressure from ground level to the corresponding maximum altitude, as shown in Figure VIII-17 by line A-B. However, should the change be greater, it is desirable to determine from test data one or two intermediate values to ascertain the curvature of the line which should bend upward at lower pressure, as shown in Figure VIII-17 by line A-B'. It is also possible that if an appreciable tendency for radiant heat transfer between the components and the case exists, the equivalent thermal capacity would be reduced at constant pressure but increased ambient temperature. Altitude chamber test data at two pressures, and with two temperatures at each would indicate this tendency and the curve could be extrapolated to ground level pressure, as shown in Figure VIII-17 by line F-E. These additional tests and their data should only be used when best accuracy is desired in the analyses to be performed subsequently.

The calculation procedure for determination of the equivalent thermal capacity corresponding to any test condition is identical with those for the pressurized units. Using the bench test data obtained by careful screening of the test area from extraneous drafts and radiation effects, the same procedure as described on page 258, applied to pressurized and sealed units cooled by free convection and radiation is used. However, for altitude chamber test data this method is not suitable for determination of external convection at the case surface because forced convective conditions usually exist in a chamber due to air circulation with a blower. Therefore, the alternate method for determining external convective heat transfer rates described on page 277 for pressurized and sealed units cooled by forced convection should be used. The form of equation (VIII-10) for determination of the effective case convection coefficient H_c in watts per °C is altered since the case heat dissipation rate at equilibrium is the total heat generation rate. Therefore, the equation has the form

$$H_c = (q - q_{c-rd})/(t_{c-eq} - t_o) \tag{VIII-38}$$

The remainder of the procedure is the same as that for the bench-test data, except that instead of expressing, for any ime during the test, the case heat dissipation rate as

$$q_{ds-c} = F_c\,(q_{c-rd} + q_{c-cv}), \tag{VIII-39}$$

it is determined by

$$q_{ds-c} = q_{c-rd} + H_c\,(t_c - t_o) \tag{VIII-40}$$

The values of q_{c-rd} and q_{c-cv} are those calculated with the aid of the charts in Figures VI-2 and -3, respectively. The heat absorption rate is at any time the difference between the rates of heat generation and case dissipation. Finally, as pointed out previously, the case temperature is the representative temperature to be used in equation (VIII-6) to determine the thermal capacity K_{ev} on basis of the total heat absorption calculated for each test condition, in the same manner as outlined on page 258 for the pressurized unit.

3. Methods of Evaluating Component-Temperature Variation in Non-Steady State Operation

The two methods of determining the temperature-time history of components on the basis of the average case temperature variation, as discussed on pages 259 to 263 for the pressurized unit, also apply to the closed vented unit. The only differences in the procedures are that for some components either transient temperature curves for several pressure levels must be available to use the first method, or equilibrium component-to-case temperature differences at several pressure levels must be known to use the second method. This is necessitated by the fact that the ambient pressure, being also that within the unit, would affect the temperature difference between component and case surfaces for all components cooled to an appreciable extent by free convection.

Some qualitative effects of reduced pressure conditions on the equilibrium temperature rise above the case for various types of components are shown in Figure VIII-18. Components (1) having relatively poor thermal contact with the chassis and through the chassis with the case, such as large tubes mounted in ceramic sockets, would be affected most by the pressure level, particularly if they are mounted with liberal spacings. Also, non-heat producing components (2), such as capacitors, may be affected appreciably if they are so installed that they are subjected to much heat gain by chassis or lead

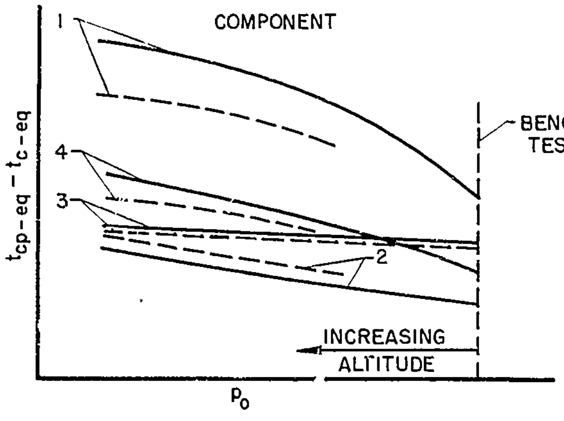


Figure VIII-18. Effect of Pressure Level on Component Equilibrium Temperature Rise

conduction and radiation from surrounding heat producing components and depend, therefore, on convection to dissipate the heat gained. In contrast, components (3), such as transformers which do generate heat but are in good thermal contact with the chassis so that they can rely principally on conductive cooling, will maintain an almost constant temperature difference with the case, regardless of pressure level. Only if such a component (4) is installed adjacent to a high-temperature component which tends to raise the chassis temperature appreciably would convective conditions affect its temperature level noticeably. In that case, with good convective conditions, like at ground level, the component may operate at a surface temperature lower than the surrounding chassis. With poor convective conditions it may operate at a surface temperature higher than the chassis and dissipate its own heat generation by conduction and radiation without receiving heat from adjacent components. In respect to the case, the component temperature rise would be appreciably greater under the poorer convective conditions of increased altitude.

For the prediction of component temperature-time histories by the second method, discussed on pages 262 to 264, which is for this type of equipment relatively approximate, it is necessary to obtain curves such as those shown in Figure VIII-18 by means of at least three points which implies one bench test and two altitude chamber tests at different pressures. It is also desirable to check the effect of temperature level by running at each of the reduced pressures, one test at higher ambient temperature which may produce effects as shown by the dashed lines of Figure VIII-18.

The temperature-time histories of the types of components whose temperature rise relative to the case is affected by the environmental pressure level would also be subject to change with pressure. In Figure VIII-19 characteristic temperature patterns are shown for several types of components. The solid lines would correspond to patterns obtained in a bench test, the dashed lines to those in an altitude chamber test at low pressure. Component (1) represents high-temperature heat generating components for which under all operating conditions radiation and conduction would predominate. Such

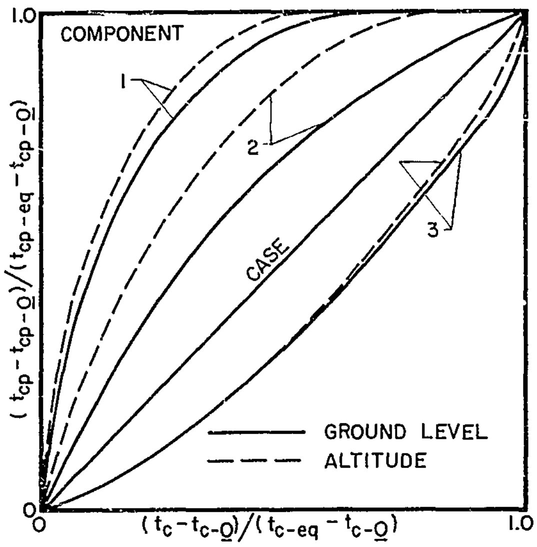


Figure VIII-19. Effect of Pressure Level on Typical Temperature Patterns of Components in a Closed Vented Unit

components would be subjected to slightly faster temperature rise at altitude and would reach near-equilibrium temperature earlier. Component (2) is representative of those with somewhat greater thermal capacity per unit heat generation and surface area, depending to considerable extent on convective heat transfer. At altitude, a tendency would exist for more rapid initial temperature rise. In the later phases of operation the temperature level would be high enough so that radiation and conduction would be sufficiently increased to substitute for reduced convection. Components with high thermal capacity, such as component (3) would be affected very slightly by altitude since they usually also have good conductive heat transfer to the case. However, in the later operating phase it is likely that the temperature rise would occur at a faster rate.

For the prediction of component temperature-time histories by the first method, discussed on pages 262 to 263, it is desirable to obtain the plots shown in Figure VIII-19 first by one bench test and one altitude chamber test to ascertain the differences of the curves due to pressure. Also, a second test at the same altitude pressure but higher temperature level would disclose any differences caused by temperature. Depending on the magnitude of the differences for the critical components, the need for additional reduced-pressure data would become apparent. If, over the extreme pressure range expected, the differences are not great, interpolation may be resorted to in the prediction of component temperatures at intermediate altitudes.

In all details, the methods of evaluating instantaneous component temperatures using either one of the two methods are the same as those discussed on pages 258 to 262. The evaluation must be based on the temperature-time history of the case obtained by solving equation (VIII-4). It must be kept in mind that the equivalent thermal capacity K_{ev} for this type unit varies with the environmental air pressure and, therefore, even for constant environments, the solution of equation (VIII-4) would differ as a function of operating altitude. However, the equilibrium temperature of the case at any environmental condition can be predicted using the methods for steady-state analysis given in Chapter VI. The corresponding component equilibrium temperatures are based on the experimentally determined temperature differentials for the appropriate air pressure and, if necessary, temperature level as shown in Figure VIII-18. Based on the temperature-time history of the case and the equilibrium temperatures, the temperature-time history of the components can be determined by either one of the two methods.

4. Procedures for Evaluation of Operational Conditions

The evaluation procedures for closed vented units differ only in a few details from those for pressurized units, described on pages 264 to 266. The procedures for constant and variable environments are different.

a. Constant Environment and Heat Generation. The same procedures as for the pressurized unit are applicable. The correction factor F_c, to be used to calculate the rate of heat dissipation from equation (VIII-39), is that determined from the bench test data, unless it is felt that the air circulation conditions within the installation compartment would resemble those of the altitude chamber. In the latter event, equation (VIII-40) can be used where the value of H_c must be that obtained in the altitude chamber test at the same air pressure as the operating conditions being investigated. The plots A, B and C, as shown in Figure VIII-3 for the pressurized unit, are obtained for the vented unit in the same manner. In constructing plot D, the actual temperature-time history of the case, from plot C, it must be remembered that the value of the equivalent thermal capacity K_{ev} must be obtained from a plot such as shown in Figure VIII-17 at a value of the abscissa corresponding to the air pressure at the operating conditions being investigated. Component temperature variations with time can then be determined by use of the pattern plots shown in Figure VIII-19, interpolating between lines of temperature rise ratios at known pressures. The basic procedures, not including interpolation for altitude effects, are illustrated numerically in Example VIII-2, page 269.

b. Variable Environment and Variable Heat Generation

The same step-by-step trial-and-error procedures as for pressurized units are applicable and require the same basic working plots of ambient air pressure and temperature, surrounding wall surface temperatures, and heat generation versus operating time. In addition, a working plot of equivalent thermal capacity versus time can be prepared by obtaining from data such as in Figure VIII-17, values corresponding to instantaneous ambient pressures. If the value of K_{ev} has been found to depend also on the temperature level, two curves, defined by the extreme temperature range in the plot of K_{ev} versus p_o, should be plotted versus τ.

The procedure described for the pressurized unit on pages 265 and 266 is then followed also for the vented unit. In determining the average rate of heat dissipation during a time interval, either equation (VIII-39 or -40) may be used. However, the latter can only be used if the value of H_c is defined for each pressure condition by chamber tests, or preferably by mock-up tests simulating actual compartment air circulation conditions. In determining the calculated case temperature rise by equation (VIII-7), the value of K_{ev} must be read from the working plot versus operating time, previously prepared. It should be the average value for the chosen time interval. If temperature effects on K_{ev} have been ascertained, this average should be determined at the assumed average case temperature for the interval and by interpolation between the curves of K_{ev} versus τ plotted for constant temperatures.

In the determination of component temperature-time histories by use of temperature patterns such as in Figure VIII-19, the calculation of "would-be" equilibrium temperatures of the case at the end of each time interval is again necessary and is performed using the same method of determining the rate of heat dissipation from the case as used in the interval calculation. The corresponding "would-be" equilibrium component temperatures are then defined from a plot such as Figure VIII-18. The component temperatures at the end of the time interval are defined using these data and interpolated curves in Figure VIII-19, corresponding to the ambient pressure.

<u>Vented Units with Open Case and Through-Flow of Atmospheric Air by Natural Convection</u>

Because of the complexity of the heat transfer mechanism of this type of unit, prediction of the temperature-time history can only be made in a qualitative sense. By means of analysis, based on a limited quantity of test data, reasonably good estimates can be obtained of the general component temperature level. Restricting the prediction to the variation of a general effective temperature level, rather than attempting to predict individual component temperature-time histories, makes it possible to analyze qualitatively all types of non-steady-state operating conditions.

1. Test Procedures

The necessary data for the evaluation of non-steady-state operation of this type unit, within the above-mentioned limitations, consist of the heat transfer characteristics of the components and the equivalent thermal capacity. In this type unit convective heat transfer occurs directly from components to the ambient air and also from the case which receives normally heat from components by radiation and conduction. Radiant heat transfer to the environment occurs only from the case. The test data must provide a definition of the convective heat transfer mechanism from all heat transmitting surfaces to the ambient air. For that purpose, a minimum of four test runs at equilibrium conditions are required. One is a bench test in a well-screened location, the other three are altitude chamber tests at different pressures, one being the lowest pressure at which operation is expected to be desired. The altitude chamber tests must be performed with the methods and the auxiliary test box described in Chapter IV, pages 68 and 69. In all tests it is necessary to measure at thermal equilibrium environmental air temperature, surface temperatures, and pressure, case surface temperatures (one for each 10 square inches of case surface), as many component surface temperatures as possible, and rate of heat generation. The environmental temperature should be approximately the same in all tests. The data will be affected by the temperature level to some extent. However, since the general method of analysis is only approximate, additional tests at different temperatures at the same pressures are not required.

For the determination of equivalent thermal capacity, at least the bench test must be performed as a transient test in which all temperatures, air pressure, and electrical input and output must be recorded continuously for the entire test period, starting when electrical operation is initiated and terminating when practical thermal equilibrium is attained. It is important to maintain the environment and the rate of heat generation constant during the entire test. The component temperatures and case temperatures in particular should be measured with the same thermocouples located as in the equilibrium chamber tests, discussed above.

2. Evaluation of Heat Transfer Mechanism and of Equivalent Thermal Capacity

The procedure of defining the heat transfer mechanism of the unit in an approximate manner is based on the assumption that the convective heat transfer is dependent on the physical dimensions of the system, the effective component temperature t_{ef}, being the average of a large number of individual component temperatures, and on the ambient temperature and pressure. In accordance with the free convection equation, the effective heat transfer coefficient should be directly proportional to the effective component temperature rise $(t_{ef} - t_o)$ raised to a fractional exponent. Also, for small bodies and relatively short surfaces, it should be proportional to the square root of the ambient pressure p_o. Thus the free convective heat transfer from the unit would be defined by

$$q_{ds-cv} = C\,(t_{ef} - t_o)^n\,(p_o^{0.5}). \qquad \text{(VIII-41)}$$

The radiant heat transfer rate from the case is not likely to be exactly expressed by equation (VI-4), using t_c determined as the average of a large number of individual temperatures and $t_w = t_o$. Therefore, a correction factor F_{rd} should be introduced to give the correct value from the calculated value. Thus, at equilibrium, when the total heat generated is dissipated, the heat balance equation is

$$q_{ds-eq} = q = C\,(t_{ef} - t_o)^n_{eq}\,(p_o^{0.5}) + F_{rd}\,q_{c-rd-eq} \qquad \text{(VIII-42)}$$

Equation (VIII-42) may be written for each test condition and the measured and calculated values substituted to yield an equation where C, n and F_{rd} would be the unknowns. Thus, with equilibrium test data available for at least three conditions, three equations can be written and solved for the unknowns. The solution for n would have to be made graphically, but without involving any complications. The value of n so determined should be greater than 1.0 but is likely to be smaller than 1.3.

The equivalent thermal capacity of the unit is actually somewhat affected by pressure level because part of the heat dissipated by the components is transmitted to the case and from there to the environment. However, since the method here proposed is approximate, the calculations of variable equivalent thermal capacity as function of altitude are not warranted. Thus, the determination of the equivalent thermal capacity is based on the transient bench test data and the constants determined, as described above. Having determined the numerical values of C, n, and F_{rd}, the rate of heat absorption at any time during the bench test is computed by

$$q_{ab} = q - C\,(t_{ef} - t_o)^n\,(p_o^{0.5}) - F_{rd}\,q_{c-rd}, \qquad \text{(VIII-43)}$$

where values of q, t_{ef}, t_o and p_o are obtained from the test data, and q_{c-rd} is the calculated radiant heat dissipation from the case based on the instantaneous average case surface temperature. From a plot q_{ab} versus time τ, the total heat absorption E during the test can be determined by graphical or mechanical integration like for the other types of units. Then, the equivalent thermal capacity is found from equation (VIII-6) with the effective component temperature t_{ef} as the representative temperature. Therefore,

$$K_{ev} = E/(t_{ef-eq} - t_{ef-\underline{0}}).$$

3. Procedure for Evaluation of Operational Conditions

The evaluation procedure proposed in the following is applicable to all environmental and functional conditions, but will only yield approximate indications of the temperature-time history of the effective component temperature. Some reasonable estimate can be made for the relation of any individual component temperature with the effective temperature, although it must be recognized that the relation changes with pressure and temperature.

It is assumed that the effective temperature is the principal indication of the thermal state of the unit and defines its heat transfer mechanism. The arbitrary assumption is made that the case is always at a temperature lower than the effective component temperature by the difference observed at equilibrium in the tests made at variable pressure. The difference is determined from a plot of $(t_{ef} - t_c)_{eq}$ versus p_o as measured in the tests. It is obvious that this assumption is incorrect in the initial phases of operation, or when the environment tends to provide radiant heat gain to the case. However, as long as the radiant heat dissipation, as calculated from the case temperature determined by this assumption, does not constitute more than 20 per cent of the total instantaneous heat dissipation, the error would be tolerable in the light of the approximate nature of the results expected.

The calculation procedure consists in a step-by-step trial-and-error process and should be based, like for other units, on working plots of all the variable conditions during operation. The other working plot is the mentioned variation of $(t_{ef} - t_c)_{eq}$ versus p_o which is assumed applicable under all temperature conditions and in all operating stages.

The step-by-step process is executed in the same manner as for other units. For a time interval $\Delta\tau$, chosen in length inversely to the magnitude of the general trend of temperature change, the change of effective component temperature Δt_{ef} is assumed. For the assumed value of t_{ef} at the end of the interval the estimated value of t_c is found from the plot of $(t_{ef} - t_c)$ versus p_o. Then for the average value of t_{c-av} during the time interval, the average rate of radiant heat transfer of the case $q_{c-rd-av}$ may be calculated. Similarly, the average convective heat dissipation rate is determined from the average condition, including the assumed t_{ef-av}, so that the average rate of heat absorption during the interval $\Delta\tau$ can be determined from equation (VIII-43) in the form

$$q_{ab-av} = q_{av} - C(t_{ef-av} - t_{o-av})^n (p_{o-av}^{0.5}) - F_{rd}\, q_{c-rd-av} \qquad \text{(VIII-45)}$$

The effective component temperature rise is then, similar to equation (VIII-7),

$$\Delta t_{ef} = q_{ab-av}\, \Delta\tau / K_{ev} \qquad \text{(VIII-46)}$$

The temperature rise so determined must equal the assumed value. Otherwise the procedure must be repeated. In performing the calculations, it is convenient to plot the trend of t_{ef} versus τ to facilitate successive assumptions.

As a result of these calculations, a temperature-time history of the effective component temperature is obtained which is qualitatively correct. The temperature-time history of the case, also obtained from the calculations, is likely to be considerably more inaccurate. However, the entire procedure recommended for this type unit can fulfill the useful purpose of providing some measure of knowledge of the operating temperature level to be expected which should be accepted as preferable to complete ignorance of the subject.

Vented Units with Open Case and Forced or Induced Through-Flow of Atmospheric Air

Prediction of the temperature-time history of a unit of this type can be made with good accuracy. It is necessary to secure sufficient test data to define the heat transfer mechanism between the components, the case, and the through-flowing air. However, the prediction of the temperature-time histories of individual components is not likely to be as accurate as those for pressurized units cooled externally by forced convection. While in that type of unit, component temperatures are dependent only on the case temperature, in the vented unit air temperature, case temperature and air flow rate all affect component temperatures directly. Because of the predominance of convective heat transfer in the vented unit, good accuracy is obtained by basing the prediction of component temperatures and heat transfer rates on an effective component surface temperature. In this manner, it is feasible to analyze all types of non-steady-state operating conditions of the variety of units of this type.

1. Test Procedures

The necessary data for evaluation of non-steady-state operation of this type of unit consist principally of (1) temperature-time histories of all critical components, (2) heat transfer characteristics of the component surfaces and the case, (3) air flow rates as function of environmental conditions, or flow resistance characteristics of the unit and performance of the air supply system, and (4) the equivalent thermal capacity.

The temperature-time relationships of individual components can best be related to the general temperature level of the unit. The latter is expressed in terms of the effective component temperature which constitutes the average of the surface temperatures of as many components as possible. The necessary data can be obtained in a bench test performed according to the procedures described in Chapter IV, page 60, except that all measurements, including power input and output, must be made continuously from the time the unit is energized until thermal equilibrium is obtained. Care should be taken to maintain the environmental conditions constant during the test.

The heat transfer characteristics of the component surfaces and the case can be ascertained by the series of test runs with variable air flow rate and bare case described in Chapter IV, page 62. Only equilibrium data need be taken in these tests.

If the air flow rates obtained under various operational environmental conditions will have to be calculated from known air supply characteristics, the air pressure drop across the unit must be measured in all tests with different air rates. Otherwise, it is necessary to perform air flow tests including operation of the unit's blower under the environmental conditions expected to be encountered. Since the methods would then depend on heat balance, test procedures described in Chapter IV, page 70, should be used. However, when the unit can be insulated externally, use of the test box shown in Figure IV-1 is not necessary.

As a rule, the transient bench test performed to obtain the temperature-time histories o the components also provides the data necessary for the determination of equivalent thermal capacity. It is only necessary that, if the unit has an integral blower, it be operated in conjunction with an auxiliary air flow apparatus to obtain a reliable measurement of the air flow rate throughout the test. However, forced flow units with multiple outlets in more than one surface of the case do not lend themselves to a reliable measurement of the air outlet temperature which is absolutely required for the determination of equivalent thermal capacity. For such units an additional transient test must be run in which outlets only in one surface are permitted to remain open so that exit temperatures can be determined more accurately by probing. The assumption of uniform exit velocity would be permissible. Since this alters the flow resistance, care must be taken to measure the air flow with an auxiliary flow apparatus, while environmental conditions are maintained constant. The internal flow pattern in this test is usually modified from that pertaining to the unit with all discharge openings free. Therefore, this test is not suitable for component temperature-time study and represents, as stated above, an additional test.

2. Evaluation of Equivalent Thermal Capacity

The procedures for evaluation of equivalent thermal capacity are basically the same as those for pressurized and sealed units cooled by forced convection. Like for all units the instantaneous rates of heat absorption must be determined by solving equation (VIII-1) when the rate of heat generation q is known by measurement, and the rate of heat dissipation q_{ds} is calculated. The rate of heat dissipation for this type unit is usually composed of two main parts. Namely, (1) the forced convective heat dissipation to the air passing over the component surfaces determined from equation (VIII-8), (2) the heat dissipation from the case which may consist of radiation and free convection, or radiation and forced convection.

The heat dissipation from the case would differ in character in accordance with the type. In all instances, it can be determined experimentally at equilibrium conditions by the right-hand half of equation (VIII-9). Then, by comparing the experimental result with the calculated value based on an assumption of the heat transfer characteristics, the necessary correction factors can be determined to permit determination of actual conditions during the transient part of the test by calculation.

Units with induced through-flow, having one end of the case as inlet and a single outlet at the opposite end, should in bench test only have radiant heat transfer from the inlet and outlet ends, and both radiant and free convective heat transfer from the top surface and the two sides. It is not likely that much flow would be aspirated over the latter three surfaces. Therefore, the value of F_c determined as the ratio of the calculated case heat dissipation applied to the case surfaces as mentioned, to that found by equation (VIII-9) should be close to unity and applicable with good accuracy to the determination of heat dissipation rates throughout the test. Units with induced through-flow having inlets in several surfaces and a blower located internally, can be treated in the same manner, calculating radiation only for all surfaces with inlets and both radiation and convection from the other surfaces. The construction of the air discharge opening may cause some air aspiration in the vicinity, but the effect should not be appreciable enough to change the nature of the convective heat transfer from free to forced convection. A value of F_c approaching 1.5 would indicate the latter and would require use of equation (VIII-10) for $H_e = H_c$ to insure best accuracy in the determination of equivalent thermal capacity.

Units with forced through-flow having a single inlet and the outlet in only one other side of the case can be treated like the corresponding induced-flow unit. However, it is more likely that air flow is aspirated over the closed case surfaces so that the value of F_c may be considerably greater than unity. In that event, it is desirable to define H_c by equation (VIII-10) to insure accuracy for the equivalent thermal capacity to be evaluated.

Units with forced through-flow having multiple outlets in several case surfaces do not permit the evaluation of the equivalent thermal capacity with the unit operating as designed, since, as mentioned on page 308, it is not possible to obtain a representative measurement of the air discharge temperature. Resorting to the test procedure, mentioned on page 308, which involves blocking all outlets except those in one side of the case, introduces some deviations of component temperatures. Thus, the effective component temperature at the same air flow may differ from that obtained with all outlets free to discharge. This would introduce some inaccuracy in effective component temperature rise of the unit calculated on basis of the equivalent thermal capacity so determined. However, there is no alternative to the choice of this minor modification of the unit in this instant which would yield more accurate data. Thus the unit, in fact, becomes one with single inlet and outlet in opposite surfaces and can be treated in the same manner to determine the necessary coefficients for the calculation of instantaneous rates of heat dissipation during the entire test.

The subsequent procedure for evaluation of heat absorption rates for all variations of this type unit is the same. It follows that for the pressurized and sealed units, as described on page 277, using plots of t_1, t_2, t_c, W and t_o versus τ. For the calculation of heat dissipation rates equation (VIII-12a or -12b) may be used, depending on which would be found appropriate in accordance with the criteria discussed above. In particular, it should be remembered that the heat dissipation rates of the case would be calculated based on portions of the case surface, rather than its entirety, which are assumed to have radiant heat dissipation alone, or convective heat dissipation also.

In defining the equivalent thermal capacity from equation (VIII-6), the effective component temperature, computed as the average of as many component surface temperatures as possible, is taken as the representative temperature. It must be evaluated at the non-operative equilibrium condition at the beginning of the test as $t_{ef-\underline{O}}$, presumably identical to $t_{o-\underline{O}}$, and at the operative equilibrium condition at the end of the test as t_{ef-eq}. Thus, the appropriate form of equation (VIII-6) for this type unit is the same as equation (VIII-44) for the unit with natural through-flow.

3. Methods of Evaluating Component Temperature Variation in Non-Steady-State Operation

Variation in component temperatures during non-steady-state operation for a unit having an open case and forced or induced flow of atmospheric air would be predicted from characteristic temperature patterns of the components and the effective temperature of the components t_{ef}. The characteristic temperature patterns would be established from bench-test data of the temperature-time histories of the various thermally critical components by plotting the parameters

$$(t_{cp} - t_{cp-\underline{O}})/(t_{cp-eq} - t_{cp-\underline{O}}) \text{ versus } (t_{ef} - t_{ef-\underline{O}})/(t_{ef-eq} - t_{ef-\underline{O}})$$

Component temperatures during transient operation of the unit would be predicted from the characteristic temperature patterns, the predicted variation of the effective temperature with time of operation and the equilibrium component temperature differential $(t_{cp-eq} - t_{ef-eq})$. The more approximate method for predicting component temperatures presented on pages 262 to 264 is not recommended for use with a unit of this type, but would be employed whenever the characteristic temperature patterns described above are not available and it is necessary to make a rough estimation of the variation of component temperature with time of operation.

4. Evaluation Procedures for Determining Non-Steady State Thermal Performance

Non-steady state evaluation procedures are discussed first for units admitting air in one end of the case and discharging it through the opposite end. This arrangement for through-flow of atmospheric air may exist with forced or induced flow created by a blower or by ram action. Bench tests provide steady-state data with the case surfaces insulated and bare when the air flow is not metered, and for the external surfaces bare only, when the air flow is metered. The heat dissipated to the cooling air q_{ds-cv} would be evaluated from equation (VIII-8), using the measured values of the air rate and air temperature at inlet and outlet of the unit. When an air metering device, such as the auxiliary air flow apparatus, is not employed during test, the air rate would be evaluated by equation (VIII-8) using measured values of the inlet and outlet air temperature when the case surface is insulated and the rate of heat dissipation is equal that of heat generation. The case heat transfer rate q_c when the case surface is bare would be evaluated by heat balance as the difference of the rate of heat generation q and the rate of heat dissipation to the through-flow of air q_{ds-cv}.

A value for the correction factor F_c is defined from this case heat transfer and the corresponding calculated case heat transfer by free convection and radiation. Values of case heat transfer and heat dissipation to the through-flow of air would be used to construct two working charts needed for non-steady state analysis. The parameter $q_{ds-cv}/(t_{ef} - t_1)$ would be plotted as a function of the air rate W, where t_{ef} is the effective temperature of the components and t_1 is the temperature of the through-flowing air at entrance to the unit. The plot so defined represents in a general manner the heat transfer process between the components and the through-flow of air, and would be used to define the heat dissipation to the cooling air corresponding to specified values of the effective and inlet air temperature and the rate of air flow. The second working chart consists of a plot of the parameter $q_c/(t_{ef} - t_c)$ as a function of the temperature differential $(t_{ef} - t_c)$. Values of the parameter $q_c/(t_{ef} - t_c)$ would be evaluated at each test air rate and plotted as illustrated in Figure VIII-20. The value of the parameter is zero at a temperature differential of zero. Hence, the variation of the parameter with the temperature differential $(t_{ef} - t_c)$ is established in an approximate manner by constructing curves starting at the origin and passing through the test points. The curves would be drawn slightly concave upward since radiant heat transfer increases at a rate greater than in direct proportion to the temperature difference. Each curve corresponds to a constant air rate W. When the unit is tested and is to be evaluated subsequently for a constant or nearly constant air rate, a single curve so constructed would be used. Then, however, it is advantageous for purposes of analysis to convert the curve so constructed into a plot of case heat transfer q_c as a function of the temperature differential $(t_{ef} - t_c)$, since values of case heat transfer may be read directly from this type plot. Additional data to aid in the construction of the working plot illustrated in Figure VIII-20 would be obtained by testing the unit with various thicknesses of insulation on the external surface to decrease the case heat transfer, and by creating forced air flow over the external surface to increase the case heat transfer.

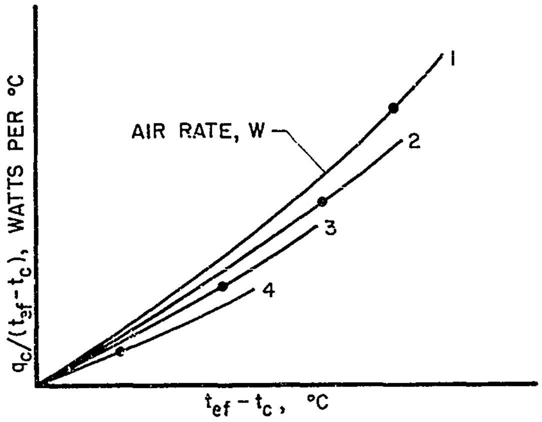


Figure VIII-20. Working Plot for Evaluation of Case Heat Transfer

The evaluation procedure for determining non-steady state thermal conditions requires a step-by-step calculational process. A time interval of operation is selected, which defines the average elapsed time of operation for the interval, and from which average values for the ambient air pressure and temperature and surrounding wall temperature are selected. Next, values for the change in the effective temperature Δt_{ef} and the case temperature Δt_c are assumed. The average effective temperature t_{ef-av} and the average case temperature t_{c-av} would then be defined for the selected time interval. The correctness of the assumed value of the case temperature change Δt_c may now be checked. The case heat transfer rate q_{c-av} by free convection and radiation between the surface of the case and its environment would be evaluated using the assumed average case temperature, the correction factor F_c established from bench test and the average environmental conditions of pressure and temperature. A second value for the case heat transfer is defined from the working plot illustrated in Figure VIII-20, corresponding to the temperature differential $(t_{ef-av} - t_{c-av})$ and the average air rate W_{av}. When flow induced by a blower exists, it is necessary to assume at this point in the analysis, a tentative value for the average air rate to be used with Figure VIII-20, which would be verified in a subsequent part of the analysis. Agreement of this second value of the average rate of case heat transfer q_{c-av} determined from Figure VIII-20, with the first value determined by evaluation from the external heat transfer characteristics would indicate that a correct value of the case temperature change Δt_c has been assumed.

The average rate of heat dissipation to the through-flow of air q_{ds-cv} would be evaluated next. With forced-flow of air created by a blower, the air rate remains unaffected by the temperature change of the cooling air during passage through the unit. The average air rate would be defined corresponding to the operational conditions of the blower and the pressure and temperature of the environment. The value of the parameter $q_{ds-cv}/(t_{ef} - t_1)$

is then defined by the average air rate from the working plot of this parameter versus air rate established from bench test of the unit, and the value of the parameter would be used to evaluate $q_{ds-\underline{cv}-av}$ corresponding to the assumed average effective temperature t_{ef-av} and the inlet air temperature t_{1-av}. The inlet temperature t_{1-av} equals the average ambient temperature plus any temperature rise of the air created by the blower.

The average rate of heat absorption of the unit q_{ab-av} would be evaluated by heat balance, using the average values of the case heat transfer rate and the unit's heat generation rate. The heat balance is defined by

$$q_{ab-av} = q_{av} - q_{ds-\underline{cv}-av} - q_{c-av} \qquad \text{(VIII-47)}$$

Lastly, the change in the effective temperature Δt_{ef} would be evaluated from the calculated average rate of heat absorption and the unit's equivalent thermal capacity by the equation

$$\Delta t_{ef} = q_{ab-av} \Delta\tau / K_{ev} \qquad \text{(VIII-48)}$$

Agreement of this value of Δt_{ef} with the value assumed at the start of the procedure indicates that Δt_{ef} is correct for the selected time interval of operation. This same procedure of analysis would be repeated for all subsequent time intervals of operation.

If the blower induces air flow through the unit, the air rate will vary with time of operation to a small extent, because of the change in air density at the blower inlet as the result of the variable heat dissipation rate to the cooling air. Neglecting any effect of pressure drop of the air across the unit on the air density at the blower inlet, the variation in induced air rate with the average rate of heat dissipation to the cooling air would be evaluated by use of the equation

$$W_{av} = \left[W_{eq}(273 + t_{1-eq}) + (q_{ds-\underline{cv}-eq} - q_{ds-\underline{cv}-av})/456\right] \Big/ (273 + t_{1-av}) \qquad \text{(VIII-49)}$$

The equilibrium values of the air rate, inlet air temperature and heat dissipation to the air would be used as those determined during steady-state bench test of the unit. The use of equation (VIII-49) produces results of fairly good accuracy for most units of this type, unless the operational condition of the unit corresponds to high altitude where the pressure drop of the air across the unit may produce an appreciable change in the air density. Under the latter conditions, the variation in air rate would be by the procedures outlined in Chapter V.

The non-steady state evaluation procedure when the air flow is induced through the unit is basically the same as for forced flow, with the exception of accounting for the variation in cooling air rate. To include this latter effect, referring to the previously described procedure, after the correct change in case temperature Δt_c has been established it is then necessary to assume a value for the average rate of heat dissipation $q_{ds-\underline{cv}-av}$. The assumption would be based on an estimated air rate, the effective temperature t_{ef-av}, the inlet temperature t_{1-av}, and the value of the parameter $q_{ds-cv}/(t_{ef} - t_1)$, since the air rate variation is small. The actual average air rate W_{av} would then be evaluated by equation (VIII-49), after which the heat dissipation to the cooling air $q_{ds-cv-av}$ would once again be calculated. Agreement of this latter value of $q_{ds-cv-av}$ with the value originally assumed defines the correct value. The remainder of the evaluation procedure is identical to that previously outlined, for forced flow. The evaluation procedure with induced flow is illustrated in Example VIII-5 at the end of this section.

With units having induced flow and the air admitted through openings in end and side panels of the case, the heat dissipation to the cooling air and the case heat transfer are defined from the bench test data in a manner identical to that outlined on page 309 for determination of thermal capacity. The heat transfer characteristics would be generalized according to the previous procedure for units admitting and discharging the air through opposite ends of the case. Also, the evaluation procedure for predicting non-steady state thermal conditions is the same as previously presented. When evaluating the external case heat transfer rate, the entire case surface would be used in defining the radiant heat transfer but only those portions of the case surface not having inlets or outlets would be used in defining free convective heat transfer.

Units having forced flow of cooling air with multiple outlets in the case would be evaluated in the same general way as the previous types. The heat transfer characteristics would be defined from bench tests following the procedures outlined on page 309 for determination of thermal capacity. The same evaluation procedure for determining non-steady state thermal performance would be used as that outlined in the previous paragraphs for the unit admitting and discharging air through opposite ends of the case. Case heat transfer by radiation and free convection would be evaluated according to the procedure for units having induced flow and multiple openings in the case.

Example VIII-5. Prediction of the Non-Steady State Thermal Conditions for a Vented Unit with Open Case and Mechanically Induced Through-Flow of Atmospheric Air

It is desired to predict the thermal condition of a vented unit cooled by through-flow of atmospheric air when operated for one-half hour in an environment having a uniform temperature of 71°C and a pressure equal to that during bench test, 29.5 inches of mercury absolute. The prediction of the thermal state is based upon data secured during bench test of the unit, where the environment is at a uniform temperature of 25°C.

The unit generates heat at a constant rate of 292 watts. Cooling air is induced through the unit by a blower. The air is admitted through multiple openings in one end of the case and is discharged through a single outlet on the opposite end where the blower-motor set is mounted. The case is 8 inches wide, 6 inches high and 15 inches long, and all surfaces are painted black. The equivalent thermal capacity K_{ev} has been determined from bench test and is 95 watt-minutes per °C.

A steady-state bench test of the unit provides the following data.

Environment	
temperature, t_o	25°C
pressure, p_o	29.5 inches mercury absolute
wall temperature, t_w	25°C
confinement	large room
Air rate, W	0.033 pound per second
Average case temperature, t_c	48°C
Effective component temperature, t_{ef}	115°C
Air outlet temperature, t_2	38.9°C

a. Reduction of Bench-Test Data

The bench-test data are used to define the generalized heat transfer characteristics of the unit. The heat dissipated to the cooling air is evaluated by equation (VIII-8)

$$q_{ds-\underline{cv}} = 456 \times 0.033\ (38.9 - 25) = 209 \text{ watts}$$

Hence, by heat balance, the case heat transfer rate is

$$q_c = q - q_{ds-\underline{cv}} = 292 - 209 = 83 \text{ watts}$$

The correction factor F_c is now determined by calculating the case heat transfer from the measured average case temperature of 48°C and the air pressure of 29.5 inches of mercury absolute, by use of the free convection and radiation charts in Figures VI-2 and -3. It is assumed that free convective heat transfer occurs only from the two sides and top surfaces of the case and radiant heat transfer from the entire case surface. Following the procedure outlined in Example VI-1, the free convective and radiant heat transfer rates are calculated as 27.7 and 49 watts, respectively. Thus,

$$F_c = q_c/q_{c-calc} = 83/(27.7 + 49) = 1.08$$

The forced convection heat transfer parameter corresponding to the air rate of 0.033 pound per second is

$$q_{ds-\underline{cv}}/(t_{ef} - t_1) = 209/(115 - 25) = 2.32 \text{ watts per } ^{\circ}\text{C}$$

The case heat transfer parameter is

$$q_c/(t_{ef} - t_c) = 83/(115 - 48) = 1.24 \text{ watts per } ^{\circ}\text{C},$$

corresponding to the temperature differential $(t_{ef} - t_c) = 115 - 48 = 67^{\circ}$C.

Three working plots are next constructed: (1) the parameter $q_{ds-\underline{cv}}$ versus the air rate W, (2) the parameter $q_c/(t_{ef} - t_c)$ versus the temperature differential $(t_{ef} - t_c)$, and (3) the case heat transfer rate q_c versus the case temperature t_c. The first plot is presented in Figure

VIII-21, where the curve has been constructed by assuming the parameter to vary in proportion to the air rate raised to the 0.8 power, since bench-test data at variable air rate have not been obtained for this unit. Ordinarily, a typical value for the exponent of the air rate would be known for any unit from other bench tests. Also, the air rate during non-steady state operation does not vary appreciably from the steady-state value, so the importance of the accuracy of an assigned exponent is not too great, if it is between 0.6 and 0.9.

The second working plot is shown in Figure VIII-22, where the variation of the parameter $q_c/(t_{ef} - t_c)$ with the temperature differential $(t_{ef} - t_c)$ has been constructed in an approximate manner. A curve has been drawn from the origin through the value of the parameter determined previously from the bench-test data. The accuracy of this plot would not be expected to be good in the region of low values of the temperature differential. However, this region corresponds to the early phase of non-steady state operation where the case heat transfer rate is but a small percentage of the rate of heat absorption by the unit, and even a relatively large error in the plot would have a minor effect on the temperature-time history of the unit. The third working chart is constructed as an aid to the evaluation procedure to reduce the amount of computation. Since the environmental temperatures and pressure are constant for the operational conditions to be studied, 71°C and 29.5 inches of mercury absolute, respectively, the heat transfer between the case and its environment by free convection and radiation are evaluated for a number of values of the case temperature t_c, using the charts in Figures VI-2 and -3 and the correction factor F_c equal to 1.08. The resulting plot is presented in Figure VIII-23, where for any assumed value of the case temperature the heat transfer rate q_c may be read directly.

The equation for evaluating the cooling air rate induced by the blower is derived from equation (VIII-49). Equilibrium values of W, t_1 and q_{ds-cv} may be used as those existing during steady-state bench test of the unit, i.e., 0.033 pound per second, 25°C and 209 watts, respectively. Hence,

$$W_{av} = (10.3 - q_{ds-cv-av}/456)/344$$

where the inlet temperature t_{1-av} in equation (VIII-49) is constant and equal to 71°C for this example.

b. Prediction of Non-Steady State Thermal Conditions

The variation of the effective component temperature t_{ef} with time of operation is evaluated by the procedure outlined on pages 312 to 314. The procedure is illustrated in this example for an intermediate time interval of operation. After 20 minutes of operation, the temperature of the case t_c and the effective temperature of the components t_{ef} have been determined to be 118.5° and 78°C, respectively. A time interval $\Delta\tau$ equal to 5 minutes is selected. Hence, τ_1 = 20 minutes, τ_2 = 25 minutes, t_{c-1} = 78°C, t_{ef-1} = = 118.5°C, and the temperatures t_{c-2} and t_{ef-2} are to be evaluated.

The change in the effective temperature Δt_{ef} and the case temperature Δt_c for the time interval are assumed equal to 8° and 2°C, respec-

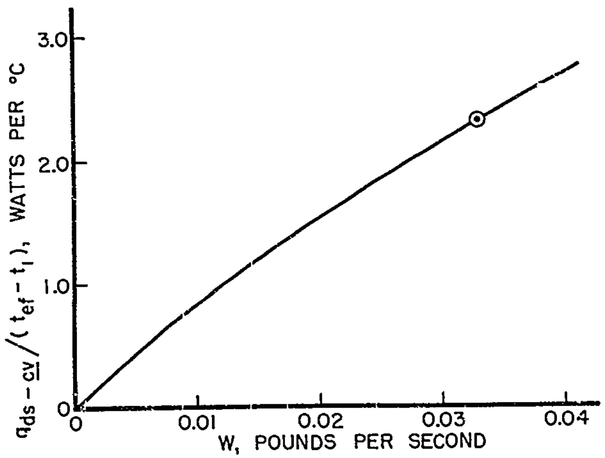


Figure VIII-21. Working Plot for Evaluation of Heat Dissipation to Through-Flow of Atmospheric Air (Example VIII-5)

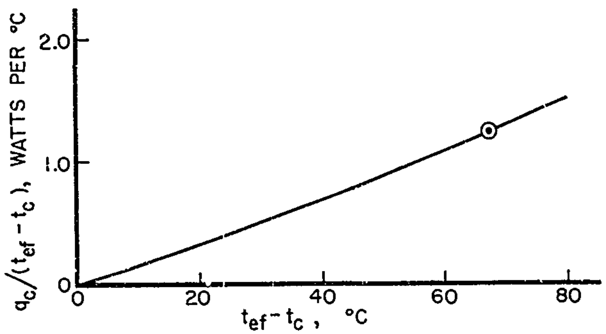


Figure VIII-22. Working Plot for Evaluation of Case Heat Transfer Rate (Example VIII-5)

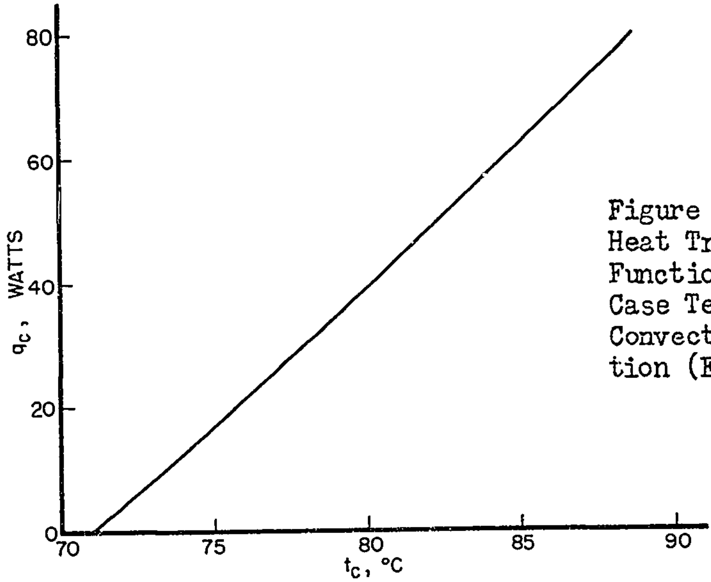


Figure VIII-23. Case Heat Transfer Rate as Function of the Average Case Temperature, Free Convection and Radiation (Example VIII-5)

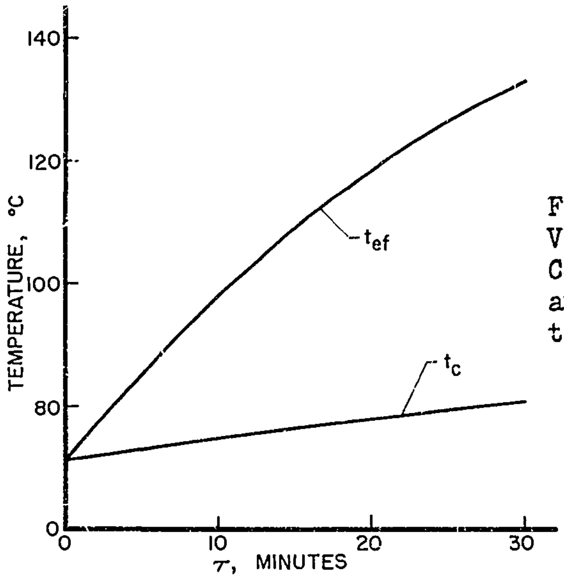


Figure VIII-24. Predicted Variation of Effective Component and Case Temperature with Time of Operation (Example VIII-5) tively. Thus, $t_{ef-av} = 118.5 + 8/2 = 122.5^\circ C$ and $t_{c-av} = 78 + 2/2 = 79^\circ C$. The temperature differential $(t_{ef-av} - t_{c-av}) = 122.5 - 79 = 43.5^\circ C$ defines from Figure VIII-22 a value for $q_c/(t_{ef} - t_c) = 0.76$ watt per $^\circ C$, so that $q_{c-av} = 0.76 \times 43.5 = 33$ watts. A second value for the case heat transfer rate q_c is defined by $t_{c-av} = 79^\circ C$ from Figure VIII-23; $q_{c-av} = 34.7$ watts. Since the values are slightly different, a new value for Δt_c is assumed equal to $1.5^\circ C$, whereupon the case heat transfer rates agree at 33.5 watts.

Next, it is necessary to assume a value for $q_{ds-cv-av}$ to define the air rate. Set $q_{ds-cv-av} = 110$ watts, and from the air rate equation defined in part (a) of this example,

$$W_{av} = (10.3 - 110/456)/344 = 0.0292 \text{ pound per second.}$$

From Figure VIII-21 for this value of the air rate, $q_{ds-cv}/(t_{ef} - t_1) = 2.1$ watts per $^\circ C$. Hence, $q_{ds-cv} = 2.1\ (122.5 - 71) = 108$ watts. This value is close enough to the assumed value of 110 watts, so another trial would have no effect on the air rate and the value of 108 watts is correct. The average rate of heat absorption by the unit over the 5-minute time interval is, by equation (VIII-47),

$$q_{ab-av} = 292 - 108 - 33.5 = 150.5 \text{ watts}$$

Hence, by equation (VIII-48),

$$\Delta t_{ef} = 150.5 \times 5/95 = 7.92^\circ C$$

This value is sufficiently close to the assumed value of $8^\circ C$ that another set of computations is not required. Therefore, the effective temperature and the case temperature after 25 minutes of operation are equal to,

$$t_{ef-2} = 118.5 + 7.9 = 126.4^\circ C$$

$$t_{c-2} = 78 + 1.5 = 79.5^\circ C$$

The variation in effective component temperature and case temperature for the unit operating for a period of one-half hour in an environment at $71^\circ C$ are shown in Figure VIII-24. Temperature-time histories of the components would be evaluated by the procedure outlined on page

APPENDIX I

PHYSICAL PROPERTIES OF AIR

Physical Properties of Dry Air

Knowledge of physical properties of dry air is needed for analysis of heat transfer processes, flow resistance and energy or thermal balances associated with the general problem of cooling airborne electronic equipment. Table A-I-1 presents the physical properties of specific heat at constant pressure c_p, thermal conductivity k, viscosity μ, and Prandtl number Pr, for the temperature range $-100°$ to $500°C$. The equivalent range is $-148°$ to $932°F$ and the corresponding values are indicated on the right of the table. The temperature intervals of $50°C$ permit reasonably accurate evaluation of the physical properties at intermediate temperatures by linear interpolation. The effect of pressure on these properties may be ignored when used for thermal evaluation work. Two sets of units are used. Figure A-I-1 (in back pocket), presents graphically these same data, but for only one set of units.

Specific heat data in Table A-I-1 are expressed in the units Btu per pound-°F and watt-hours per pound-°C. Given, for example, air flowing at the rate of 100 pounds per hour and undergoing an increase in temperature from 0° to 100°C, the heat dissipation rate to the air may be evaluated directly in watts by selecting c_p in the units watt-hours per pound-°C at the average temperature of 50°C; i.e., 100 x 0.1268 x (100 - 0) = 1268 watts. On the other hand, if the same temperature rise were expressed as 32° to 212°F, the heat dissipation rate to the air could be evaluated directly in Btu per hour by selecting c_p in Btu per pound-°F at the average temperature of 122°F; i.e., 100 x 0.2405 (212 - 32) = 4330 Btu per hour. Similarly, thermal conductivity data are presented in two sets of units; Btu per hour-foot-°F units to be used with degrees Fahrenheit and rate of heat transfer expressed in Btu per hour; and in watts per foot-°C to be used with degrees centigrade and rate of heat transfer expressed in watts.

Viscosity data are expressed in pound-seconds per square foot and pounds per foot-hour in Table A-I-1 and in pounds per foot-second in Figure A-I-1. The latter two sets of units include the dimensional constant 4.16×10^8 feet per hour squared, respectively 32.17 feet per second squared, for conversion to absolute (mass) units. The former set of units, i.e., pound-seconds per square foot, is based on the gravitational system with weight expressed as force. In practical applications of analyzing fluid flow or heat transfer, viscosity data are needed for evaluation of the Reynolds number, defined as $WL'/(g\mu A')$ when flow rate W and viscosity μ are expressed in the gravitational system. Since Reynolds number is a dimensionless parameter and L' is some significant dimension expressed in feet and A' is the cross sectional area of the flow passage, inspection of the equation shows that with W expressed in pounds per second, μ should be used in the units pound-seconds per square foot. However, the gravitational constant g could be included in the viscosity to yield units of pounds per foot-second

Table A-I-1. Physical Properties of Dry Air

Temperature °C	Specific Heat at Constant Pressure, c_p		Thermal Conductivity, k		Viscosity, μ		Prandtl number Pr	Temperature °F
	Btu per pound-°F	Watt-hours per pound-°C	Btu per hour-foot-°F	Watts per foot-°C	Pound-seconds per square foot	Pounds per foot-hour		
-100	0.2391	0.1261	0.89×10^{-2}	0.47×10^{-2}	0.245×10^{-6}	0.0283	0.756	-148
- 50	0.2395	0.1263	1.16×10^{-2}	0.61×10^{-2}	0.308×10^{-6}	0.0357	0.744	- 58
0	0.2400	0.1265	1.40×10^{-2}	0.74×10^{-2}	0.363×10^{-6}	0.0420	0.720	32
50	0.2405	0.1268	1.63×10^{-2}	0.86×10^{-2}	0.411×10^{-6}	0.0476	0.701	122
100	0.2414	0.1273	1.85×10^{-2}	0.98×10^{-2}	0.457×10^{-6}	0.0529	0.689	212
150	0.2427	0.1280	2.10×10^{-2}	1.09×10^{-2}	0.500×10^{-6}	0.0579	0.682	302
200	0.2446	0.1290	2.27×10^{-2}	1.20×10^{-2}	0.540×10^{-6}	0.0626	0.677	392
250	0.2480	0.1308	2.47×10^{-2}	1.30×10^{-2}	0.579×10^{-6}	0.0670	0.673	482
300	0.2497	0.1317	2.67×10^{-2}	1.41×10^{-2}	0.616×10^{-6}	0.0714	0.670	572
350	0.2524	0.1331	2.86×10^{-2}	1.51×10^{-2}	0.651×10^{-6}	0.0754	0.667	662
400	0.2552	0.1346	3.05×10^{-2}	1.61×10^{-2}	0.684×10^{-6}	0.0792	0.664	752
450	0.2581	0.1361	3.23×10^{-2}	1.70×10^{-2}	0.715×10^{-6}	0.0828	0.662	842
500	0.2607	0.1375	3.41×10^{-2}	1.80×10^{-2}	0.745×10^{-6}	0.0863	0.660	932

Source of data: "An Investigation of Aircraft Heaters, II-Properties of Gases," by Myron Tribus and L. M. K. Boelter, NACA W.R. W-9, originally issued October, 1942.

or pounds per foot-hour. Thus, Reynolds number may be evaluated as $WL'/(\mu A')$, where with W expressed in pounds per second μ would be expressed in pounds per foot-second, or if W has the units pounds per hour then μ would be in pounds per foot-hour. The choice of the set of units for viscosity varies with the type of evaluation problem.

The Prandtl number Pr is a dimensionless ratio equal to $c_p\mu/k$ used in the analysis of forced convection heat transfer.

Density of Dry Air

For most thermal evaluation work it is permissible to neglect the effect of presence of water vapor in air and evaluate air density on the basis of dry air alone. As such, the relation between air density ρ, absolute static pressure p, and absolute temperature T is expressed by the perfect gas law

$$\rho = 0.738\ p/T \qquad \text{(A-I-1)}$$

where ρ has the units of pounds per cubic foot, p inches of mercury absolute, and T is in $^{\circ}K = 273 + {}^{\circ}C$.

The absolute static pressure in an enclosure is the sum of the static pressure and the barometric pressure. The static gage pressure is indicated by any ordinary pressure gage or manometer placed in the ambient atmosphere. If the static gage pressure is measured by a column of water, the absolute static pressure in inches of mercury may be determined by the equation

$$p = p_{bar} + 0.0738\ p_g \qquad \text{(A-I-2)}$$

inches mercury = inches mercury + 0.0738 (inches water gage)

A graphical solution to equation (A-I-1) is presented in Figure A-I-2 (in back pocket), where for any air temperature and absolute static pressure the density of dry air may be read directly.

Example A-I-1. Calculation of Density of Dry Air

It is required to define the density of dry air at 200°C and 20 inches water static gage pressure. The barometric pressure is 29.5 inches mercury.

By equation (A-I-2),

$$p = 29.5 + 0.0738 \times 20 = 30.98 \text{ inches mercury absolute}$$

Also,

$$T = 200 + 273 = 473^{\circ}K$$

Then, by equation (A-I-1),

$$\rho = 0.738 \times 30.98/473 = 0.0483 \text{ pound per cubic foot}$$

The same value may be read from Figure A-I-2 for 200°C and 30.98 inches mercury absolute.

Properties of Standard Atmospheres

In comparative performance analysis of airborne equipment it is necessary to assume standard variations of atmospheric pressure and temperature with altitude. This is particularly important for air-cooled electronic equipment which utilizes atmospheric air as coolant. The properties of the N.A.C.A. standard atmosphere are given in Table A-I-2. This standard has been used extensively. Particularly the atmospheric pressure variation with altitude it prescribes is generally accepted. However, its temperatures are lower than those which are usually encountered in the atmosphere. The Air Force summer atmosphere given in Table A-I-3 describes higher temperatures in the lower altitude regions and is, therefore, more conservative for cooling estimates. Its pressures are the same as those of the N.A.C.A. standard atmosphere. Still other variations of atmospheric temperature with altitude are given in Figure A-I-3. Various variations of minimum temperatures are shown which, however, do not have particular significance in general cooling problems. The temperature variation indicated with a dashed line assumes higher values than the N.A.C.A. standard and Air Force summer atmospheres at altitudes above 40,000 feet and is being used in the analysis of equipment cooling problems at higher altitudes.

Available Cooling Air Temperature in Flight

The temperature of cooling air which may be taken aboard an aircraft in flight would differ from the atmospheric temperature by virtue of ram compression and/or skin friction. This is particularly true at high flight speed. The temperature rise that may be expected is best correlated with the flight Mach number M which is the ratio of the aircraft velocity to the velocity of sound. The resulting temperature is designated as the total temperature and is

$$T_t = T_o (1 + 0.2 M^2) \qquad \text{(A-I-3)}$$

The temperature rise in degrees centigrade is very closely approximated by

$$\Delta t = (v/100)^2, \qquad \text{(A-I-4)}$$

where v is the flight speed in miles per hour.

The pressure of the air in an aircraft compartment is not as specifically defined as the temperature of the air which may be taken aboard since the available pressure is greatly dependent on the configuration and location of the intake.

Table A-I-2. Properties of N.A.C.A. Standard Atmosphere

Altitude, feet	Temperature °F	Temperature °C	Pressure, inches mercury absolute	Pressure divided by pressure at sea level	Density, pound per cubic foot	σ, density divided by density at sea level	Velocity of sound, miles per hour
Sea level	59.00	15.00	29.92	1.0000	0.07650	1.0000	761.6
1,000	55.44	13.02	28.85	0.9644	0.07429	0.9711	759.0
2,000	51.87	11.04	27.82	0.9298	0.07212	0.9428	756.3
3,000	48.31	9.06	26.82	0.8963	0.07001	0.9152	753.7
4,000	44.74	7.08	25.84	0.8637	0.06794	0.8881	751.0
5,000	41.18	5.10	24.90	0.8321	0.06592	0.8617	748.4
6,000	37.62	3.12	23.98	0.8014	0.06395	0.8359	745.7
7,000	34.05	1.14	23.09	0.7717	0.06202	0.8107	743.0
8,000	30.49	- 0.84	22.22	0.7428	0.06013	0.7860	740.4
9,000	26.92	- 2.82	21.39	0.7148	0.05829	0.7620	737.7
10,000	23.36	- 4.80	20.58	0.6877	0.05650	0.7385	734.9
11,000	19.80	- 6.78	19.79	0.6614	0.05474	0.7156	732.2
12,000	16.23	- 8.76	19.03	0.6360	0.05303	0.6952	729.5
13,000	12.67	-10.74	18.29	0.6113	0.05136	0.6714	726.8
14,000	9.10	-12.72	17.58	0.5875	0.04972	0.6500	724.0
15,000	5.54	-14.70	16.89	0.5644	0.04813	0.6292	721.2
16,000	1.98	-16.68	16.22	0.5420	0.04659	0.6090	718.5
17,000	- 1.59	-18.66	15.57	0.5203	0.04507	0.5892	715.7
18,000	- 5.15	-20.64	14.94	0.4994	0.04360	0.5699	712.9
19,000	- 8.72	-22.62	14.34	0.4792	0.04216	0.5511	710.1
20,000	-12.28	-24.60	13.75	0.4596	0.04076	0.5328	707.3
21,000	-15.84	-26.58	13.18	0.4406	0.03940	0.5150	704.5
22,000	-19.41	-28.56	12.64	0.4223	0.03807	0.4976	701.6
23,000	-22.97	-30.54	12.11	0.4047	0.03677	0.4807	698.8
24,000	-26.54	-32.52	11.60	0.3876	0.03551	0.4642	695.9
25,000	-30.10	-34.50	11.10	0.3711	0.03428	0.4481	693.1
26,000	-33.66	-36.48	10.63	0.3552	0.03309	0.4325	690.2
27,000	-37.23	-38.46	10.17	0.3399	0.03192	0.4173	687.3
28,000	-40.79	-40.44	9.727	0.3251	0.03079	0.4025	684.4
29,000	-44.36	-42.42	9.299	0.3108	0.02970	0.3882	681.5
30,000	-47.92	-44.40	8.886	0.2970	0.02862	0.3741	678.5

Table A-I-2 (continued)

Altitude, feet	Temperature °F	Temperature °C	Pressure, inches mercury absolute	Pressure divided by pressure at sea level	Density, pound per cubic foot	σ, density divided by density at sea level	Velocity of sound, miles per hour
31,000	-51.48	-46.38	8.488	0.2837	0.02759	0.3606	675.6
32,000	-55.05	-48.36	8.105	0.2709	0.02657	0.3473	672.6
33,000	-58.61	-50.34	7.737	0.2586	0.02559	0.3345	669.7
34,000	-62.18	-52.32	7.381	0.2467	0.02463	0.3220	666.7
35,000	-65.74	-54.30	7.040	0.2353	0.02371	0.3099	663.7
35,332	-67.6	-55.33	6.923	0.2314	0.02339	0.3058	662.1
36,000	-67.6	-55.33	6.714	0.2244	0.02280	0.2981	662.1
37,000	-67.6	-55.33	6.397	0.2138	0.02176	0.2845	662.1
38,000	-67.6	-55.33	6.098	0.2038	0.02074	0.2711	662.1
39,000	-67.6	-55.33	5.810	0.1942	0.01977	0.2584	662.1
40,000	-67.6	-55.33	5.538	0.1851	0.01884	0.2463	662.1
41,000	-67.6	-55.33	5.278	0.1764	0.01795	0.2347	662.1
42,000	-67.6	-55.33	5.030	0.1681	0.01711	0.2237	662.1
43,000	-67.6	-55.33	4.796	0.1603	0.01631	0.2132	662.1
44,000	-67.6	-55.33	4.569	0.1527	0.01554	0.2032	662.1
45,000	-67.6	-55.33	4.356	0.1456	0.01481	0.1936	662.1
46,000	-67.6	-55.33	4.150	0.1387	0.01412	0.1846	662.1
47,000	-67.6	-55.33	3.955	0.1322	0.01346	0.1759	662.1
48,000	-67.6	-55.33	3.770	0.1260	0.01282	0.1676	662.1
49,000	-67.6	-55.33	3.593	0.1201	0.01222	0.1598	662.1
50,000	-67.6	-55.33	3.426	0.1145	0.01165	0.1523	662.1
60,000	-67.6	-55.33	2.130	0.0713	0.00721	0.0942	662.1
70,000	-67.6	-55.33	1.320	0.0442	0.00447	0.0584	662.1
80,000	-67.6	-55.33	0.820	0.0274	0.00277	0.0362	662.1
90,000	-67.6	-55.33	0.509	0.0170	0.00172	0.0225	662.1
100,000	-67.6	-55.33	0.317	0.0106	0.00107	0.0140	662.1
104,987	-67.6	-55.33	0.249	0.00831	0.000842	0.0110	662.1
110,000	-47.4	-44.1	0.197	0.00658	0.000633	0.00827	679.0
120,000	- 7.2	-21.8	0.127	0.00426	0.000373	0.00488	711.1
130,000	33.0	0.6	0.0859	0.00287	0.000231	0.00302	742.5
140,000	73.3	22.9	0.0595	0.00199	0.000148	0.00193	771.8
150,000	113.5	45.3	0.0425	0.00142	0.0000979	0.00128	800.5

Source of data: "Notes and Tables for Use in the Analysis of Supersonic Flow," by The Staff of the Ames 1- and 3-foot Supersonic Wind-Tunnel Section, N.A.C.A. Technical Note No. 1428, December, 1947.

Table A-I-3. Properties of Air Force Summer Atmosphere

(Variation of pressure with altitude same as for N.A.C.A. standard atmosphere)

Altitude, feet	Temperature °F	Temperature °C	Density, pound per cubic foot	σ, density divided by N.A.C.A. sea level standard	Velocity of sound, miles per hour
Sea level	100.0	37.8	0.07090	0.927	790.8
1,000	96.4	35.8	0.06881	0.899	788.4
2,000	92.8	33.8	0.06678	0.873	785.7
3,000	89.2	31.8	0.06476	0.847	783.2
4,000	85.6	29.8	0.06283	0.821	780.7
5,000	82.0	27.8	0.06093	0.796	778.2
6,000	78.4	25.8	0.05910	0.772	775.5
7,000	74.8	23.8	0.05729	0.749	772.9
8,000	71.2	21.8	0.05549	0.725	770.3
9,000	67.6	19.8	0.05376	0.703	767.8
10,000	64.0	17.8	0.05212	0.681	765.2
11,000	60.4	15.8	0.05047	0.659	762.7
12,000	56.8	13.8	0.04887	0.639	760.0
13,000	53.2	11.8	0.04729	0.618	757.4
14,000	49.6	9.8	0.04575	0.598	754.6
15,000	46.0	7.8	0.04427	0.579	751.9
16,000	42.4	5.8	0.04282	0.560	749.2
17,000	38.8	3.8	0.04140	0.541	746.6
18,000	35.2	1.8	0.04002	0.523	743.9
19,000	31.6	- 0.2	0.03867	0.505	741.1
20,000	28.0	- 2.2	0.03738	0.489	738.5
21,000	24.4	- 4.2	0.03609	0.472	735.8
22,000	20.8	- 6.2	0.03487	0.456	733.1
23,000	17.2	- 8.2	0.03365	0.440	730.3
24,000	13.6	-10.2	0.03249	0.424	727.6
25,000	10.0	-12.2	0.0313	0.410	724.8
26,000	6.4	-14.2	0.0302	0.395	721.9
27,000	2.8	-16.2	0.0291	0.381	719.1
28,000	- 0.8	-18.2	0.0281	0.367	716.3
29,000	- 4.4	-20.2	0.0271	0.354	713.6
30,000	- 8.0	-22.2	0.0261	0.341	710.7
31,000	-11.6	-24.2	0.0251	0.328	707.8
32,000	-15.2	-26.2	0.0242	0.316	705.0
33,000	-18.8	-28.2	0.0233	0.304	702.2
34,000	-22.4	-30.2	0.0224	0.292	699.3
35,000	-26.0	-32.2	0.0225	0.281	696.4
36,000	-29.6	-34.2	0.0207	0.270	693.5
37,000	-33.2	-36.2	0.0199	0.260	690.6
38,000	-36.8	-38.2	0.0191	0.250	687.6
39,000	-40.4	-40.2	0.0184	0.240	684.6
40,000	-44.0	-42.2	0.0177	0.231	681.8

Table A-I-3 (continued)

Altitude, feet	Temperature °F	Temperature °C	Density, pound per cubic foot	σ, density divided by N.A.C.A. sea level standard	Velocity of sound, miles per hour
41,000	-47.6	-44.2	0.0170	0.222	678.9
42,000	-51.2	-46.2	0.0163	0.214	675.9
43,000	-54.8	-48.2	0.0157	0.206	672.9
44,000	-58.4	-50.2	0.0151	0.198	669.9
45,000	-62.0	-52.2	0.0145	0.190	666.9
46,000	-65.6	-54.2	0.0140	0.183	663.9
46,390	-67.0	-55.0	0.0137	0.180	662.1
47,000	-67.0	-55.0	0.0134	0.175	662.1
48,000	-67.0	-55.0	0.0128	0.167	662.1
49,000	-67.0	-55.0	0.0122	0.159	662.1
50,000	-67.0	-55.0	0.0116	0.152	662.1
55,000	-67.0	-55.0	0.00914	0.119	662.1
60,000	-67.0	-55.0	0.00721	0.094	662.1
65,000	-67.0	-55.0	0.00566	0.074	662.1

Source of data: "High-Altitude Cooling. I - Resume of the Cooling Problem," by Abe Silverstein, N.A.C.A. L-771, originally issued September, 1944.

Density of Moist Air

Air generally available to cool electronic equipment is not dry air, but a mixture of dry air and water vapor. This mixture is generally referred to as moist air. Although for most thermal evaluation work it is entirely satisfactory to calculate air density on the basis of dry air alone, the effect of the presence of water vapor in the air should be taken into account when a more precise value of the air density is required. In general, the simplest procedure by which the density of moist air may be determined is by measurement of the so-called dry- and wet-bulb temperatures. These temperatures may be measured with a sling psychrometer which consists of two ordinary mercury-stem glass thermometers mounted on a plate. The plate is swiveled at one end and may be revolved rapidly. Thus, the bulbs of the thermometers, being located at the extreme end of the plate opposite to the swivel, are made to move through the air at appreciable speed. The bulb of one of the two thermometers is covered with a soft water-absorbent wick which must be saturated with water before being revolved. This thermometer is the wet-bulb thermometer and indicates the wet-bulb temperature which, unless the air is saturated, is usually lower than the actual air temperature since it is affected by the evaporation of water from the wick. The other thermometer, having no evaporation of water surrounding the mercury bulb during the revolving process, indicates the actual temperature of the air, referred to as the dry-bulb temperature. The instrument should be revolved continuously until no further drop in the wet-bulb temperature is observed. The usual precautions against possible radiation effects on the temperatures indicated should be maintained.

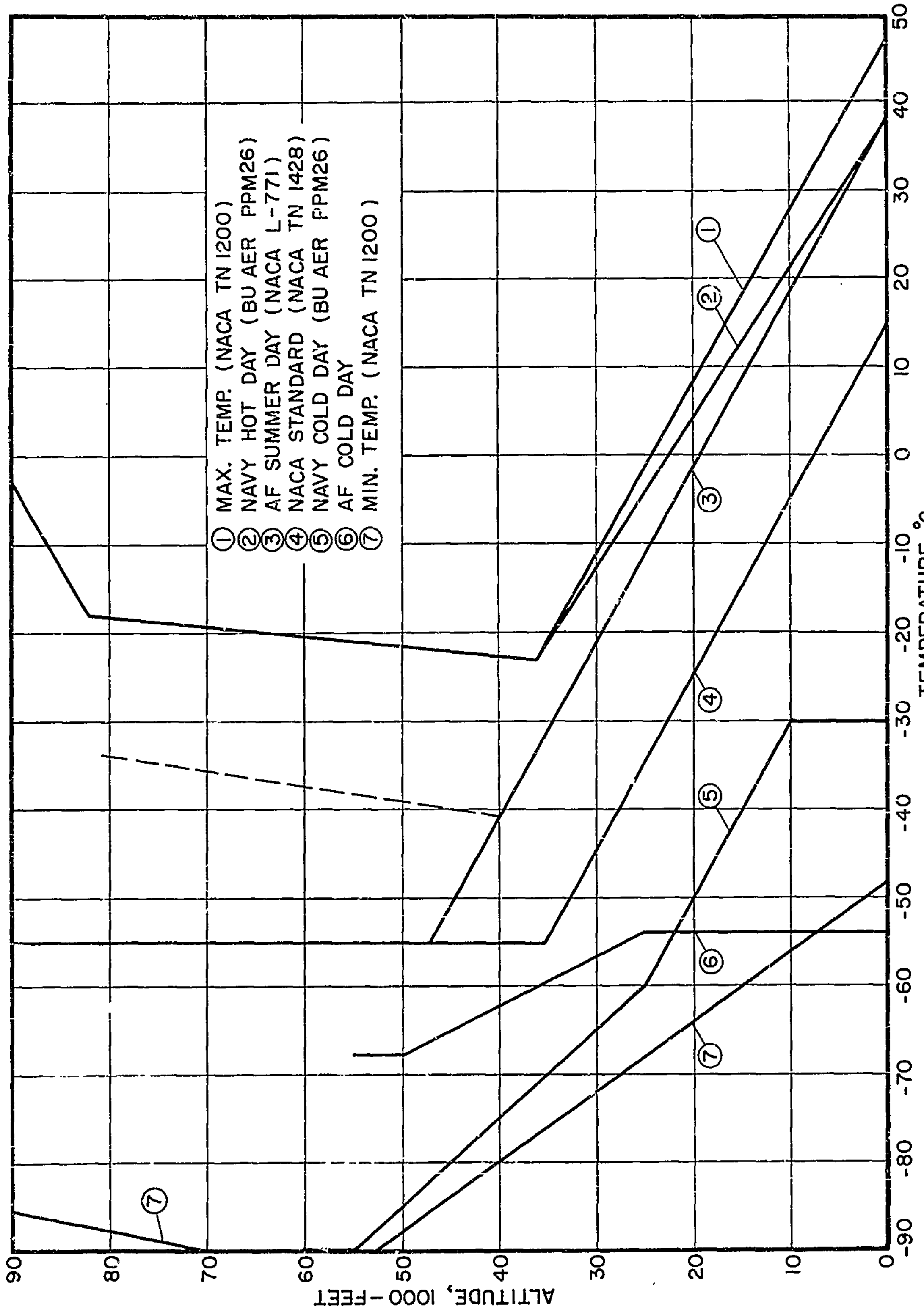


Figure A-I-3. Various Temperature-Altitude Standards

Once the wet- and dry-bulb temperatures have been determined it is possible to calculate the partial pressure of the water vapor and, thereafter, the density of the moist air. The partial pressure of the water vapor p_{wv}, expressed in inches of mercury absolute, may be calculated by use of the following simplified form of Carrier's equation*,

$$p_{wv} = p_{ww} - \left[p_{bar} \left(t_{db} - t_{wb} \right) \right] / 1500 \qquad \text{(A-I-5)}$$

where t_{db} and t_{wb} are the dry- and wet-bulb temperatures in °C, respectively, and p_{ww} is the saturation pressure of the water vapor at the wet-bulb tem-

Table A-I-4. Saturation Pressure of Water Vapor

Temp. °C	Saturation pressure, inch mercury	Temp. °C	Saturation pressure, inches mercury	Temp. °C	Saturation pressure, inches mercury
0	0.180	17	0.572	34	1.571
1	0.194	18	0.609	35	1.661
2	0.208	19	0.649	36	1.755
3	0.224	20	0.690	37	1.848
4	0.240	21	0.734	38	1.957
5	0.257	22	0.781	39	2.065
6	0.276	23	0.829	40	2.179
7	0.296	24	0.881	41	2.298
8	0.317	25	0.935	42	2.422
9	0.339	26	0.993	43	2.553
10	0.362	27	1.053	44	2.689
11	0.387	28	1.116	45	2.831
12	0.414	29	1.183	46	2.979
13	0.442	30	1.253	47	3.135
14	0.472	31	1.327	48	3.298
15	0.503	32	1.404	49	3.467
16	0.536	33	1.486	50	3.644

*See, for example, Heating, Ventilating and Air Conditioning Fundamentals, Severns and Fellows, John Wiley and Sons, 1950.

perature in inches of mercury absolute. It is apparent from equation (A-I-5) that when the wet- and dry-bulb temperatures are alike the partial pressure of the water vapor is equal to the saturation pressure. Then the air is said to be saturated. Table A-I-4 gives the saturation pressure of water vapor for the range of temperatures likely to be encountered during bench test work.

With the partial pressure of the water vapor determined by equation (A-I-5), the density of moist air ρ_{a-wv} may be calculated directly from the equation of state

$$\rho_{a-wv} = 0.738\,(p_{bar} - 0.378\,p_{wv})/T_{db}, \qquad \text{(A-I-6)}$$

where the density ρ_{a-wv} is in pounds per cubic foot, the absolute dry-bulb temperature T_{db} is in ^{o}K, and the pressures are in inches of mercury absolute.

The wet- and dry-bulb temperatures also serve to define humidity which is desirable to record for most tests. By definition

$$\text{per cent humidity} = 100\,(p_{wv}/p_{wd}) \qquad \text{(A-I-7)}$$

Here p_{wd} is the saturation pressure of water vapor at the indicated dry-bulb temperature and may be read directly from Table A-I-4.

Example A-I-2. Calculation of Density of Moist Air

During the bench test of an electronic equipment the dry- and wet-bulb temperatures were found to be 25° and 22°C, respectively. The barometer indicated a pressure of 29.0 inches mercury. Determine the density and per cent humidity of the moist air.

From Table A-I-4

$$p_{ww} = 0.781 \text{ inch mercury absolute}$$

$$p_{wd} = 0.935 \text{ inch mercury absolute}$$

By equation (A-I-5),

$$p_{wv} = 0.781 - 29.0\,(25 - 22)/1500 = 0.723 \text{ inch mercury absolute}$$

Hence, by equation (A-I-6),

$$\rho_{a-wv} = 0.738\,(29.0 - 0.378 \times 0.723)/(273 + 24) = 0.0711 \text{ pound per cubic foot}$$

By equation (A-I-7),

$$\text{per cent humidity} = 100 \times 0.723/0.935 = 77.3$$

APPENDIX II

BASIC FLOW AND ENERGY RELATIONSHIPS

A number of basic thermodynamic and fluid dynamic relationships are necessary for use in the thermal evaluation of electronic equipment. In particular, it is necessary to be able to calculate air density, volume or weight rate of flow, air temperature rise resulting from heat dissipation from components or equipment cases, blower power, etc. The purpose of this appendix is to present the basic equations and illustrate their application to different typical flow processes involved in the air cooling of electronic equipment.

Flow Equation (Conservation of Mass)

When a fluid flows during a steady-state process, i.e., a process having the flow variables invariant with time, through any type of flow channel, the relation between the weight rate of fluid flow W, the flow area A', the mean density of the fluid ρ, and the mean flow velocity u is expressed by the so-called equation of continuity

$$W = \rho A' u \qquad \text{(A-II-1)}$$

where W has the units of pounds per second when ρ, A', and u are expressed in pounds per cubic foot, square feet, and feet per second, respectively. The mean velocity u corresponds to the effective velocity by which the fluid is transported across any section of a flow channel and the area A' is the available flow area taken as the area transverse to the direction of flow.

The volume rate of flow Q" at any transverse section along a flow system is given by

$$Q'' = A' u \qquad \text{(A-II-2)}$$

where Q" has the units of cubic feet per second when the area A' and the mean velocity u are used in square feet and feet per second, respectively. It is important to observe that for any closed flow system, i.e., one having no intermediate fluid sources or sinks, the weight rate of fluid flow W is the same for all sections along the flow system, while the volume rate of flow Q" may and generally does vary. Hence, it is always necessary to calculate on the basis of constancy of weight flow, unless, of course, the density of the fluid happens to remain constant.

The relative variation in the mean flow velocity u may be established in terms of a reference velocity and the associated area and density ratios. The equation describing this relative variation is

$$u_n/u_1 = (A_1'/A_n')(\rho_1/\rho_n) \qquad \text{(A-II-3)}$$

Energy Equation and Heat Balances

An equation of great help in the thermal evaluation of electronic equipment is the so-called general energy equation. This equation, based on the principle of conservation of energy, describes a heat balance between energy sources and sinks; the sources in the case of electronic equipment being the differences between the electrical inputs and outputs throughout the circuits and the sinks any medium or structure to which this input energy is dissipated, whether by conduction, radiation, or forced and free convection. For steady-state operation of the equipment an over-all heat balance based on the general energy equation is a statement to the effect that the thermal equivalent of the electrical input to the equipment minus the electrical output is equal to the heat dissipated from the equipment by any or all modes of heat transfer. However, it is frequently desirable and necessary to break up the over-all heat balance to allow separate evaluation of heat dissipation to the various sinks, rather than the total to all sinks. For example, it is usually desirable to evaluate separately heat dissipated to cooling air passing through an equipment, and heat dissipated from the equipment's case surface directly to the environment. This type of heat balance necessitates a breakdown of the total energy into the various energy forms describing the physical process in terms of the physical variables of the system.

The purpose of this section is to present the general energy equation applicable to steady-flow processes involved in thermal evaluation work and to illustrate its application to various flow processes likely to be encountered.

1. Forms of Energy

An understanding of only a few of the many possible forms or types of energy is needed for thermal evaluation of electronic equipment. These are:

a. Kinetic Energy. The energy of translation of a moving stream of fluid is $u^2/2g$ foot-pounds per pound. For fluid flow at the rate of W pounds per second, the kinetic energy of the stream is $Wu^2/2g$ foot-pounds per second. The definition of kinetic energy by $Wu^2/2g$ is, strictly speaking, valid for a uniform or an approximately uniform velocity profile only. When the velocity varies to an appreciable extent over any transverse section of the flow channel, it is necessary to evaluate kinetic energy per second by

$$(\rho/2g) \int_{A'} u^3 \, dA'$$

which in most situations may best be calculated by sub-division of the flow cross-section in small equal areas over which the velocity would be assumed constant. With this procedure the kinetic energy per second of the stream is evaluated by

$$(\rho/2g) \sum_n u_n^3 A_n'$$

where u_n represents the mean velocity throughout the area subdivision A_n. The above expressions only account for variation in flow velocity, not including variation in density due to temperature or pressure.

b. Potential Energy. The energy of fluid by virtue of position, or elevation. Since only the change in potential energy is of interest, the evaluation may be based upon an arbitrarily chosen datum plane. Potential energy per unit weight of fluid is Z' foot-pounds per pound, or just Z' feet of fluid. For fluid flow at the rate of W pounds per second, the potential energy of the stream is WZ' foot-pounds per second. When dealing with gases it is permissible to neglect this change in energy, for any elevation change encountered in electronic equipment represents a negligible quantity of energy in comparison with other energy variations.

c. Flow Work or Pressure Head. The energy associated with the work required to displace or transport a fluid from one point to another within a flow system is generally referred to as flow work when dealing with compressible fluids and as pressure head when dealing with incompressible fluids. Flow work per unit weight of fluid is $70.7\ p/\rho$ foot-pounds per pound or feet of fluid, where p is the absolute pressure in inches mercury and ρ the density in pounds per cubic foot. For fluid flow at the rate of W pounds per second, flow work is $70.7\ W\ p/\rho$ foot-pounds per second.

d. Internal Energy. The energy inherent to the fluid and due to molecular activity and configuration. It is defined by such factors as translation, rotation, vibration of the molecules, and internal molecular forces governed by relative positions of molecules. It is necessary for use in evaluating temperature change of a fluid since temperature is related to the internal energy level of the fluid. Internal energy per unit weight of fluid is represented by Btu per pound. For W pounds per second of fluid, the internal energy of the stream is We Btu per second. When dealing with a perfect gas, i.e., a fluid obeying an equation such as (A-I-1) where $\rho \propto p/T$, the internal energy change per unit weight of fluid may be evaluated by the equation $\Delta e = \int_{T_1}^{T_2} c_v\, dT$, which for the case of constant specific heat c_v, reduces to the most usable form, $\Delta e = c_v\, \Delta T$.

e. Heat. Heat is energy in transition. Energy is not heat unless it is in the process of being transferred from one point to another. This form of energy is used to describe the magnitude or rate at which energy is added to or taken from a thermodynamic medium. As for example, the energy dissipated from an electronic component is energy in transition, which is heat. By convention, heat being added to a medium is dealt with algebraically as positive and heat withdrawn as negative. In the general energy equation this form of energy is represented by the symbol $\overline{q}$ with the units of Btu per second. In many instances there will exist a simultaneous heat gain to and a heat loss from the system, in which case it must be recognized that $\overline{q}$ in the general energy equation represents the net rate of heat transfer to or from the system. Algebraically,

$$\overline{q}_{net} = \sum \overline{q}$$

f. Mechanical Work. The energy added to or taken from a fluid by any mechanical action, such as a blower, turbine, etc. Like heat, work is energy in transition. It is represented by $\overline{wk}$ in Btu per second. By convention, work done by a system is considered positive and work done on a system negative. Where both work by and on a system occur simultaneously the term $\overline{wk}$ represents the time rate of the net work involved, which may be by or on the system.

2. General Energy Equation for Compressible Fluids

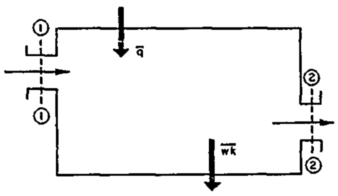


Figure A-II-1.
General Flow System

Using the forms of energy discussed in the previous section a statement of the principle of conservation of energy may be made in the form of the general energy equation. Consider, for example, the general system illustrated schematically in Figure A-II-1. A fluid flows through the system at a steady rate of W pounds per second. It shall be assumed, arbitrarily, that the net heat transfer rate $\overline{q}$ is to the system, with the net work $\overline{wk}$ done by the system. At entrance to the system, section (1-1), the fluid may possess energy in the form of potential, kinetic, internal, and flow. The same situation exists at the exit of the system, section (2-2). We have then,

$$\text{energy entering system per unit time} = \frac{WZ_1'}{J} + \frac{70.7\ Wp_1}{J\rho_1} + We_1 + \frac{Wu_1^2}{2gJ} + \overline{q}$$

$$\text{energy leaving system per unit time} = \frac{WZ_2'}{J} + \frac{70.7\ Wp_2}{J\rho_2} + We_2 + \frac{Wu_2^2}{2gJ} + \overline{wk}$$

The term J, the mechanical equivalent of heat, equal to 778 foot-pounds per Btu, is introduced to convert the units of all terms into Btu per second. With the system in thermal equilibrium, the energy entering equals that leaving and the general energy equation for a steady-state process becomes

$$\frac{WZ_1'}{J} + \frac{70.7Wp_1}{J\rho_1} + We_1 + \frac{Wu_1^2}{2gJ} + \overline{q} = \frac{WZ_2'}{J} + \frac{70.7Wp_2}{J\rho_2} + We_2 + \frac{Wu_2^2}{2gJ} + \overline{wk} \qquad \text{(A-II-4)}$$

As has been previously mentioned, when dealing with gases the effect of any change in elevation $(Z_2' - Z_1')$ likely to be encountered in this type of work represents a negligible quantity of energy, so that equation (A-II-4) may be used in the form

$$\frac{70.7Wp_1}{J\rho_1} + We_1 + \frac{Wu_1^2}{2gJ} + \overline{q} = \frac{70.7Wp_2}{J\rho_2} + We_2 + \frac{Wu_2^2}{2gJ} + \overline{wk} \qquad \text{(A-II-5)}$$

For thermal evaluation of electronic equipment where air is the thermodynamic medium it is possible to show that equation (A-II-5) is, with very good accuracy, equivalent to

$$q = 456\, W\, (t_2 - t_1) + wk \qquad \text{(A-II-6)}$$

where q and wk are in watts and the temperatures t in °C. Temperature t_2 would be indicated by a thermocouple at the system exit and temperature t_1 by a thermocouple at the system entrance. The kinetic energy terms in equation (A-II-5) are not included in equation (A-II-6) since when a thermocouple is inserted in a moving stream the fluid in the immediate vicinity of the bead is stagnated and the kinetic energy is nearly converted into an equivalent temperature rise which is indicated by the thermocouple, in addition to the actual temperature of the stream. Hence, a thermocouple reading is very nearly equivalent to an indication of the internal energy, flow energy, and kinetic energy of the stream. The indicated temperature for unshielded thermocouples, assuming radiation effects absent, will be within one per cent of the correct temperature to be used in equation (A-II-6) whenever the flow velocity is less than about 400 feet per second. With shielded thermocouple probes containing a stagnation chamber the error can be less than 1/4 per cent even for flow velocities of 500 to 600 feet per second. Thus, for ordinary thermal evaluation work, where the flow velocities are generally less than 100 feet per second, the error involved by use of equation (A-II-6) is of no significance. The specific heat at constant pressure c_p of air is assumed constant and equal to 0.432 Btu per pound-°C. This value is used in evaluating the constant of 456 in equation (A-II-6). Where fairly wide ranges in temperature are encountered and need for more accurate evaluation exists, the variation in specific heat c_p should be taken into account. Data on specific heat at constant pressure are given in Table A-I-1 and Figure A-I-1. The energy equation (A-II-6) is valid regardless of the amount of frictional loss present in the flow system.

3. General Energy Equation for Incompressible Fluids

The general energy equation (A-II-4) is applicable to incompressible fluids as well, i.e., those having constant density, providing the following general exception is followed. With an incompressible fluid the internal energy e and the transferred heat q are not available for the production of any effect other than a temperature change of the fluid. The change in energy so encountered is not available for overcoming the system's resistance and promoting flow. Hence, in reality equation (A-II-4) reduces to two separate energy equations, which are

$$\overline{q} + Wh_I/J = W(e_2 - e_1)$$

and

$$\frac{WZ_1'}{J} + \frac{70.7Wp_1}{J\rho} + \frac{Wu_1^2}{2gJ} = \frac{WZ_2'}{J} + \frac{70.7Wp_2}{J\rho} + \frac{Wu_2^2}{2gJ} + \overline{wk} + \frac{Wh_L}{J}, \qquad \text{(A-II-7)}$$

where h_L represents the frictional loss in energy during flow per unit weight of fluid, expressed in foot-pounds per pound which is equivalent to feet of fluid. The first of these two equations has no practical value in thermal evaluation work. The second, equation (A-II-7) is quite important and assumes its most convenient form by dividing out the terms W and J, so that all terms in the equation have the units of feet of fluid. This form is

$$Z_1' + 70.7p_1/\rho + u_1^2/2g = Z_2' + 70.7p_2/\rho + u_2^2/2g + J\overline{wk}/W + h_L \qquad \text{(A-II-8)}$$

When mechanical work and energy loss are absent the above equation reduces to a statement of the Bernoulli principle that the sum of the elevation, pressure, and velocity heads are constant throughout a flow process.

4. Total Pressure and Velocity Pressure. Incompressible Flow

The total pressure of a fluid is defined as that pressure attained by the fluid when it is stagnated without loss, external work, or change in elevation. Thus, the stream undergoes a process of diffusion wherein the kinetic energy of the stream is converted into pressure. The difference between the total pressure and the static pressure at any one point in the flow process is called the velocity pressure. Applying equation (A-II-8) to this process gives

$$70.7\ p/\rho + u^2/2g = 70.7\ P/\rho \qquad \text{(A-II-9)}$$

where P represents the absolute total pressure of the fluid in inches of mercury, and, as may be seen, the total pressure head $70.7\ P/\rho$ equals the sum of the static pressure head $70.7\ p/\rho$ and the velocity head $u^2/2g$.

A convenient rearrangement of equation (A-II-9) is

$$P = p + \rho u^2/4550 \qquad \text{(A-II-10)}$$

where all terms have the units of inches of mercury. Hence, the velocity pressure of an incompressible fluid is given by

$$p_v = \rho u^2/4550 \text{ inches mercury} \qquad \text{(A-II-11)}$$

or

$$p_v = \rho u^2/336 \text{ inches water} \qquad \text{(A-II-12)}$$

In all equations ρ represents the density in pounds per cubic foot, of the fluid, possessing the velocity of u feet per second.

5. Determination of Flow Velocity

The above equations illustrate the basic theory by which a combination of total pressure tube and a piezometer ring or a Pitot-static tube is used to determine flow velocity. If an open-end tube is faced into the direction of flow and connected on the other end to a manometer, or some other pressure-indicating device, a total pressure will be registered. On the other hand, if the open end of this tube is closed and a hole without burr is made on the cylindrical portion of its surface, a small distance downstream from the end, a static pressure will be indicated on the pressure-measuring instrument. This indication would be approximately the same as obtained by a tap in the wall of the flow passage. The standard Pitot-static tube of the National Association of Fan Manufacturers and the American Society of Heating and Ventilating Engineers is shown in Figure A-II-2.

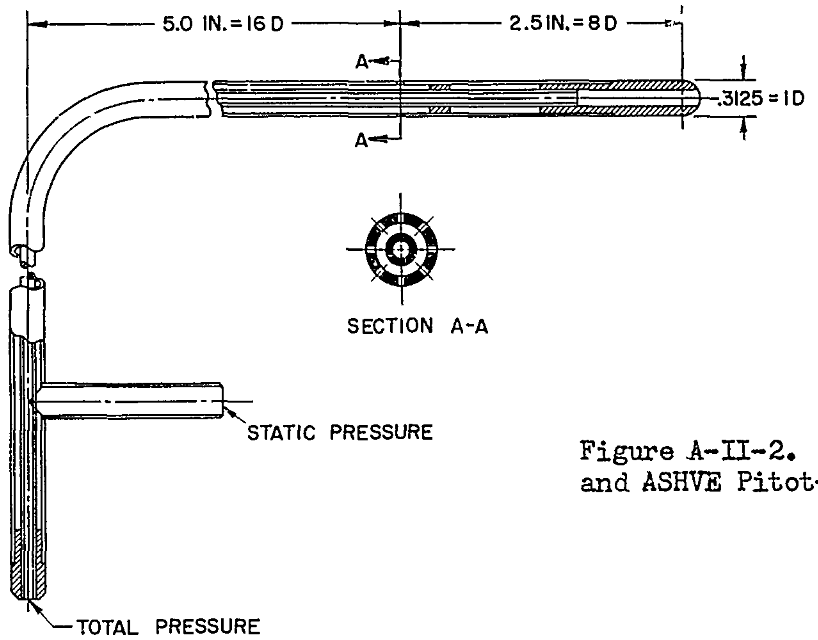


Figure A-II-2. Standard NAFM and ASHVE Pitot-Static Tube

The use of an impact tube and wall taps, as described on page 45, may in many instances be more desirable because of size and complication of manufacture. Impact tubes may be constructed with appreciably smaller diameters than Pitot-static tubes. Consequently, they occupy a smaller percentage of the total cross-sectional area of the flow passage and do not significantly alter the flow velocity at the section of measurement. Pitot-static tubes should not be employed when their cross-sectional area occupies more than 5 per cent of the cross-sectional area of the flow passage.

Equations (A-II-11 and -12) are valid as long as the density ρ does not vary during the stagnation process. With air, or any other compressible fluid stagnation does produce a change in the density as a result of the change in pressure from static to total. Hence, if the flow velocity is sufficiently high that an appreciable change in pressure occurs, the incompressible flow relations no longer apply. However, because of its simplicity the incompressible flow equation is used as long as no serious error would result. For the range of conditions commonly encountered in electronic equipment, the velocity of air so determined by the incompressible flow relation will be correct within 1/2 per cent for any flow velocity below approximately 200 feet per second. At a flow velocity of 200 feet per second and standard sea-level density the velocity pressure of air is roughly 9 inches of water.

Example A-II-1. Determination of Flow Velocity from Total and Static Pressure Measurement

A Pitot-static tube inserted in a moving stream of air indicates a total gage pressure of 20.0 inches water and a static gage pressure of 16.8 inches water. The temperature of the stream is 40°C and the barometric pressure 29.5 inches mercury. Determine the flow velocity at this point.

By equation (A-I-2)

$$p = 29.5 + 0.0738(16.8) = 30.74 \text{ inches mercury absolute}$$

By equation (A-I-1)

$$\rho = 0.738(30.74)/(273 + 40) = 0.0725 \text{ pound per cubic foot}$$

$$p_v = P_g - p_g = 20.0 - 16.8 = 3.2 \text{ inches water}$$

By equation (A-II-12)

$$3.2 = 0.0724\, u^2/336$$

$$u = 122 \text{ feet per second}$$

The error introduced by use of the incompressible flow equation (A-II-12) is roughly 0.14 per cent.

6. Determination of Flow Rate in Passage

When dealing with flow through channels of fairly small cross-sectional area and passages presenting rather tortuous paths for the fluid to follow, the flow character is generally non-uniform to the extent that measurement of velocity and temperature at any one point in one cross-section of the flow passage is insufficient to define the weight or volume rate of fluid flow actually taking place. In these instances it becomes necessary to probe the stream to determine the flow properties at many points in any cross-section of the channel. The procedure is to subdivide the cross-section of the channel into equal areas, with each area being sufficiently small that the velocity and temperature can be considered uniform in it. The total volume flow is then obtained by adding the volume flows of all area subdivisions. Hence, for flow conditions of non-uniform velocity distribution the volume rate of flow is

$$Q'' = \sum A_n' u_n = A_n' \sum u_n, \qquad \text{(A-II-13)}$$

when the area subdivisions are made equal. This is equivalent to integrating the velocity profile determined by probing the streams. It is apparent from equation (A-II-13) that the mean flow velocity u is obtained by dividing the true volume flow Q by the duct area A, or by averaging the measured velocities u_n. The weight rate of flow W is obtained by summation of the products of local velocity u_n and local density ρ_n multiplied by the magnitude of a single area subdivision

$$W = A'_n \sum \rho_n u_n \qquad \text{(A-II-14)}$$

The variation in the density ρ_n is determined by measurement of temperature and static pressure and the density equation (A-I-1). When the fluid flows in but one general direction, the static pressure will be found to be nearly constant across any transverse section of the duct. Measurement of static pressure at two diametrically opposed points in a duct should suffice to define the average static pressure at any one section.

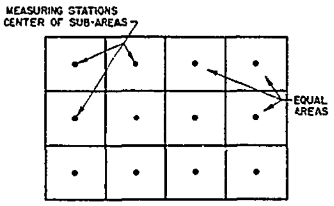


Figure A-II-3. Location of Measuring Stations in a Rectangular Duct

The method by which the flow area is subdivided depends upon the shape of the channel. For channels of rectangular cross-section the area may be divided into a series of small rectangles, with the measuring stations located at the center of each sub-area. Figure A-II-3 illustrates this method. For ducts of circular cross-section the conventional practice is to divide the total area by a series of concentric circles so located as to yield equal sub-areas. This resolves the flow in a group of parallel streams flowing through equal annular areas.

Table A-II-1. Values of r_n/r for Circular Ducts

Total number of subdivisions, r	Number of sub-area, n: 1	2	3	4	5	6	7	8	9	10
1	0.707									
2	0.500	0.866								
3	0.408	0.707	0.913							
4	0.354	0.612	0.791	0.936						
5	0.316	0.548	0.707	0.837	0.949					
6	0.289	0.500	0.645	0.764	0.866	0.958				
7	0.267	0.463	0.598	0.707	0.802	0.886	0.964			
8	0.250	0.433	0.559	0.662	0.750	0.829	0.902	0.968		
9	0.236	0.408	0.527	0.624	0.707	0.782	0.850	0.913	0.972	
10	0.224	0.387	0.500	0.592	0.671	0.742	0.806	0.866	0.922	0.975

The measuring positions should be located at the geometric mean radius or diameter of the sub-area. The equation for locating the measuring stations in a duct of circular cross-section is

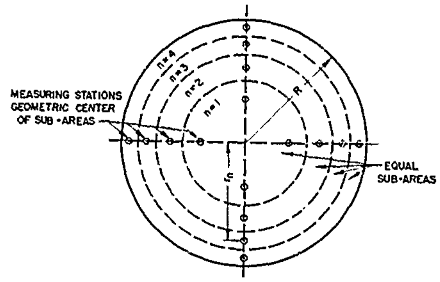


Figure A-II-4. Illustration of Area Subdivision in Circular Duct for Four Equal Sub-Areas

$$r_n/r = \sqrt{(2n - 1)/2y} \qquad \text{(A-II-15)}$$

where y represents the number of sub-areas into which the total area is equally divided, n represents the number of a particular sub-area counted from the center of the duct, (i.e., the central sub-area is number one), r the inside radius of the duct, and r_n the distance from the duct center to the geometrical mean position of the sub-area number n. Figure A-II-4 illustrates this method of area subdivision. Table A-II-1 gives values of r_n/r for various values of y and n.

Example A-II-2. Determination of Weight and Volume Flow Rates

A centrifugal blower attached to an electronic equipment is used to force air through the equipment to cool the components by forced convection heat transfer. A thermocouple and U-tube manometer installed at the 3-inch diameter blower discharge indicate the air temperature and static gage pressure at 25°C and 8 inches water, respectively. At the exit from the equipment the air has a temperature of 80°C, a pressure equal to atmospheric and it discharges through a 4-inch diameter duct. By use of a Pitot tube the mean velocity at the exit duct is found to be u_2 = 14 feet per second (the determination of flow velocity by this method is discussed on page 336). The barometric pressure is 29.7 inches of mercury. Determine by calculation (1) the volume rate of air flow at the equipment exit Q_2 and at the entrance Q_1, (2) the weight rate of air flow, W, and (3) the mean flow velocity at the blower discharge section, u_d.

At the exit:

p_2 = 29.7 inches mercury absolute

$T_2 = 273 + 80 = 353°K$

$\rho_2 = 0.738 \times 29.7/353 = 0.0621$ pound per cubic foot

$A_2' = (\pi/4)(4/12)^2 = 0.0873$ square foot

u_2 = 14 feet per second

$Q = 0.0873 \times 14 \times 60 = 73.2$ cubic feet per minute

$W = 1.22 \times 0.0621 = 0.0758$ pound per second

At the entrance

$p_1 = p_d = 29.70 + 0.0738(8) = 30.29$ inches mercury absolute

$T_1 = 273 + 25 = 298°K$

$$\rho_1 = 0.738 \times 30.29/298 = 0.0750 \text{ pound per cubic foot}$$

$$A_1' = (\pi/4)(3/12) = 0.0491 \text{ square foot}$$

$$Q_1 = W/\rho_1 = 0.0758 \times 60/0.0750 = 60.6 \text{ cubic feet per minute}$$

$$u_1 = Q_1/(60A_1') = 60.6/(60 \times 0.0491) = 20.6 \text{ feet per second}$$

Example A-II-3. Determination of Velocity Distribution, Mean Flow Velocity, Volume and Weight Flow by Method of Area Subdivision

This example illustrates the method of calculating flow conditions from data obtained by probing a flow channel. Air flows during a steady-state process through a 4-inch by 4-inch channel. The stream is probed with a Pitot-static tube and a thermocouple at the various measuring stations shown in Figure A-II-5 to determine the velocity pressures and temperatures. The measured velocity pressures in inches of water and measured temperatures in °C are given in Tables A-II-2 and A-II-3.

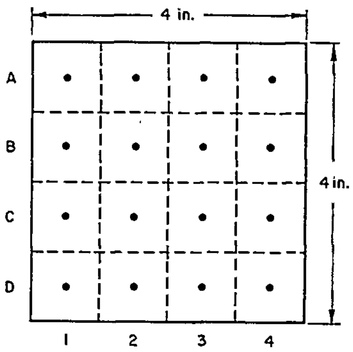


Figure A-II-5. Measuring Stations in Square Duct (Example A-II-3)

Static pressure taps located at four points around the periphery of the channel indicated static gage pressures of 2.05, 2.10, 2.08, and 2.09 inches of water. Because of this small variation, the static pressure of the stream is taken to be constant at all points throughout the transverse section and equal to the average of the measured values,

Table A-II-2. Measured Velocity Pressures (Example A-II-3)

Station	p_v, inches water			
	1	2	3	4
A	0.13	0.31	0.33	0.24
B	0.12	0.28	0.30	0.20
C	0.12	0.25	0.28	0.10
D	0.08	0.27	0.26	0.12

Table A-II-3. Measured Temperatures (Example A-II-3)

Station	t, °C			
	1	2	3	4
A	105	85	90	97
B	98	80	83	95
C	95	68	68	90
D	90	60	65	80

i.e., 2.08 inches water. The barometric pressure was 29.30 inches mercury. The absolute static pressure of the stream is, then, by equation (A-I-2)

$$p = 29.30 + 0.0738(2.08) = 29.45 \text{ inches mercury absolute}$$

a. Velocity Distribution

The velocity pressure expressed in inches of water is given by equation (A-II-12)

$$p_v = \rho u^2/336$$

Combining this equation with the density relationship (A-I-1) yields

$$p_v = pu^2/(455\ T)$$

Since the absolute static pressure p is constant and equal to 29.45 inches of mercury

$$p_v = 29.45\ u^2/(455\ T) = u^2/(15.46\ T)$$

or,

$$u = 3.93\sqrt{p_v T}$$

The velocity function, as shown by this equation, equals the square root of the product of the velocity pressure and absolute temperature of the stream. Thus, the relative magnitudes of the velocities at the various stations may be defined by comparison of this square root function. Also, if desired, the absolute magnitudes of the flow velocities may be calculated directly by this equation. The velocity function reduces to simply the square root of the velocity pressure when the temperature of the stream remains constant or nearly constant across the transverse section being probed. Table A-II-4 summarizes the measured velocities relative to the stream velocity at station B-3, and Table A-II-5 gives the absolute magnitude of the measured velocities.

Table A-II-4. Flow Velocities Relative to Value at Station B-3 (Example A-II-3)

Station	1	2	3	4
A	0.678	1.020	1.059	0.911
B	0.646	0.975	1.00	0.829
C	0.642	0.893	0.944	0.574
D	0.521	0.917	0.906	0.630

Table A-II-5. Absolute Values of Flow Velocities (Example A-II-3)

Station	u, feet per second			
	1	2	3	4
A	27.4	41.3	42.9	36.8
B	26.2	39.4	40.5	33.6
C	26.0	36.2	38.2	23.2
D	21.1	37.2	36.7	25.5

b. Calculation of Volume Flow Rate

The total rate of flow through the channel is obtained by summation of the flows through all subdivisions of flow area. Hence, by equation (A-II-13)

$$Q'' = A_n \sum u_n$$

since all subdivided areas have equal size. The subdivided area A_n is one inch square, or 0.00696 square feet. The velocity summation is obtained by adding all velocities given in Table A-II-5. Hence,

$$Q'' = 0.00696 \quad (27.4 + 41.3 + 42.9 + \ldots .)$$

Q'' = 3.70 cubic feet per second, or Q = 222 cubic feet per minute

The mean stream velocity or effective velocity by which the air is transported across this section is

$$u = Q''/A' = 3.70/(16/144) = 33.3 \text{ feet per second}$$

c. Calculation of Weight Flow Rate. The flow rate on a weight basis is given by equation (A-II-14).

$$W = A'_n \sum \rho_n u_n$$

But

$$\rho_n = 0.738(29.45)/T_n = 21.73/T_n$$

and

$$A'_n = 0.00696 \text{ square feet}$$

so that

$$W = 0.151 \sum u_n/T_n$$

Using the temperatures of Table A-II-3 and the velocities of Table A-II-5 gives

$$W = 0.151 \sum (27.4/378 + 41.3/358 + 42.9/363 + \ldots .)$$

$$W = 0.225 \text{ pound per second}$$

7. The Pitot-Static Equations for Compressible Flow

When measuring total and static pressures of a high-velocity stream of gas by use of the Pitot-static tube or a combination of total-pressure tube and wall taps, the density ρ in equation (A-II-10) is affected by the change in pressure during stagnation. For this reason the incompressible-flow relationship between velocity pressure and velocity, equation (A-II-12), is not correct and an equation taking into account this variation in the density must be employed. The correct equation for air over the temperature range encountered in electronic equipment is

$$u = 147.4 \sqrt{T \left[1 - (p/P)^{0.286}\right]} \qquad \text{(A-II-16)}$$

where T is the temperature in ^{o}K as measured by a thermocouple inserted in the air stream. For best accuracy, a shielded thermocouple embodying a stagnation chamber surrounding the bead should be used. In this relationship the total and static pressures must be on an absolute basis expressed in consistent units.

8. Heat Balance for Forced Convective Cooling with Air

With equipment cooled by a stream of air flowing in and around the components or over an equipment case the general energy equation describing an energy balance with the cooling air is given by equation (A-II-6) in the form

$$q = 456\, W(t_2 - t_1) \qquad \text{(A-II-17)}$$

since mechanical work effects are not present. In this equation q represents the heat dissipated to the flowing stream of air and is equal to the total component heat dissipation less any external heat loss of the equipment by any of the possible modes of heat transfer.

The temperatures used in equation (A-II-17) must represent the mean temperature of the stream, such that when either one is multiplied by the weight rate of flow W a true relative measure of the energy at entrance or exit is obtained. Usually the measurement of a single temperature near or at the center of the stream will be sufficiently accurate to define a mean value for the entrance to the equipment. However, at the exit of the equipment it is entirely possible and quite probable that complete mixing of the stream has not taken place, and the measurement of a single temperature at some point near or at the center of the stream may not give an accurate indication of the energy level or heat added to the air in passing through or over the equipment. When the degree of mixing and uniformity of flow is uncertain, it is best to provide a mixing or stagnation chamber at the equipment exit, or to probe the discharging air stream with a thermocouple and a Pitot-static tube. The mixing chamber should be insulated and baffled to eliminate heat loss and to insure adequate mixing. When using the mixing chamber, an auxiliary blower is required to make up the resistance to flow added by the chamber; otherwise, the air capacity of the cooling blower proper will decrease. For cases where this is not possible or where this added complexity of test apparatus is not warranted, the method of probing the stream should be used. As has been discussed on pages 337 to 339, the discharging stream should be divided into annular rings of equal area for circular ducts and equal rectangular areas for rectangular ducts. The flow velocity and temperature should be determined at the geometric center of all area elements and along several diameters or horizontal and vertical axes. After having probed the stream the mean temperature to be used in the heat balance equation (A-II-17) may be determined by the following summation process

$$T = (A_n'/W) \sum \rho_n u_n T_n \qquad \text{(A-II-18)}$$

where W is the total weight flow in pounds per second. A second method to find the mean stream temperature is to evaluate the mean stream density from equation (A-II-14), i.e.,

$$\rho = W/Q'' = (W/A_n') \sum u_n$$

and then to calculate the temperature from the density and absolute static pressure by the equation (A-I-1). For normal thermal evaluation work the second method generally requires less calculation work.

Example A-II-4. Determination of Mean Stream Temperature

Using the data of Example A-II-3, the mean stream density is

$$\rho = W/Q'' = 0.225/3.70 = 0.0609 \text{ pound per cubic foot}$$

Since,

$$p = 29.45 \text{ inches mercury absolute,}$$

$$T = 0.738 \times 29.45/0.0609 = 356.9^{\circ}K,$$

and the mean temperature defining the energy level of the stream is

$$t = 356.9 - 273 = 83.9^{\circ}C$$

This same value can be obtained by evaluation of equation (A-II-18).

9. Heat Balance for Blowers

Power or energy supplied to the input shaft of a blower is transferred to the air passing through the blower, the contacting structure, and the surrounding atmosphere. The energy equation describing a heat balance for a blower is the same as equation (A-II-6), i.e.,

$$q = 456\, W\, (t_d - t_i) + wk \qquad \text{(A-II-19)}$$

where q represents the heat loss in watts from the blower to the contacting structure and the surrounding atmosphere, W the air rate in pounds per second, t_d and t_i the temperature of the air leaving and entering the blower in $^{\circ}C$, respectively, and wk the rate of energy input to the blower in watts. With but rare exception, q is always negative. The work term wk is always negative since work is done on the air by the blower. Introducing these signs on wk and q gives for the temperature rise of the air across the blower

$$\Delta t_B = t_d - t_i = (wk - q)/(456\, W) \qquad \text{(A-II-20)}$$

The heat loss rate q is negligible in many blower installations in comparison to the work. Neglecting it entirely yields conservative results in the evaluation of the combined equipment performance since the calculated temperature of the air leaving the blower would be above its actual value. Then, neglecting heat loss,

$$\Delta t_B = wk/(456\ W) \qquad \text{(A-II-21)}$$

When the input power is expressed in horsepower, and heat loss is again neglected,

$$\Delta t_B = 1.636\ hp/W \qquad \text{(A-II-22)}$$

If the static pressure rise is not great, an equivalent form of equation (A-II-22) is

$$\Delta t_B = 0.0155\ \Delta p/(\eta_s \rho) \qquad \text{(A-II-23)}$$

where Δp is the static pressure rise of the air in inches of water and η_s is the static efficiency of the blower. Equation (A-II-23) is equivalent to equation (V-3) on page 86.

Example A-II-5. Heat Balance of Blower-Equipment Combination

During an altitude chamber test of a blower-cooled electronic equipment the following data were observed (refer to Figure A-II-6).

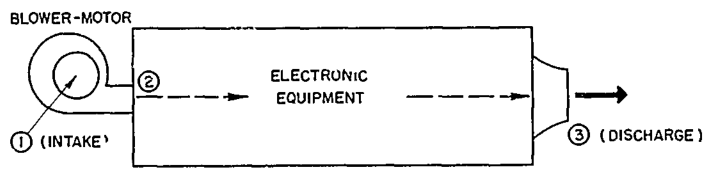


Figure A-II-6. Schematic Arrangement of Blower-Equipment Combination (Example A-II-5)

total pressure at blower inlet, $P_{g-i} = P_{g-1} = 0$ inch water gage

static gage pressure at blower discharge, $p_{g-d} = p_{g-2} = 6.1$ inches water

barometric pressure, $p_{bar} = 8.2$ inches mercury absolute

temperature at blower inlet, $t_i = t_1 = 10^\circ C$

temperature at equipment discharge, $t_3 = 60^\circ C$

In addition, it is known that the rate of heat generation within the equipment is 750 watts. The heat loss from the case by free convection and radiation has been calculated as 75 watts. It is required to determine from these data the air rate and input horsepower of the blower.

Assume dry air and a blower static efficiency of 40 per cent.

The static pressure rise across the blower is

$$\Delta p = p_{g-d} - P_{g-i} = 6.1 \text{ inches water}$$

$$\rho_1 = 0.738 \times 8.2/(273 + 10) = 0.0214 \text{ pound per cubic foot}$$

By equation (A-II-23)

$$\Delta t_B = (0.0155 \times 6.1)/(0.40 \times 0.0214) = 11^\circ C$$

Hence,

$$t_2 = 10 + 11 = 21^\circ C$$

The heat dissipation to the cooling air is

$$750 - 75 = 675 \text{ watts}$$

so that by equation (A-II-6)

$$675 = 456\, W(60 - 21)$$

$$W = 0.038 \text{ pound per second}$$

The volume flow at the blower intake is

$$Q_1 = 60\, W/\rho_1 = 60 \times 0.038/0.0214 = 106.5 \text{ cubic feet per minute}$$

Using equation (A-II-22)

$$hp = 0.038 \times 11/1.636 = 0.255$$

At this point the two assumptions employed in the analysis should be checked. The assumed blower efficiency should be verified from the calculated air rate and a known characteristic performance curve for the blower. Secondly, the assumption of zero heat loss from the blower should be checked by estimating the heat loss by radiation, conduction, and convection, so that a proper proportion of the blower input power is used when calculating the temperature rise of the air across the blower. Suppose, for example, it is estimated that 10 per cent of the blower power is dissipated to heat sinks other than the cooling air. Hence, only 90 per cent as much energy is given to the air passing through the blower and

$$\Delta t_B = 11 \times 0.9 = 9.9^\circ C$$

$$t_2 = 10 + 9.9 = 19.9^\circ C$$

Then,

$$675 = 456\, W(60 - 19.9)$$

$$W = 0.0369 \text{ pound per second}$$

$$Q = 103.3 \text{ cubic feet per minute}$$

Thus, the 10 per cent correction on the blower air temperature rise results in about 3 per cent decrease in air rate. The error introduced by neglecting this heat loss depends mostly on the magnitude of the equipment air temperature rise, and may in some instances affect the accuracy of performance calculations.

APPENDIX III

MANOMETRIC RELATIONSHIPS

A manometer is an instrument used for measuring pressures in general. Its basic theory rests upon the fact that head and pressure are synonymous, so that a column of fluid may be used as a direct indication of pressure. In general, manometers are used for measuring pressures within a range defined by a lower limit of accuracy and an upper limit of convenience. For measuring very small pressures, the error in reading the manometer, mostly due to the fluid meniscus, may represent a large percentage of the pressure registered and, thereby, render the instrument too inaccurate for use. When measuring very high pressures the required height of the column of fluid may be so great that the use of a manometer is not practical. With but rare exception, the ranges of pressure encountered in thermal evaluation of electronic equipment are such that manometers may always be used.

Specific Gravity and Conversion Factors

The most useful manometer fluids are water, mercury, red oil, alcohol, and carbon tetrachloride. Table A-III-1 lists the specific gravity and density of water for the range of ambient temperatures likely to be encountered with bench testing. Table A-III-2 gives the specific gravity of a number of manometric fluids.

Table A-III-1. Specific Gravity and Density of Water at Sea-Level Atmospheric Pressure

Temp., °C	Specific gravity	Density, pounds per cubic foot	Temp., °C	Specific gravity	Density, pounds per cubic foot
4	1.00000	62.427	26	0.99681	62.228
10	0.99973	62.410	28	0.99626	62.193
12	0.99952	62.397	30	0.99567	62.157
14	0.99927	62.381	32	0.99505	62.118
16	0.99897	62.362	34	0.99440	62.077
18	0.99862	62.341	36	0.99371	62.034
20	0.99823	62.316	38	0.99299	61.989
22	0.99780	62.289	40	0.99224	61.943
24	0.99732	62.260			

Reference: Mechanical Engineers' Handbook, L. S. Marks, fifth edition, McGraw-Hill Book Co., 1951.

Table A-III-2. Specific Gravities of Manometer Fluids Based on Water at 4°C

Liquid	Specific gravity at 21°C	Specific gravity at 0°C
Water	0.9980	0.9998
Mercury	13.543	13.595
Red oil (approx.)	0.820	0.835
Wood alcohol	0.790	0.810
Grain alcohol	0.789	0.805
Carbon tetrachloride	1.594 (20°C)	

Manometric Equation

The basic equation relating pressure change to an equivalent column of an incompressible fluid is

$$\Delta p' = \Delta Z'_{fl}\, \rho_{fl} \qquad \text{(A-III-1)}$$

where $\Delta p'$ represents the change in pressure in pounds per square foot, ρ_{fl} the density of the manometric fluid and $\Delta Z'_{fl}$ the change in elevation of the fluid producing or supporting the pressure differential $\Delta p'$. This differential elevation $\Delta Z'$ is commonly referred to as the head in feet, or $\Delta Z' \times 12 = \Delta Z$ is the head in inches of the manometer fluid. Equation (A-III-1) applies only to incompressible fluids, such as water, mercury, etc., used in manometers. The application of this basic equation to various types of manometers is illustrated in the following paragraphs.

1. Barometer

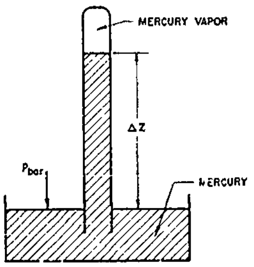


Figure A-III-1. Simple Barometer

A barometer may be constructed by filling with mercury a glass tube sealed at one end, and inverting the tube into an open vessel. This is illustrated in Figure A-III-1. The mercury level will fall in the tube until the pressure generated by the column of mercury equals the atmospheric pressure acting on the exposed surface of mercury. In the space above the mercury column at the sealed end of the tube the pressure is very nearly zero absolute, since at normal temperatures the vapor pressure of mercury is extremely low (about 0.00005 inches of mercury at 20°C). Hence, by application of equation (A-III-1)

$$\Delta p' = P'_{bar} - P'_{mercury\ vapor} \approx P'_{bar} = \Delta Z' \rho_{mercury} \qquad \text{(A-III-2)}$$

where $\Delta Z'$ represents the height of the mercury column in feet above the level in the vessel, as indicated in Figure A-III-1. It is apparent, then, that measurement of the height of the mercury column allows direct evaluation of the barometric pressure since the barometric pressure in inches of mercury p_{bar} would be identical with ΔZ in inches.

2. U-Tube Manometer

A very simple and commonly employed manometer is the so-called U-tube manometer, formed by a glass tube bent in a U shape and filled with a manometer fluid. An illustration of this instrument is given in Figure III-12. The well-type single tube manometer is shown in Figure III-13. Figure A-III-2 illustrates the application of this type of manometer to the determination of pressure level in a sealed equipment. One end of the U-tube is connected to the equipment with the other end vented to the atmosphere.

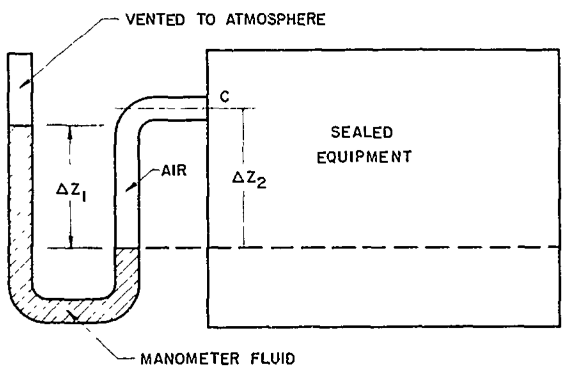


Figure A-III-2. U-Tube Manometer for Determination of Gage Pressure in Closed Container

Application of equation (A-III-1) to this manometer yields

$$\Delta p' = p'_{equip} - p'_{bar} = p'_{g-equip} = \Delta Z' \rho_{fl-1} \qquad \text{(A-III-3)}$$

Suppose, for example, the differential height indicated by the manometer is 20 inches water. This means then that the gage pressure inside the equipment is

$$p'_{g-equip} = (20/12)(62.3) = 103.8 \text{ pounds per square foot}$$

or

$$103.8/144 = 0.721 \text{ pounds per square inch.}$$

Or,

$$p_{g-equip} = 20 \text{ inches water} \approx 1.476 \text{ inches mercury}$$

With air inside the equipment or container the gage pressure calculated by the above method may be taken as the gage pressure anywhere within the equipment but with any liquid as the internal atmosphere the calculated gage pressure is the pressure at the level corresponding to the level of the lower leg of the manometer fluid, shown by the dashed line in Figure A-III-2. Calculation of the pressure at the manometer-equipment connection C requires correction for the column of fluid of height ΔZ_2 between this point and the lower manometer fluid level. This correction is made by equation (A-III-1), but creates a pressure opposing that generated by the column of fluid. If the liquid within the equipment has the density ρ_{fl-2}, as distinguished from the manometer fluid density ρ_{fl-1}, the gage pressure at the manometer connection in pounds per square foot is

$$\Delta p_C' = \Delta Z_1' \rho_{fl-1} - \Delta Z_2' \rho_{fl-2}, \qquad \text{(A-III-4)}$$

which may be expressed in inches of mercury, using a mercury tube manometer, as

$$p_{g(mercury)} = \Delta Z_1 - \Delta Z_2 (\rho_{fl-2}/\rho_{mercury}) \qquad \text{(A-III-5)}$$

As mentioned above, if the internal equipment atmosphere is air $p_g \approx \Delta Z$, since the ratio $(\rho_{fl-2}/\rho_{mercury})$ would be less than 0.001 for any pressure conceivably applicable to electronic equipment.

3. Manometers for Pitot-Static Tube

Measurement of flow velocity by use of the Pitot-static tube or an impact tube with a piezometer ring requires measurement of at least two of the three pressures, i.e., total, static, and velocity pressure (see Appendix II, page 335). With the range in flow velocity likely to be encountered in tests of electronic equipment, the ordinary U-tube manometer and the inclined-tube manometer are best suited for measurement of these pressures. An inclined-tube manometer is simply a U-tube manometer having one leg inclined at an angle such that ordinarily a scale magnification of 8 or 10 to 1 is obtained. The other leg has a large bore to produce a cistern effect so that the head equals the reading of the inclined leg. This instrument is shown in Figure III-14.

Figure A-III-3 shows the conventional manometer arrangement for the Pitot-static tube. The arrangement illustrates measurement of all three

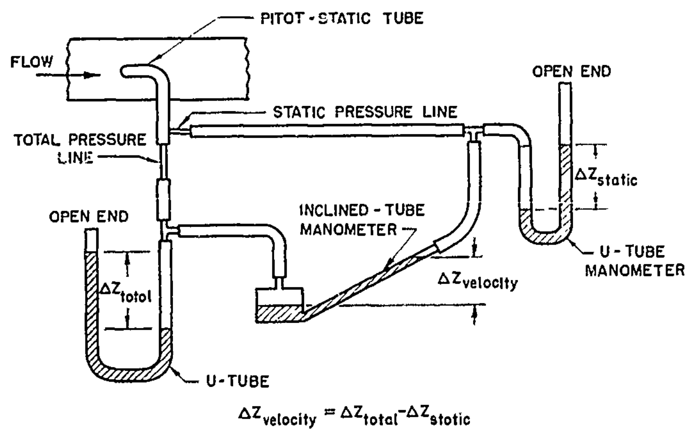


Figure A-III-3. Manometer Arrangement for Pitot-Static Tube pressures, static, total and velocity. The arrangement is such that the total and static pressures subtract to yield a direct indication of the velocity pressure in inches of manometer fluid. In actual practice, measurement of velocity pressure and either total or static pressure is sufficient.

Velocity pressures of small magnitude, such that an inclined-tube manometer is unsatisfactory, would be measured by a micromanometer. This instrument is shown in Figure III-15.

APPENDIX IV

AIR FLOW TEST APPARATUS

A general description of the auxiliary air flow apparatus to be used in conjunction with the thermal evaluation of air-cooled electronic equipment is given in Chapter III, pages 41 to 44. Reference should be made to Figure III-16, page 42, for a schematic view of the arrangement of the components constituting the apparatus, and to Figure III-17, page 42, for the general arrangement when used for forced- and induced-flow cooling. The purpose of this appendix is to present the structural details of this apparatus and the flow charts associated with the orifice meter for determination of the rate of air flow. An alternate arrangement in which the orifice meter is replaced by a variable area meter is also discussed.

1. Structural Details of Apparatus with Orifice Meter

Figures A-IV-1, -2, and -3 show structural details of the flow straightener, the duct containing the metering section, duct flanges for orifice meter, the thin-plate orifices, pressure taps, and the bleed and throttle valves. Details on the various drawings are sufficient to allow construction of the apparatus. It is desirable to construct all parts of brass, but other materials can be substituted. The orifice plates should be made of a material not subject to atmospheric corrosion.

Constructional details of an orifice plate are somewhat arbitrary. They are defined in Section III of Gas Measurement Committee Report No. 2 issued by the American Gas Association, May 6, 1935, as follows:

> "The thickness of the orifice plate shall be at least 1/16 inch. For use in pipe of over 4-inch inside diameter the thickness shall be at least 1/8 inch and not over 1/4 inch, except that for pipes 16 inch and over in diameter the thickness may exceed 1/4 inch.
>
> The upstream face of the orifice shall be as flat as can be obtained commercially and shall be perpendicular to the axis of the pipe, when in position between the orifice flanges. Since perfect flatness is difficult to obtain, any plate which, when clamped between flanges does not depart from flatness along any diameter by more than 0.01 inch per inch of pipe radius shall be considered flat.
>
> The upstream edge of an orifice shall be square and sharp so that it will not appreciably reflect a beam of light when viewed without magnification. It shall be maintained in this condition at all times. Moreover, the plate is to be kept clean at all times and free from accumulations of dirt and other extraneous material.
>
> The thickness of the orifice plate at the orifice edge shall not exceed:

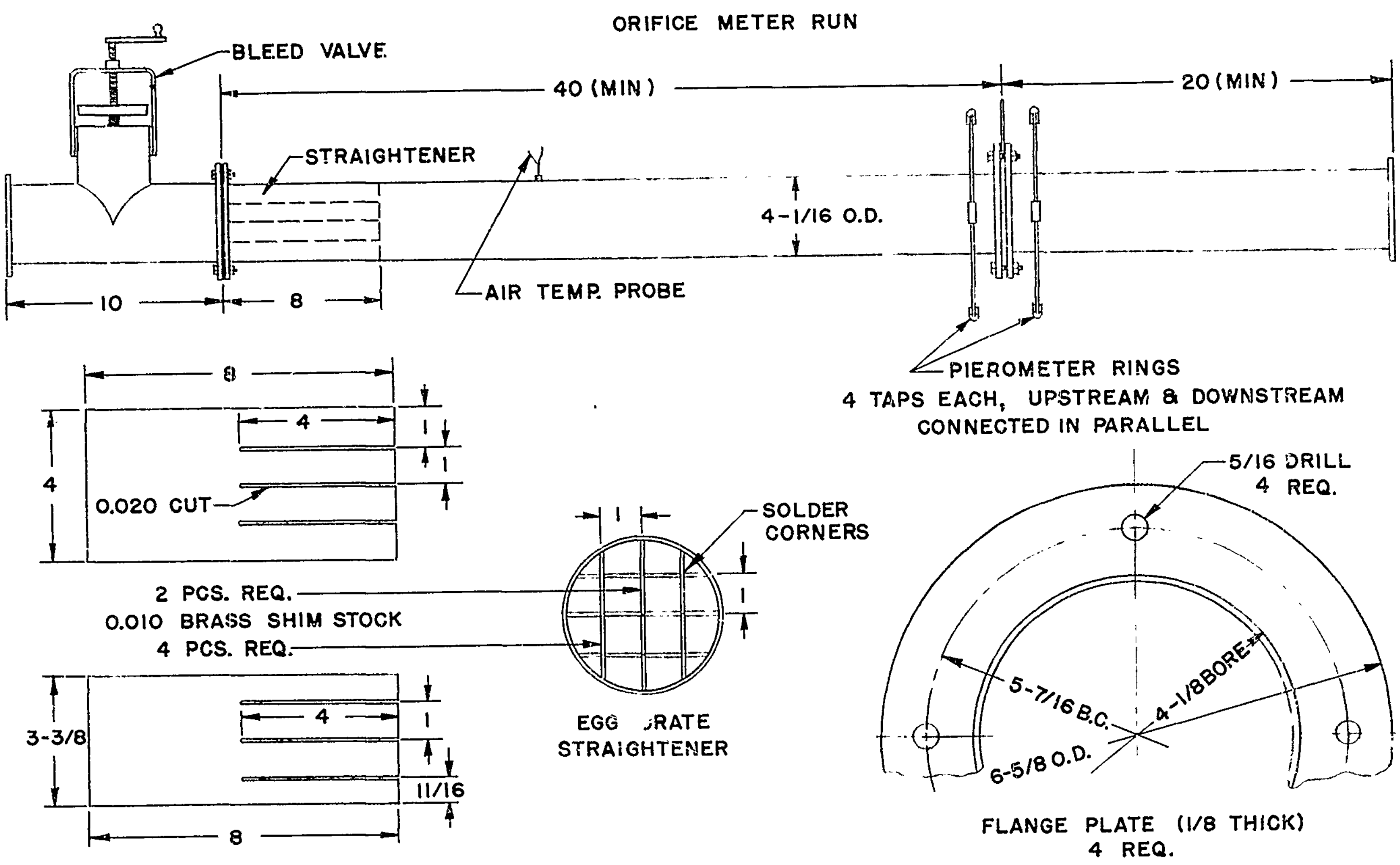


Figure A-IV-1. Orifice Meter Duct, Flange and Flow Straightener Details

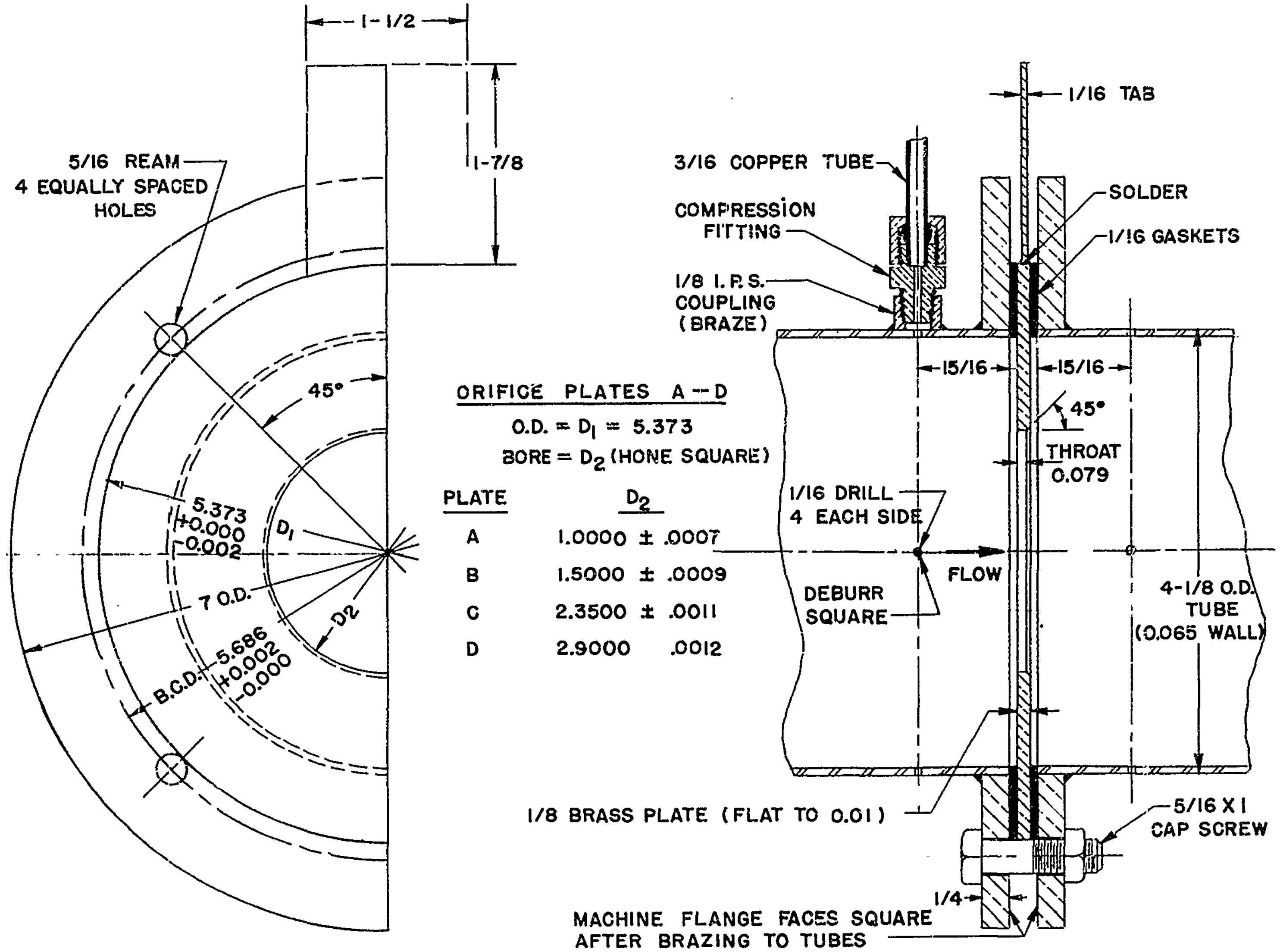


Figure A-IV-2. Orifice Plate, Flange and Pressure Tap Details

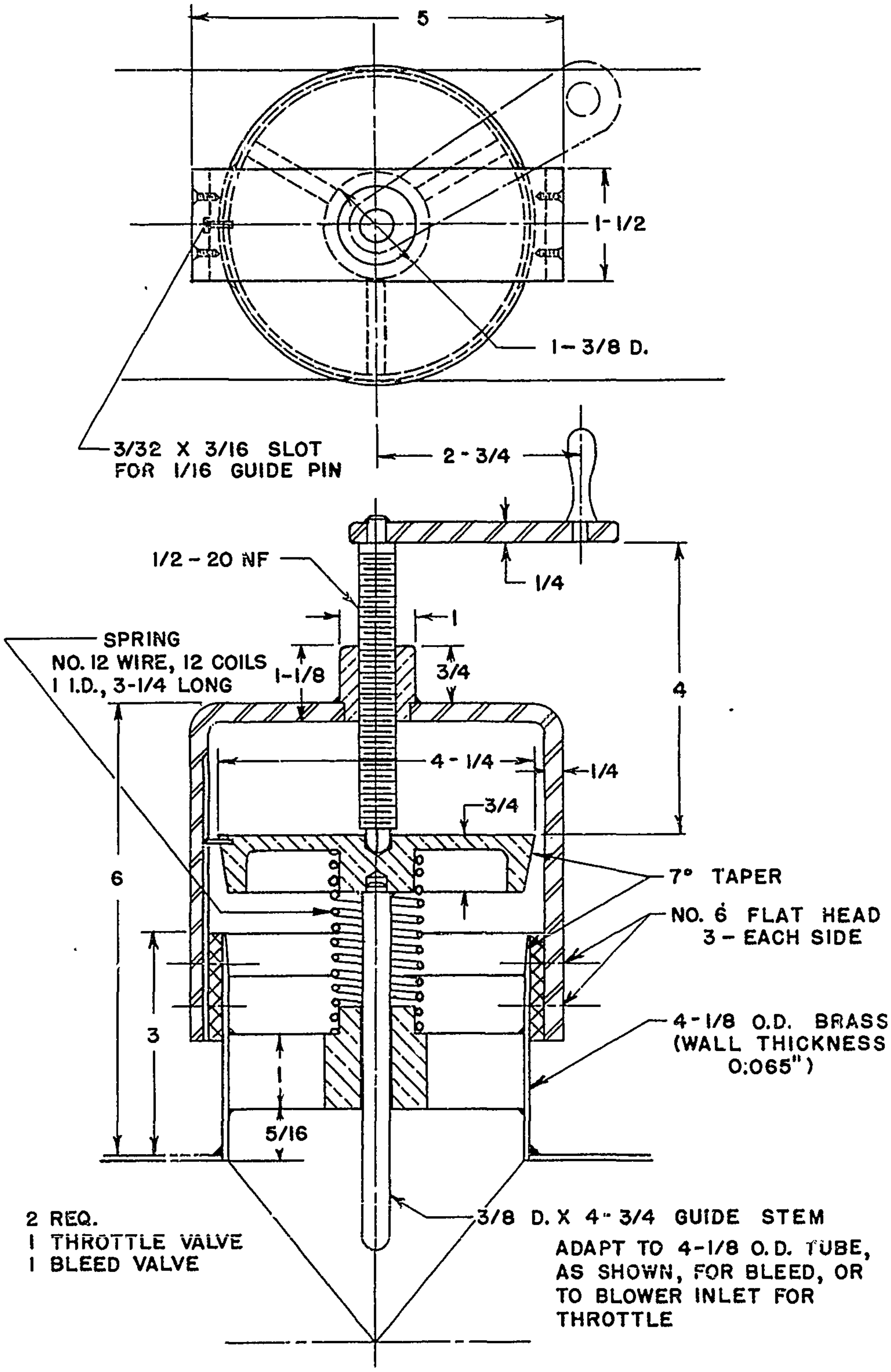


Figure A-IV-3. Throttle and Bleed Valve Assembly

(1) 1/30 of the pipe diameter, D
(2) 1/8 of the orifice diameter, D_2
(3) 1/4 of the dam height, $(D - D_2)/2$

the minimum of these requirements governing in all cases. It is recommended that whenever practicable the requirement (1) be reduced to 1/50 of the pipe diameter, D."

The dimensions shown in Figure A-IV-2 for the four orifice plates required to cover the flow range from 0.01 to 0.3 pound per second (8 to 235 cubic feet per minute standard air), fall within these requirements.

2. Air Flow Charts

The charts of Figures A-IV-4, -5, -6 and -7 are for orifice plates which have diameters of 1.00, 1.50, 2.35 and 2.90 inches, respectively, and are machined according to the above specifications. The charts allow direct evaluation of the air flow rate in pounds per second with less than 2 per cent error in terms of the measurable flow variables, i.e., static pressure drop across the orifice in inches of water, absolute static pressure ahead of orifice in inches of mercury, and the upstream air temperature in °C. The procedure of using the charts is apparent by following the dashed lines. Except for the one-inch orifice, Figure A-IV-4, a slight correction must be made on the air flow rate for any pipe diameter which differs somewhat from the standard of 4 inches. The correction factor is determined from the plot on the right side of each chart. It is desirable to keep the orifice static pressure drop in excess of one inch of water to insure best accuracy, which may be done in most all cases by proper selection of the orifice size.

Example A-IV-1. Use of Air Flow Charts

A 2.35-inch orifice plate installed in a 4-1/8-inch line indicates a static pressure differential across the orifice of 2.3 inches water. The gage pressure measured upstream of the orifice is 5.5 inches water and the upstream air temperature is 22°C. The barometer reads 28.8 inches mercury. Determine the air flow rate in pounds per second.

By equation (A-I-2)

$$p = p_{bar} + 0.0738\ p_g = 28.8 + 0.0738\ (5.5) = 29.21 \text{ inches mercury absolute}$$

Hence, using Figure A-IV-6, starting on the abscissa with a static pressure drop of 2.3 inches water, reading upwards to the intersection with a static pressure of 29.21 inches mercury absolute, horizontally to the right to a temperature of 22°C, and vertically upwards gives a flow rate 0.147 pound per second. On the right of the chart, reading for an inside diameter of 4-1/8 inches, the correction factor is 1.010. Thus, the air rate is

$$W = 1.01 \times 0.147 = 0.148 \text{ pound per second}$$

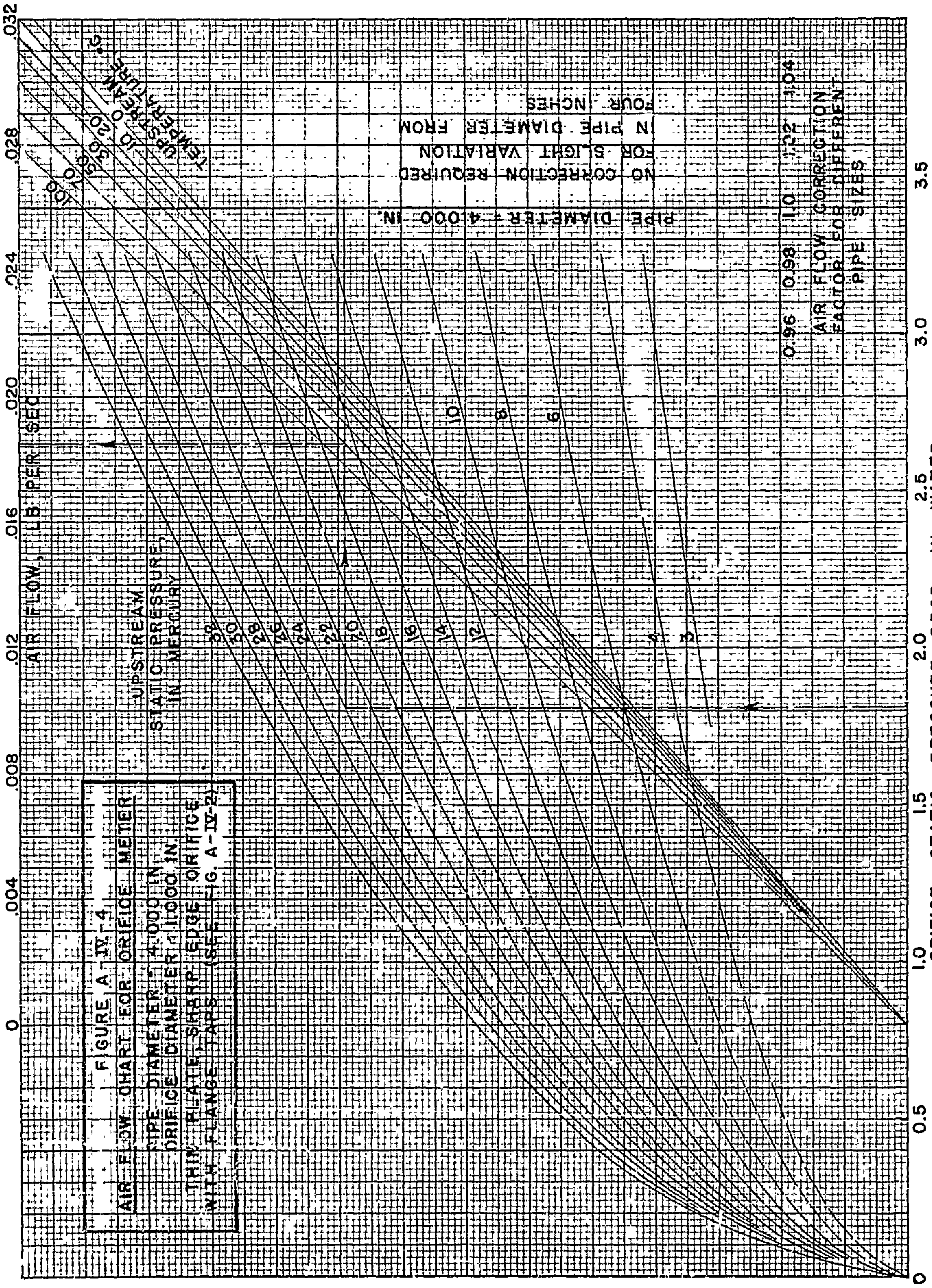


FIGURE A-IV-4
AIR FLOW CHART FOR ORIFICE METER
PIPE DIAMETER = 4.000 IN.
ORIFICE DIAMETER = 1.000 IN.
THIN PLATE, SHARP EDGE ORIFICE
WITH FLANGE TAPS (SEE FIG. A-IV-2)
UPSTREAM
STATIC PRESSURE,
IN. MERCURY
32 30 28 26 24 22 20 18 16 14 12 10 8 6 4 3
UPSTREAM
TEMPERATURE
100 70 50 30 10 0 -10
PIPE DIAMETER = 4.000 IN.
NO CORRECTION REQUIRED
FOR SLIGHT VARIATION
IN PIPE DIAMETER FROM
FOUR INCHES
0.96 0.98 1.0 1.02 1.04
AIR FLOW CORRECTION
FACTOR FOR DIFFERENT
PIPE SIZES
AIR FLOW, LB PER SEC
0 .004 .008 .012 .016 .020 .024 .028 .032
ORIFICE STATIC PRESSURE DROP, IN. WATER
0 0.5 1.0 1.5 2.0 2.5 3.0 3.5

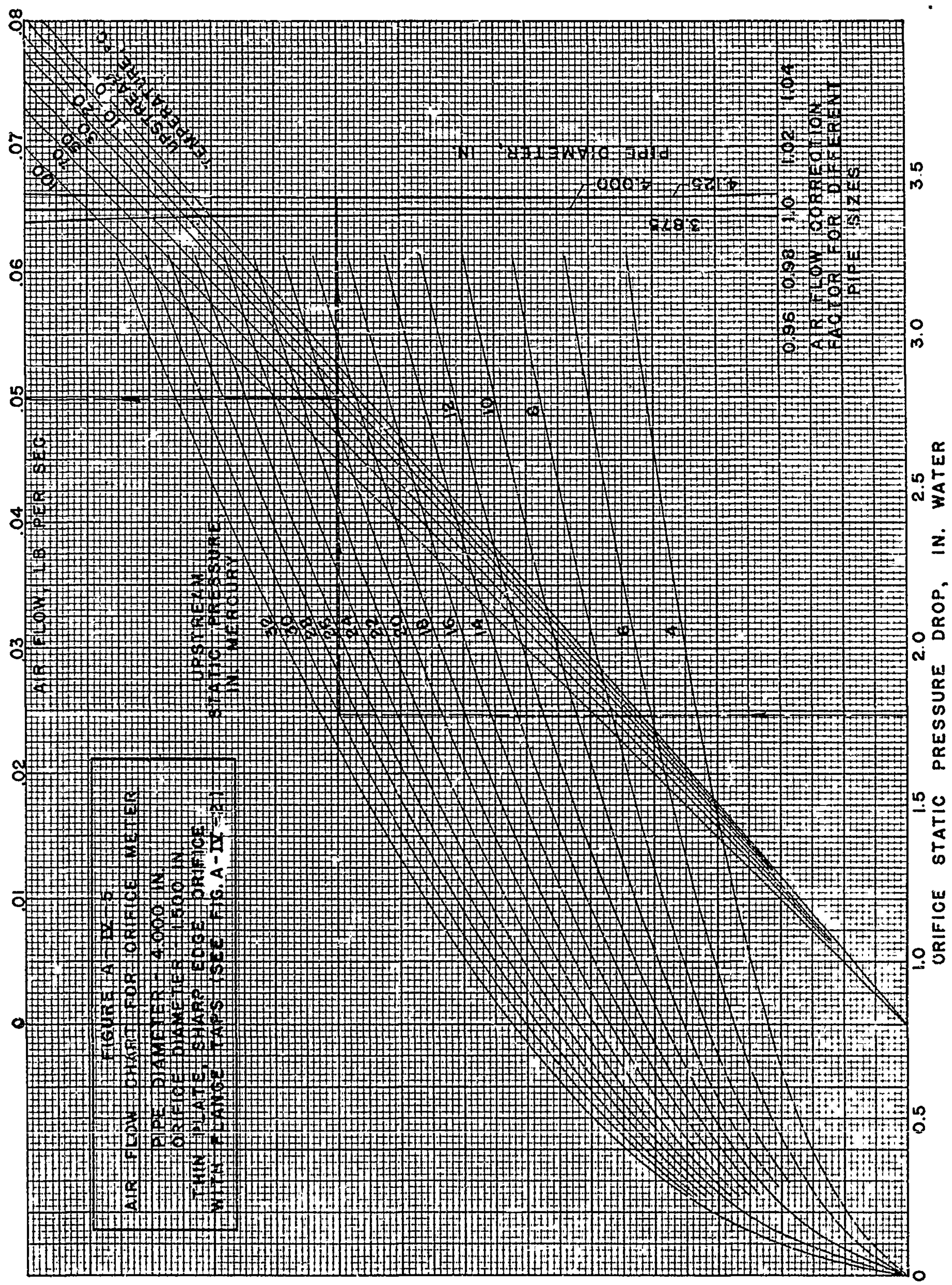


FIGURE A-IX-5
AIR FLOW CHART FOR ORIFICE METER
PIPE DIAMETER 4.000 IN.
ORIFICE DIAMETER 1.500 IN.
THIN PLATE, SHARP EDGE ORIFICE
WITH FLANGE TAPS (SEE FIG. A-IX-2)
AIR FLOW, LB PER SEC
0 .01 .02 .03 .04 .05 .06 .07 .08
ORIFICE STATIC PRESSURE DROP, IN. WATER
0 0.5 1.0 1.5 2.0 2.5 3.0 3.5
UPSTREAM STATIC PRESSURE IN. MERCURY
32 30 28 26 24 22 20 18 16 14 12 10 8 6 4
UPSTREAM TEMPERATURE °F
100 80 60 40 20 0
PIPE DIAMETER, IN
3.875 4.000 4.125
0.96 0.98 1.0 1.02 1.04
AIR FLOW CORRECTION FACTOR FOR DIFFERENT PIPE SIZES

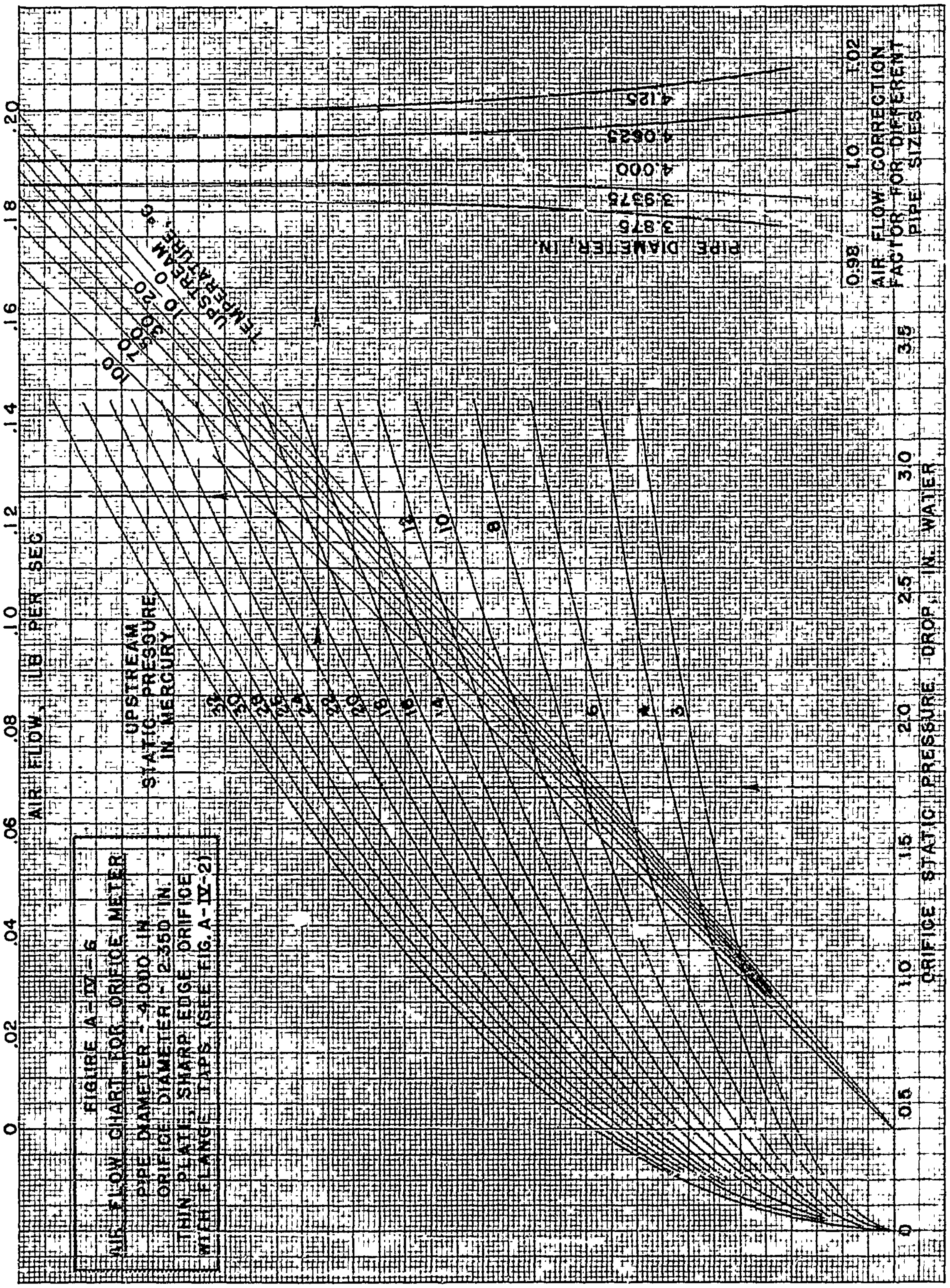


FIGURE A-IV-6
AIR FLOW CHART FOR ORIFICE METER
PIPE DIAMETER - 4.000 IN.
ORIFICE DIAMETER - 2.350 IN.
THIN PLATE, SHARP EDGE ORIFICE
WITH FLANGE TAPS (SEE FIG. A-IV-2)
AIR FLOW, LB PER SEC
UPSTREAM STATIC PRESSURE IN. MERCURY
UPSTREAM TEMPERATURE °F
PIPE DIAMETER, IN.
AIR FLOW CORRECTION FACTOR FOR DIFFERENT PIPE SIZES
ORIFICE STATIC PRESSURE DROP, IN. WATER

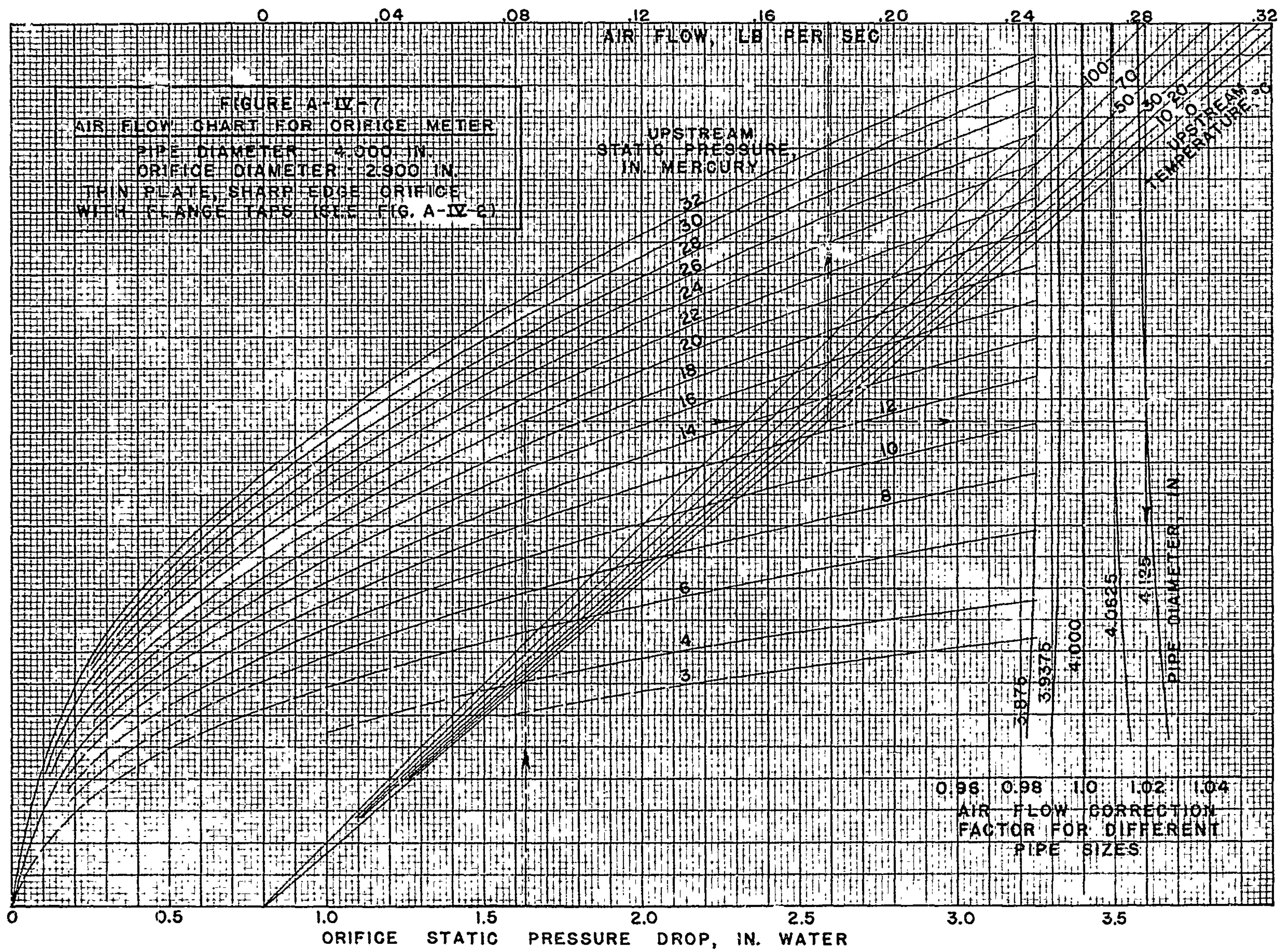


FIGURE A-IV-7
AIR FLOW CHART FOR ORIFICE METER
PIPE DIAMETER - 4.000 IN.
ORIFICE DIAMETER - 2.900 IN.
THIN PLATE, SHARP EDGE ORIFICE
WITH FLANGE TAPS (SEE FIG. A-IV-2)

3. Alternate Apparatus with Variable-Area Meters

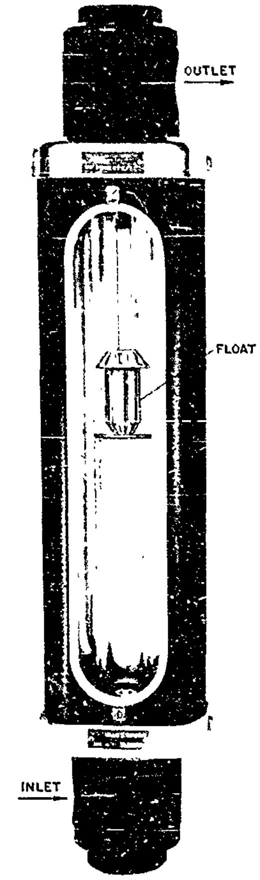


Figure A-IV-8. Variable-Area Flow Meter (Fischer and Porter Company)

The construction of the flow apparatus can be simplified by use of direct-reading variable-area flow meters, such as show in Figure A-IV-8. The air flow is vertically upward within the glass tube having a small internal taper. Thus, the float within the tube rises higher the greater the flow and remains in a fixed position for constant flow. The engraved scale on the tube is usually linear and covers a range of one to ten. Thus for the required flow range, two meters of different capacity would be required. They would be installed in parallel and can be used individually or jointly. The piping connections to the required meters would be 1-1/4 inch and 4 inch diameter. The large meter sizes are required since this type meter has inherently a fairly high pressure drop and for application in the air flow apparatus it is desirable to hold the pressure drop to less than 3 inches water. This is for the purpose of permitting use of a low-pressure blower with the apparatus and also to make the meters adaptable to blower test work, as mentioned in Appendix V.

The flow apparatus with the variable-area meters is appreciably shorter than with the orifice meter, since the long pipe sections are not required. The variable-area meters can be installed immediately adjacent to the blower discharge or inlet and equipment inlet or output, depending on whether forced or induced flow is used. Each should have a gate valve in its branch for shut-off purposes.

The compact installation obtained by means of these meters exceeds in cost that with the orifice meter, but has not only the advantage of reduced over-all dimensions, but also of greater convenience. An entire flow range of one to one hundred can be covered by operation of the valves alone while, when using the orifice meter, the plates must be changed.

If the installation of the flow apparatus in a large altitude chamber is contemplated, the variable area meters cannot be used for the same weight flow rates and have greatly reduced capacity. Suitable meters would be of prohibitive size.

4. Blower Requirements

The blower for the auxiliary air flow apparatus to be used in bench test work only may be a squirrel-cage type centrifugal unit capable of up to 300 cubic feet per minute air flow at a static pressure of 5 to 6 inches water. Greater pressure capacity is desirable if the use of fewer orifice plates to cover a given flow range is desired, or if the sizes of the variable-area meters, if used, are to be reduced. In the latter case, it is usually possible to reduce the cost of the meters appreciably by providing a blower capable of producing a static pressure of about 18 inches water.

If the air flow apparatus is to be installed in an altitude chamber, its volumetric capacity must be considerably greater. To operate at 50,000 feet altitude pressure (p_o = 3.426 inches mercury) and 15°C air temperature at an air flow rate of 0.1 pound per second, the capacity of the blower should be almost 700 cubic feet per minute. Its corresponding static pressure at ground level should be about 35 inches water. This would require a high-speed centrifugal impeller of small width.

APPENDIX V

BLOWER TEST AND PERFORMANCE EVALUATION METHODS

Thermal evaluation of most air-cooled electronic equipments requires knowledge of the performance of blowers used as the source of air motion. In some instances, manufacturers' data, such as given in Chapter V, are available describing a blower's performance. However, frequently detailed information on the performance of a given blower can only be obtained by test of the unit.

The purpose of this appendix is to recommend methods for test, evaluation and presentation of performance of blowers used as a source of air motion for cooling airborne electronic equipment.

Definition of Performance Parameters

Absolute static pressure p defines the actual absolute pressure of the air at inlet or outlet of the blower. Its magnitude is expressed in units of inches of mercury. Gage static pressure p_g defines the actual pressure of the air above or below the ambient barometric pressure, expressed in the units of inches of water. Gage static pressure p_g is equal to the difference of the absolute static pressure and barometric pressure (see equation A-I-2).

Velocity pressure p_v is a pressure change which would be manifested when the flowing air is ideally stagnated. Its magnitude is a measure of the kinetic energy of the flowing air and may be evaluated by equation (A-II-12), or by the equivalent but more convenient relationship for performance evaluation of blowers,

$$p_v = (\sigma/763)(Q/A)^2 \qquad \text{(A-V-1)}$$

where σ is the ratio of the actual air density to the standard value of 0.0765 pound per cubic foot (see Figure V-8), Q the flow rate in cubic feet per minute and A the cross-sectional flow area in square inches, all evaluated at the same position along the flow path. In general, velocity pressures at inlet and outlet of a blower are different in magnitude.

Total pressure P is defined as the algebraic sum of static pressure and velocity pressure. It is expressed as an absolute pressure in inches of mercury and as a gage pressure in inches of water.

The total pressure rise of a blower ΔP in inches of water is defined as the increase in total pressure of the air between inlet and discharge of the blower. Total pressure rise of a blower is a measure of the total energy imparted to the air by the blower.

The static pressure rise of a blower Δp is defined as the difference between the static gage pressure of the air at the discharge of the blower and the total gage pressure of the air at the inlet to the blower, all in inches of water. Hence,

$$\Delta p = p_{g-d} - P_{g-i} \qquad \text{(A-V-2)}$$

Volume flow of air Q is the cubic feet per minute of air flowing through a specified section, either at inlet or discharge. The recommended practice is to rate the volumetric capacity of a blower on basis of inlet volume flow. For most blowers the inlet and discharge volume flows are essentially equal.

Air horsepower (ahp) is defined as the rate at which energy is imparted to the air by the blower in a form resulting in pressure change of the air. Air horsepower may be evaluated on basis of static or total pressure rise in inches of water by the equations

$$ahp_s = Q\Delta p/6350 \qquad \text{(A-V-3)}$$

$$ahp_t = Q\Delta P/6350 \qquad \text{(A-V-4)}$$

The shaft horsepower (hp) of a blower is the actual shaft power input required to drive the unit. The ratio of air horsepower to shaft horsepower defines the efficiency of a blower, called static efficiency η_s, when air horsepower is evaluated on basis of static pressure rise Δp, and total efficiency η_t with air horsepower defined by total pressure ΔP. Hence,

$$hp = Q\Delta p/6350\ \eta_s = Q\Delta P/6350\ \eta_t \qquad \text{(A-V-5)}$$

The electrical input power to the driving motor ep expressed in watts, is related to the shaft horsepower by

$$hp = \eta_m\ ep/746 \qquad \text{(A-V-6)}$$

where η_m represents the efficiency of the electric motor. The combined efficiency of the blower-motor unit, the set efficiency η_{s-m}, is defined as the product of the static efficiency of the blower and the efficiency of the motor. It represents the fraction of the input power to the motor given to the air passing through the blower in the form of air horsepower.

Rotational speed N is defined as the revolutions per minute of the blower.

Standard air density is defined as 0.0765 pound per cubic foot, corresponding to standard sea level pressure and temperature of 29.92 inches of mercury and 15°C, respectively.

Test Set-Up

It is recommended that performance of centrifugal and axial-flow blowers be determined by blower test equipment consisting of the auxiliary airflow

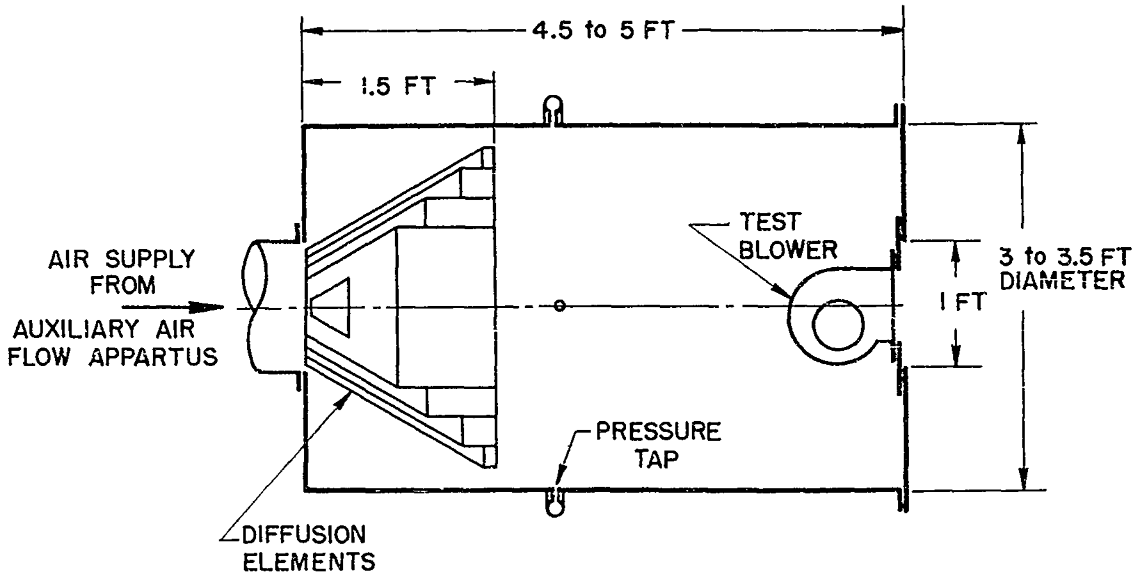


Figure A-V-1. Plenum Chamber for Blower Tests

apparatus described in Appendix IV and a plenum chamber constructed as shown in Figure A-V-1. The plenum chamber would be circular or rectangular in cross-section and substantially air-tight. The diameter, or minimum width of any side panel when rectangular, shall be at least three-times the diameter of the test blower outlet, and the length of the chamber about four-times this dimension. Uniform distribution of air in the test chamber is necessary and would be obtained by providing at the inlet end of the chamber three to five diffusion elements, consisting of concentric truncated cones when the chamber is circular and truncated four-sided pyramids when the chamber is rectangular. The diffusion elements would be designed and arranged to provide equal flow area between any two cones or pyramids at both their entrance and exit. Unidirectional flow at exit of the diffusion elements would be obtained by addition of cylindrical lips, as illustrated in Figure A-V-1. The horizontal length of the diffusion elements shall not exceed 35 to 40 per cent of the chamber length. The chamber should be designed structurally to withstand a pressure differential equal to the maximum pressure generated by any blower to be tested.

A plenum chamber designed for permanent installation in a test laboratory and intended to be used for test of any blower likely to be associated with electronic equipment should have a diameter of 3 to 3-1/2 feet and a length of 4-1/2 to 5 feet. The diffusion elements should be of sheet-metal construction, having a horizontal length of about 1-1/2 feet. For simplification of construction an ordinary baffle plate, about 1-1/2 feet in diameter may be substituted as the diffusion element and will satisfactorily distribute the air to the chamber when the flow rate does not exceed 300 to 400 cubic feet per minute and the blower or electronic equipment installed therein occupies only a small fraction of the chamber volume. The baffle plate would be located concentrically with the inlet opening of the chamber and about 1-1/2 feet downstream from it. The chamber should be designed to withstand a pressure differential of from 30 to 40 inches of water, necessitating the use of at least 12-gage sheet metal construction. The discharge end of the chamber should have a concentric circular opening of about one foot diameter, permitting accommodation of smaller-size blowers by attachment of circular plates to the chamber opening.

Static pressure of the air within the chamber is measured through four flush taps equally spaced around the periphery of the chamber in a plane approximately 2-1/2 feet from the chamber inlet. The four static pressure taps are connected to a common manifold, external to the chamber, to permit measurement of average static pressure within the chamber. Standard manometers should be connected to the manifold to indicate the magnitude of the chamber pressure. The type of manometer employed should have an accuracy of reading within two per cent at the maximum pressure to be measured. Hence, with blowers developing a pressure change greater than approximately 4 to 5 inches of water, ordinary U-tube manometers filled with water or standard manometer fluid may be used. Blowers developing maximum pressures less than 4 to 5 inches water would require use of inclined-tube manometers.

The temperature of the air within the plenum chamber may be taken equal to the temperature measured ahead of the air meter in the auxiliary air flow apparatus, obviating the need of provision for temperature measurement within the chamber. An observation window should be provided in one side of the plenum chamber for purposes of viewing the test blower, an equipment, and/or instruments associated with determination of blower performance.

Testing of Blowers

For test, the blower would be mounted in the plenum chamber shown in Figure A-V-1. Since the purpose of the tests is to determine the blower's pressure-producing ability, volumetric capacity, power consumption and efficiency, it is necessary to conduct the test and record information in a manner such that accurate evaluation of these performance parameters may be made.

1. Test Data

Experimental observations required during the test of any blower are: (1) barometric pressure at location of test set-up, (2) temperature and static pressure of air at entrance to orifice or area meter in auxiliary air flow apparatus, (3) static pressure differential across the orifice meter when used in the auxiliary air flow apparatus, (4) average pressure of air in plenum chamber, (5) rotational speed of blower, (6) electrical power input to blower's driving motor, and (7) dry- and wet-bulb temperature of air at discharge from plenum chamber. This latter measurement would not normally be made in routine testing of blowers used in conjunction with cooling of electronic equipment. It should be made when it is desired to conduct the test according to standard code practice, or when it is known that the air used in the test is nearly saturated with water vapor and possible future correction is desired.

Barometric pressure should be measured by a mercury barometer located in the same room or in the vicinity of the test. Items (2) and (3) are measured by instrumentation provided as an integral part of the auxiliary air flow apparatus. The average pressure in the plenum chamber would be measured by the method recommended on page 366. The average pressure so determined defines the static gage pressure in the plenum chamber p_{g-ch} in inches of water which is related to the total gage pressure of the air in the chamber P_{g-ch} in inches of water by

$$P_{g-ch} = p_{g-ch} + (\sigma_{ch}/763)(Q_{ch}/A_{ch})^2 \tag{A-V-7}$$

where σ_{ch} is the ratio of the air density in the chamber to the standard air density of 0.0765 pound per cubic foot, Q_{ch} the flow rate through the chamber and the blower expressed in cubic feet per minute and A_{ch} the cross-sectional area of the chamber in square inches. However, the difference between the total and static pressure of the air within the chamber is negligible, less than 0.001 inch water, except when blowers having air capacity in excess of several thousand cubic feet per minute are being tested. Therefore, only under such conditions would the total pressure of the air in the chamber have to be calculated from measured values by equation (A-V-7).

The volume flow of air through the chamber Q_{ch} in cubic feet per minute is related to the weight rate of flow W in pounds per second, by the equation

$$Q_{ch} = 784\ W/\sigma_{ch} \tag{A-V-8}$$

Rotational speed of the blower may be measured by (1) revolution counter and calibrated timing device, (2) hand tachometer, (3) electric tachometer, or (4) electronic tachometer (strobotac). The method to be employed depends upon the type of blower being tested and accessibility to any rotating member of the blower-motor unit. Use of a revolution counter and an accurate timing device is recommended when a rotating member of the unit being tested is directly accessible or may be made accessible by a shaft extension on the counter. Use of the strobotac is necessary when a rotating member is accessible by observation only. For example, the strobotac may be focused on the rotor of the blower through an observation window in the wall of the plenum chamber.

Electrical power input to the blower's drive motor would be measured by use of one or several wattmeters. Power delivered to the blower would be determined from the efficiency curve of the electric motor. Dry- and wet-bulb temperatures for evaluation of density of moist air would be measured by a sling psychrometer, discussed on page 326.

2. Blower Test Procedure

Preceding any test or series of tests, all instrumentation should be checked to assure accuracy of reading when in use. Instruments known to be somewhat unreliable or affected by continuous use should be re-calibrated periodically.

When starting a test, the blower should be operated for a period of time sufficient to assure steady-state conditions before any readings are taken. Eight or more test runs are desirable for any one blower, varying the air flow, by throttling between the auxiliary air supply duct and the chamber inlet, in approximately equal increments from the maximum capacity of the blower to zero air delivery. Three complete sets of data should be recorded for each air flow setting. Each complete set of data requires simultaneous reading of temperature, pressure and pressure differential at the air flow meter in the auxiliary air flow apparatus, pressure in the plenum chamber, rotational speed of the blower and electrical power input to the blower's drive motor. An average of the three values obtained for each variable would be taken to represent the true value of the variable for the particular air flow setting.

Normally, the first of a series of tests on a blower would be at its maximum air capacity, corresponding to zero static pressure rise. The rate of air flow delivered by the auxiliary air flow apparatus would be increased until the pressure in the plenum chamber equals the atmospheric pressure, at which point the blower operates against zero static pressure rise and delivers the maximum quantity of air. Upon completion of the series of readings for this air flow setting, the air supplied by the auxiliary air flow apparatus would be reduced by from 10 to 15 per cent, resulting in reduction of the plenum chamber pressure below atmospheric pressure and causing the blower to develop a pressure rise of the air in order that the air may be discharged from the plenum chamber to the atmosphere. Again, a series of three complete sets of readings would be taken. The procedure would be repeated until the test for zero air delivery is completed.

With many blowers, both centrifugal and axial-flow, operation at air capacities from 10 to 15 per cent of their maximum air capacity will produce pulsating air flow conditions resulting in unstable operation of the blower. When operation with pulsating flow is considered detrimental to the equipment or produces test results of questionable accuracy, the tests would be conducted only from maximum air capacity to that reduced capacity at which pulsation is first encountered, the latter point corresponding to the maximum stable pressure-producing ability of the blower.

Types of Blower Tests

Various arrangements for testing of blowers are required. In many instances it will be possible to remove the blower from the electronic equipment in which it is installed, making the mechanics of a blower test relatively simple. In other instances removal of the blower from an equipment will prove impractical, necessitating that the blower be tested while remaining an integral part of the equipment.

1. Centrifugal Blowers with Impeller Housing and Diffuser, Removable from Equipment

A common type centrifugal blower used in electronic equipment has a single or double ductless inlet and a scroll-type housing forming the diffuser and discharge section, so that the air is delivered from the blower through a single discharge opening. With the blower removed from the equipment, the recommended test set-up is as illustrated in Figure A-V-1. The rotational speed may be measured by an electronic tachometer (strobotac) focused through an observation window in the wall of the plenum chamber on the impeller of the blower.

The static pressure rise of the air from blower inlet to discharge equals numerically the gage pressure measured in the plenum chamber, corrected by equation (A-V-7), but is of opposite sign. The static pressure rise of the blower is, from equation (A-V-2)

$$\Delta p = 0 - P_{g-ch}, \qquad \text{(A-V-9)}$$

since the blower discharges to the atmosphere and the static gage discharge pressure is, therefore, zero.

The total gage pressure at the blower discharge equals the velocity pressure at the discharge and would be evaluated by equation (A-V-1) in the form

$$p_{v-d} = P_{g-d} \, (\sigma_d/763)(Q_d/A_d)^2, \qquad \text{(A-V-10)}$$

where A_d is the blower discharge area in square inches, σ_d is the density ratio and Q_d the flow volume corresponding to blower discharge conditions. With most blowers the air density ratio σ_d is, for all practical purposes, equal to the chamber density ratio σ_{ch}, so that the outlet volume flow Q_d may be taken equal to the chamber volume flow Q_{ch}. If, however, the pressure increase of the air across the blower is appreciable, σ_d should be defined by the atmospheric pressure and a measured blower discharge temperature. Then, the volume flow should be corrected for change in air density from inlet to discharge according to $Q_d = Q_{ch}(\sigma_{ch}/\sigma_d)$. The total pressure rise of the blower is the algebraic difference of the discharge and inlet total gage pressures.

2. Axial-Flow Blowers, Removable from Equipment

The tests of an axial-flow blower separate from its equipment would be conducted in essentially the same manner as for a centrifugal blower removed from its equipment. The blower would be mounted in the outlet section of the plenum chamber, with air flow rate controlled by the auxiliary air flow apparatus. Evaluation of inlet and discharge static and total pressure would follow the procedures outlined in the preceding paragraphs for centrifugal blowers. Depending upon the design of the blower, rotational speed would be measured with a hand tachometer, revolution counter or an electronic tachometer. Blowers of very small capacity usually require use of the electronic tachometer, since direct contact of any instrument with a rotating part of the blower-motor unit may alter its rotational speed.

3. Blowers Not Removable from Equipment

Frequently, it is not feasible to remove a blower from an equipment in which it is installed. Hence, under these circumstances performance evaluation of the blower by test may be obtained only by connecting the equipment to the discharge of the plenum chamber or installing it within the chamber. Not all equipments are adaptable to the plenum chamber for integral testing of internal blowers.

In general, two types of blower and air flow arrangements within an equipment make possible test of blowers not removable from equipments. If the blower induces air through an equipment and discharges it through a single outlet in one end of the equipment case, it would normally be possible to install the entire equipment within the plenum chamber in a manner such that the air outlet is located at the discharge opening of the plenum chamber. This general arrangement is illustrated in Figure A-V-2. The static pressure at the discharge of the blower would be zero gage. The static pressure at the inlet of the blower should be measured by a pressure tap located in the intake section of the blower. Flexible tubing connected to the static tap would transmit the pressure to the outside of the plenum chamber where its magnitude may be measured by a suitable manometer arrangement. The difference between the static pressures in the plenum chamber and at the inlet of the blower may be used to evaluate the flow resistance of the equipment. The total gage pressure at the discharge of the blower would be evaluated from equation (A-V-10), and that at the inlet by

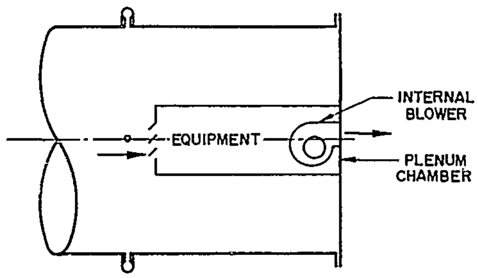


Figure A-V-2. Blower Test Arrangement for Equipment with Single Discharge

$$P_{g-i} = p_{g-i} + (\sigma_i/763)(Q_i/A_i)^2 \qquad \text{(A-V-11)}$$

The static pressure rise of the blower Δp is numerically equal to the inlet total pressure. Measurement of rotational speed may require installation of small observation window in the equipment case.

Forced flow with a single inlet to the blower in one end of an equipment case is the second general type of flow arrangement adaptable for testing a blower not removable from its equipment. The equipment would be mounted external of the plenum chamber as illustrated in Figure A-V-3. Air flowing from the plenum chamber passes through the blower into the equipment and then discharges to the atmosphere. When the inlet section of the blower is adjacent to the discharge section of the plenum chamber, total pressure at the blower inlet would be equal to the pressure measured in the plenum chamber. However, should the inlet section of the blower be some distance

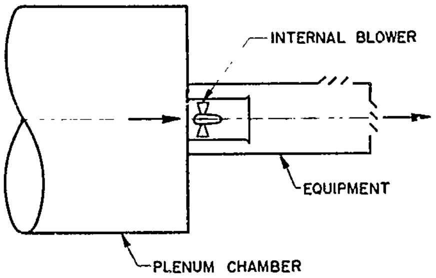


Figure A-V-3. Blower Test Arrangement for Equipment with Single Inlet

from the discharge section of the plenum chamber, it would be necessary to install a static pressure tap in the inlet section of the blower and to calculate the inlet total pressure by equation (A-V-11).

4. Blowers Used for Internal Circulation or Spot-Cooling

Blowers used for circulating internal air in closed equipment and spot-cooling of thermally critical components may require testing for evaluation of their performance. The removal of such a blower from its equipment for purposes of testing is mandatory, even though flow conditions at inlet and outlet of the blower comparable to those in the actual installation may not be simulated. The blower would be mounted in the plenum chamber and tested similarly to any centrifugal or axial-flow blower. However, blowers for spot-cooling applications are usually not capable of producing much pressure and the information of principal interest is concerned with maximum discharge capacity at atmospheric pressure in the plenum chamber and with effects on flow reduction due to the proximity of a surface near the discharge.

Generally, blowers of this type are quite small and, hence, would not require testing in a plenum chamber of the size indicated in Figure A-V-1. It is recommended that a plenum chamber of approximately one foot diameter and one and three-quarter to two feet in length be constructed when fairly extensive test programs are contemplated for small blowers. A simple plate baffle could be installed near the inlet section of the chamber. The method of measuring average chamber pressure by four taps equally spaced around the periphery of the chamber and connected to a common manifold is recommended.

Reporting Test Results

Results of tests on blowers should be summarized in a form suitable for ready reference to the type and conditions of test and performance determined. The report should include general information such as test number, type and description of blower, date and location of test, purpose of test, test observers, name-plate data of blower and the drive motor, impeller diameter, impeller width, dimensions of blower outlet and inlet, description of test set-up and identification and description of all instrumentation. All data recorded during test of the blower should be included in the report. The extent and type of data to be recorded for any test are discussed on pages 366 and 367.

Reduction of Test Results for Definition of Characteristic Performance Curves of Blowers

Characteristic performance curves of blowers are obtained by plotting pressure rise, power and efficiency as a function of the volume flow of air, as illustrated in Figure A-V-4, page 374. Conventional practice would be either to correct all performance data to a chosen reference rotational speed N_r and the standard air density where $\sigma = 1.0$, or to correct to the standard air density and to indicate speed variation resulting from change in load on blower and motor. Correction for both speed and density is more frequently employed and is called for in blower test codes.

Correction of performance data is made by use of the laws of blower performance. Referring to page , the laws state that for constant diameter

$$Q \ \alpha \ N$$

$$\Delta p \ \alpha \ \sigma \ N^2$$

$$hp \ \alpha \ \sigma \ N^3$$

Hence, with N_r representing the reference rotational speed to which all performance data are corrected and σ_i the air density ratio at inlet to the blower, the corrected performance parameters are evaluated by the equations

$$Q_{corr} = (N_r/N)\, Q \qquad \text{(A-V-12)}$$

$$\Delta p_{corr} = (N_r/N)^2(\Delta p/\sigma_i) \qquad \text{(A-V-13)}$$

$$\Delta P_{corr} = (N_r/N)^2(\Delta P/\sigma_i) \qquad \text{(A-V-14)}$$

$$hp_{corr} = (N_r/N)^3(hp/\sigma_i), \qquad \text{(A-V-15)}$$

where Q, Δp, ΔP and hp represent values obtained by test of the blower. The reference speed N_r would be chosen as the rated speed of the drive motor or as a representative value in the speed range through which the blower is intended to operate. This process of correcting test data to standard conditions may also be conducted graphically by use of Figure V-12 by plotting test data in quadrant (1) and correcting for speed variation and density in quadrants (3) and (4). Corrected values of volume flow and pressure rise would be read on the coordinate of quadrant (5). Similarly, the actual power would be introduced into quadrant (8), corrected for density and speed in quadrants (8) and (7), to give corrected values of power read on the top ordinate scale of quadrant (5).

An example illustrating the reduction of test data to standard conditions is presented in the following. A typical characteristic performance plot is shown in Figure A-V-4, page 374.

Example A-V-1. Reduction of Test Results to Establish Characteristic Performance Curves of a Blower

A centrifugal blower is tested to determine its characteristic performance as defined by corrected static and total pressure rise, corrected volume flow, corrected shaft horsepower and efficiency at constant rotational speed and standard air density. The test arrangement is as illustrated in Figure A-V-1.

Air flow rate is measured by an orifice meter located in the auxiliary air flow apparatus. Shaft horsepower delivered to the blower is evaluated from measured electrical power input to the drive motor and its known efficiency variation. The dimensions of the impeller in the blower are 4-9/16 inches diameter and 1-1/8 inch width. The outlet area of the blower is 7.11 square inches. Rated rotational speed of the drive motor is 7200 revolutions per minute.

Averages of data observed during test at each air flow setting are listed in Table A-V-1. The barometric pressure and air temperature remained constant during test at 29.20 inches mercury and 26°C, respectively.

Table A-V-1. Averaged Results of Blower Test (Example A-V-1)

Test No.	Weight flow rate, pound per second	Plenum chamber pressure, in. water gage	Rotational speed, revolutions per minute	Shaft horsepower
1	0.298	0	7165	0.490
2	0.260	-2.55	7170	0.440
3	0.216	-4.95	7175	0.330
4	0.173	-6.55	7185	0.240

The method for reduction of test data to yield corrected performance of the blower will be illustrated for Test No. 3. The reference rotational speed of the blower is chosen as 7200 revolutions per minute, the rated speed of the drive motor.

The density ratio of the air within the plenum chamber and at entrance to the blower is evaluated by equation (A-I-1) and Figure V-8.

$$p_{ch} = p_{bar} + 0.0738\, p_{g-ch} = 29.20 + 0.0738(-4.95) = 28.87 \text{ inches mercury absolute}$$

$$T_{ch} = 273 + t_{ch} = 273 + 26 = 299^{\circ}$$

$$\sigma_{ch} = 9.63\, p_{ch}/T_{ch} = 9.63 \times 28.87/299 = 0.930$$

The volume flow of air through the chamber is defined by equation (A-V-8),

$$Q_{ch} = 784\ W/\sigma_{ch} = 784 \times 0.216/0.98 = 182 \text{ cubic feet per minute}$$

The corrected volume flow handled by the blower is, by equation (A-V-12),

$$Q_{corr} = Q_{ch}\ (N_r/N)$$

$$Q_{corr} = 182(7200/7175) = 183 \text{ cubic feet per minute}$$

Since the outlet static pressure of the blower is zero gage, the static pressure rise of the blower is equal to

$$\Delta p = 0 - P_{g-ch} = -(-4.95) = 4.95 \text{ inches water}$$

and, by equation (A-V-13),

$$\Delta p_{corr} = 4.95(7200/7175)^2/0.93 = 5.36 \text{ inches water}$$

Velocity pressure at the blower outlet is defined by equation (A-V-10), which gives

$$p_{v-d} = (0.93/763)(182/7.11)^2 = 0.80 \text{ inch water}$$

and equals the outlet total gage pressure P_d. Hence, actual total pressure rise of the blower is

$$\Delta P = P_{g-d} - P_{g-ch} = 0.80 - (-4.95) = 5.75 \text{ inches water}$$

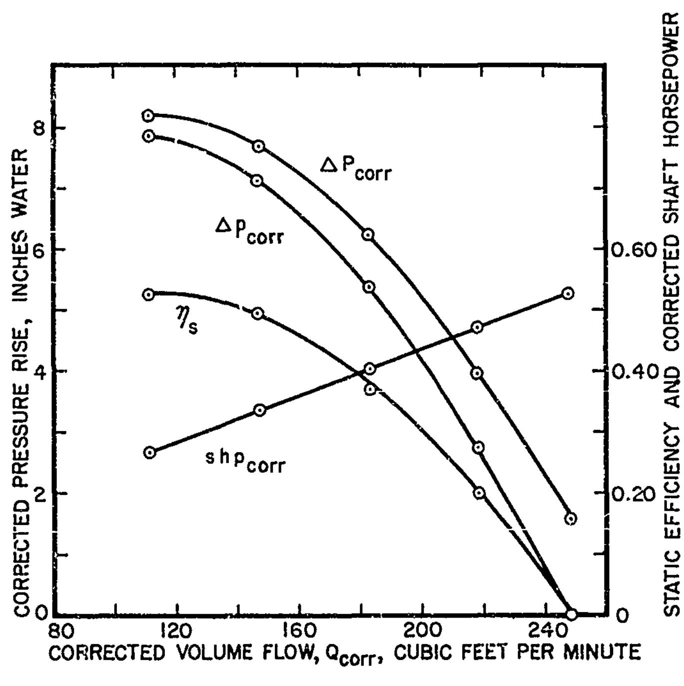


Figure A-V-4. Blower Characteristics Obtained by Reduction of Test Data (Example A-V-1) and, by equation (A-V-14),

$$\Delta P_{corr} = 5.75\ (7200/7175)^2/0.93 = 6.21 \text{ inches water.}$$

Corrected shaft horsepower is evaluated by equation (A-V-15), giving

$$shp_{corr} = 0.33\ (7200/7175)^2/0.93 = 0.402 \text{ horsepower}$$

The static efficiency of the blower is, by equation (A-V-5),

$$\eta_s = 182 \times 4.95/(6350 \times 0.33) = 0.384$$

The corrected performance for all other tests is evaluated from the data of Table A-V-1 by the same procedure. The resulting data are presented graphically in Figure A-V-4.

References

(1) Madison, R. D. Fan Engineering, Buffalo Forge Company, Buffalo, N. Y., 1949.

(2) Engineering Committee of the National Association of Fan Manufacturers and Fan Test Code Committee of the American Society of Heating and Ventilating Engineers, NAFM-ASHVE Standard Code for the Testing of Centrifugal and Axial Fans, National Association of Fan Manufacturers, Detroit, Mich., 1938, Third Printing, 1941.

(3) A.S.M.E. Power Test Code Committee, Test Code for Fans, The American Society of Mechanical Engineers, New York, N. Y., 1946.

AFTR 6379

(1) [illegible] line 6, "---1/8" should be "---3/8"

(2) page 10[illegible], equation (V-10), T_q should be Tq

(3) pages 118 and 119, Figures V-23 and -24, diagrams should be interchanged to correspond to titles given

(4) page 262, Figure VIII-2, "constucted" on lower right of diagram should be "constructed"

TO:

WCLSM

Mr. E. M. Polk

AFTR 6579

Aircraft Radiation Inc.

ıould be "---3/8"

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Symbol	Concept
a	free convection parameter, $(g'\beta\rho^2 c_p)/(\mu k)$ per (foot)^3-°F
A	cross-sectional area, square inches
A'	cross-sectional area, square feet
ahp	air horsepower (rate of energy input to air) of a blower
b	thickness, inches
B	width of blower impeller, inches
C	constant
c_p	specific heat at constant pressure, Btu per pound-°F
c_v	specific heat at constant volume, Btu per pound-°F
D	diameter, inches
e	internal energy of fluid, Btu per pound
E	heat absorbed during time of operation, watt-minutes
ep	electric input to blower motor, watts
F_A	radiation configuration factor, dimensionless, (see equation VI-3, page 135)
F_c	correction factor for sum of calculated free convection and radiation, dimensionless, (see page 138)
F_e	radiation emissivity factor, dimensionless, (see equation VI-3, page 135)
F_{rd}	correction factor for calculated radiation, dimensionless, (see equation VIII-42, page 305)
g	gravitational constant, 32.17 feet per (second)^2
g'	gravitational constant, 4.16 x 10^8 feet per (hour)^2
h	heat transfer coefficient, watts per square inch-°C
h'	heat transfer coefficient, Btu per hour-square foot-°F
H	heat transfer parameter, or effective heat transfer coefficient, watts per °C
h_L	frictional energy loss per unit weight of fluid, foot-pounds per pound

Symbol	Concept
hp	shaft (driving) horsepower of a blower
J	mechanical equivalent of heat, 778 foot-pounds per Btu
k	thermal conductivity, Btu per hour-square foot-°F per foot
K	thermal capacity, watt-minutes per °C
K	constant (see equation VI-8, page 151)
L	significant dimension, inches
L'	significant dimension, feet
m	exponent (see equation VI-10, page 153)
M	Mach number, (ratio of velocity to velocity of sound in fluid), dimensionless
n	exponent
n	number of unspecified item in a group of similar items
N	rotational speed, revolutions per minute
p	absolute static pressure, inches of mercury, or as static gage pressure (subscript g) and as velocity pressure (subscript v) in inches of water
p'	absolute pressure, pounds per square foot
Δp	static pressure rise or drop, inches of water
P	absolute total pressure of fluid in motion, inches of mercury, or as total gage pressure (subscript g) in inches of water
ΔP	rise or drop of total pressure, inches of water
Pr	Prandtl number, $c_p\mu/k$, dimensionless
q	heat rate, watts
q'	rate of heat transfer, Btu per hour
$\bar{q}$	heat rate, Btu per second
Q	volume flow rate, cubic feet per minute
Q"	volume flow rate, cubic feet per second
r	radius of duct or distance, in inches, from center of circular duct to geometric mean of a sub-area number n (subscript n)

Symbol	Concept
Re	Reynolds number, $WL'/(\mu A')$, dimensionless
S	surface area, square inches
S'	surface area, square feet
t	temperature, °C
t'	temperature, °F
T	absolute temperature, °K
T'	absolute temperature, °F absolute or °R
Tq	torque, pound-feet
u	fluid velocity, feet per second
v	flight speed, miles per hour
W	weight flow rate, pounds per second
wk	rate of mechanical work, watts
$\overline{wk}$	rate of mechanical work, Btu per second
x	length, inches
y	number of equal sub-areas in the division of a cross-sectional area
Z	height of fluid column, inches
Z'	potential energy, foot-pounds per pound of fluid, or feet of fluid column
ΔZ	manometer head, inches of fluid
$\Delta Z'$	manometer head, feet of fluid
α	physical property factor for air, dimensionless, (see equation VI-9, page 151)
β	coefficient of thermal expansion, per °F absolute, (for air 1/T')
Δ	difference of, or change of
η	efficiency, dimensionless
μ	absolute viscosity, pounds per foot-hour or pound-seconds per square foot
ρ	density, pounds per cubic foot
σ	air density ratio referred to standard density of 0.0765 pound per cubic foot, (see Figure V-8)
Σ	sum of
τ	time, minutes

Symbol	Concept
Φ_1	radiation factor, a dimensionless function of F_A and F_e, (see equation VI-4, page 135)
Φ_2	radiation factor, a function of the absolute temperatures of two surfaces in radiant heat exchange, watts per square inch, (see equation VI-4, page 135)

Subscript	Refers to
$\underline{0}$	initial condition of operation
1	a station in a flow system, usually equipment inlet
2	a station in a flow system, usually equipment outlet
1, 2, 3,.....n	number of one of group of similar items
a	air
ab	absorbed heat
av	average value
b	baffle surrounding a unit
B	blower
bar	barometric pressure
c	equipment case
cd	conduction heat transfer
ch	conditions in plenum chamber
corr	value corrected to a standard condition
cp	basic electronic component
cv	free convection heat transfer
$\underline{cv}$	forced convection heat transfer
d	conditions at blower discharge
db	dry-bulb conditions
ds	dissipated heat

Subscript	Refers to
e	external equipment surface
ef	effective component temperatur
eq	thermal equilibrium conditions
ev	equivalent value
ex	heat exchanger
f	average fluid film temperature
fl	fluid properties
g	gage pressure value
hz	horizontal position
i	conditions at blower inlet
ins	value obtained in test where equipment has external thermal insulation
it	internal equipment conditions
m	motor
n	one unspecified item among a group of similar items
o	ambient conditions
r	reference conditions
rc	received heat
rd	radiation heat transfer
rp	representative value
s	static conditions
sf	surface
t	total conditions
v	velocity (pressure)
vt	vertical position
w	environmental walls
wb	wet-bulb conditions
wd	saturated water vapor at dry-bulb temperature
wv	superheated water vapor
ww	saturated water vapor at wet-bulb temperature

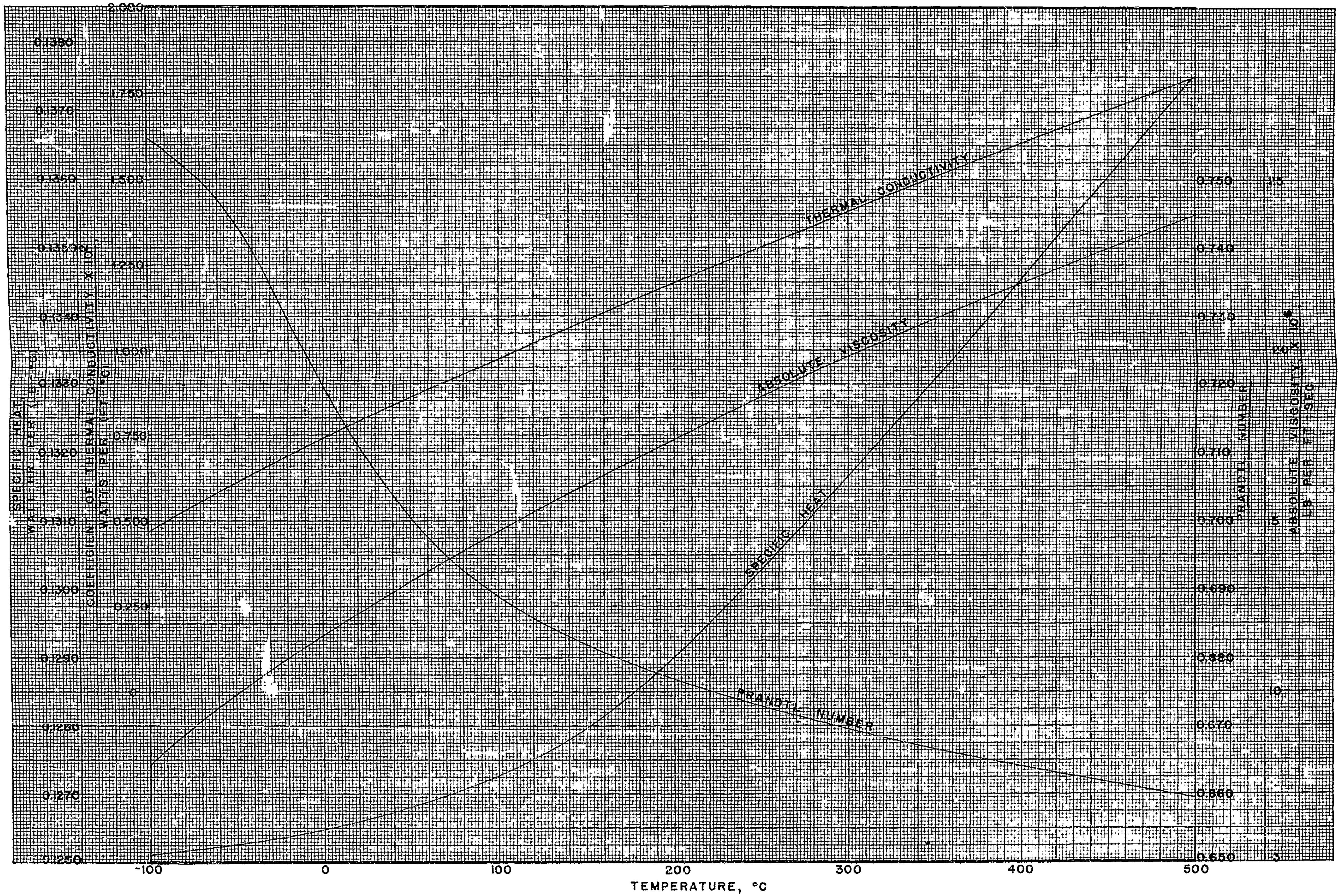


FIGURE A-I-1. PHYSICAL PROPERTIES OF DRY AIR

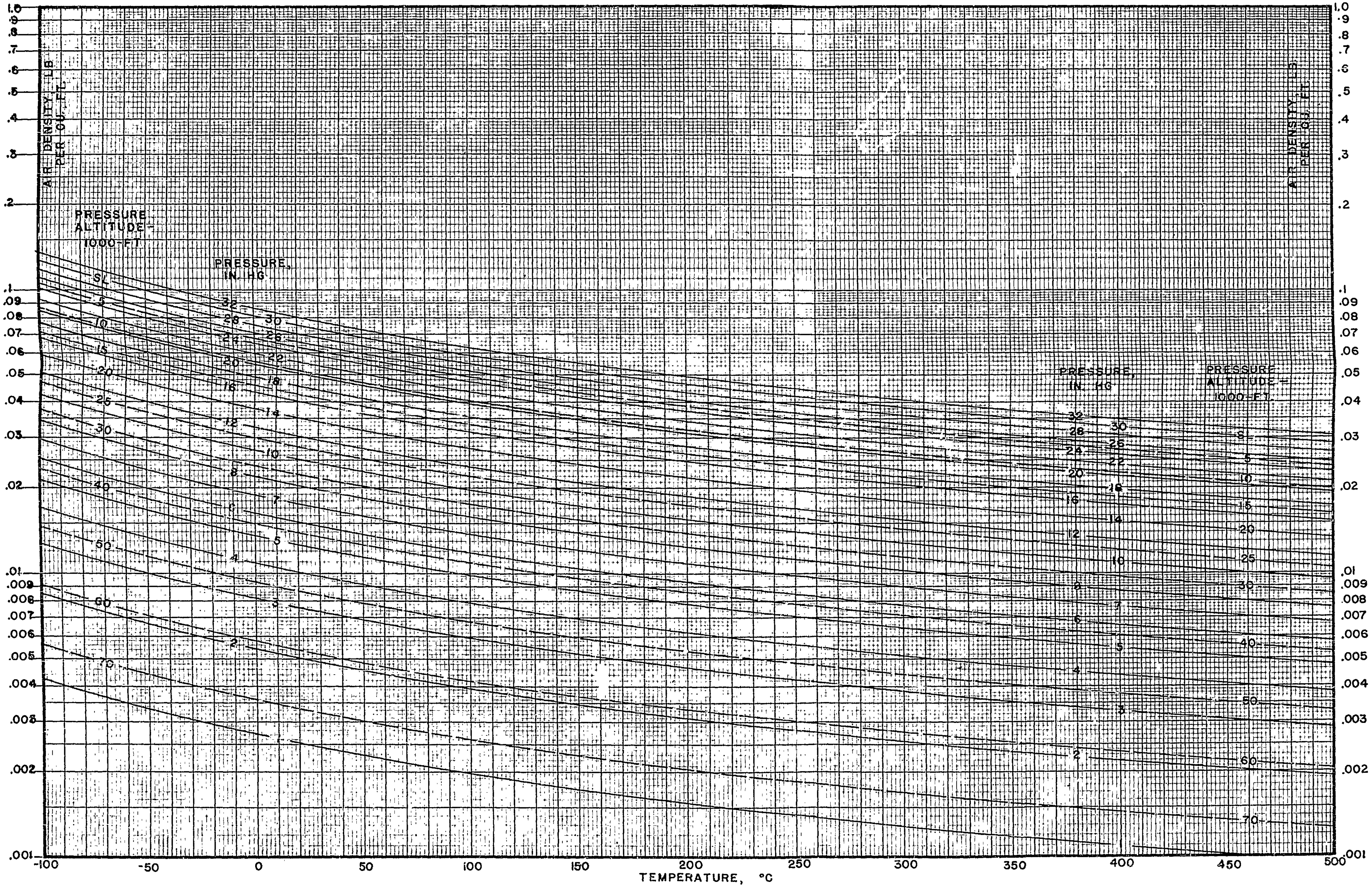


FIGURE A-I-2. DENSITY OF DRY AIR

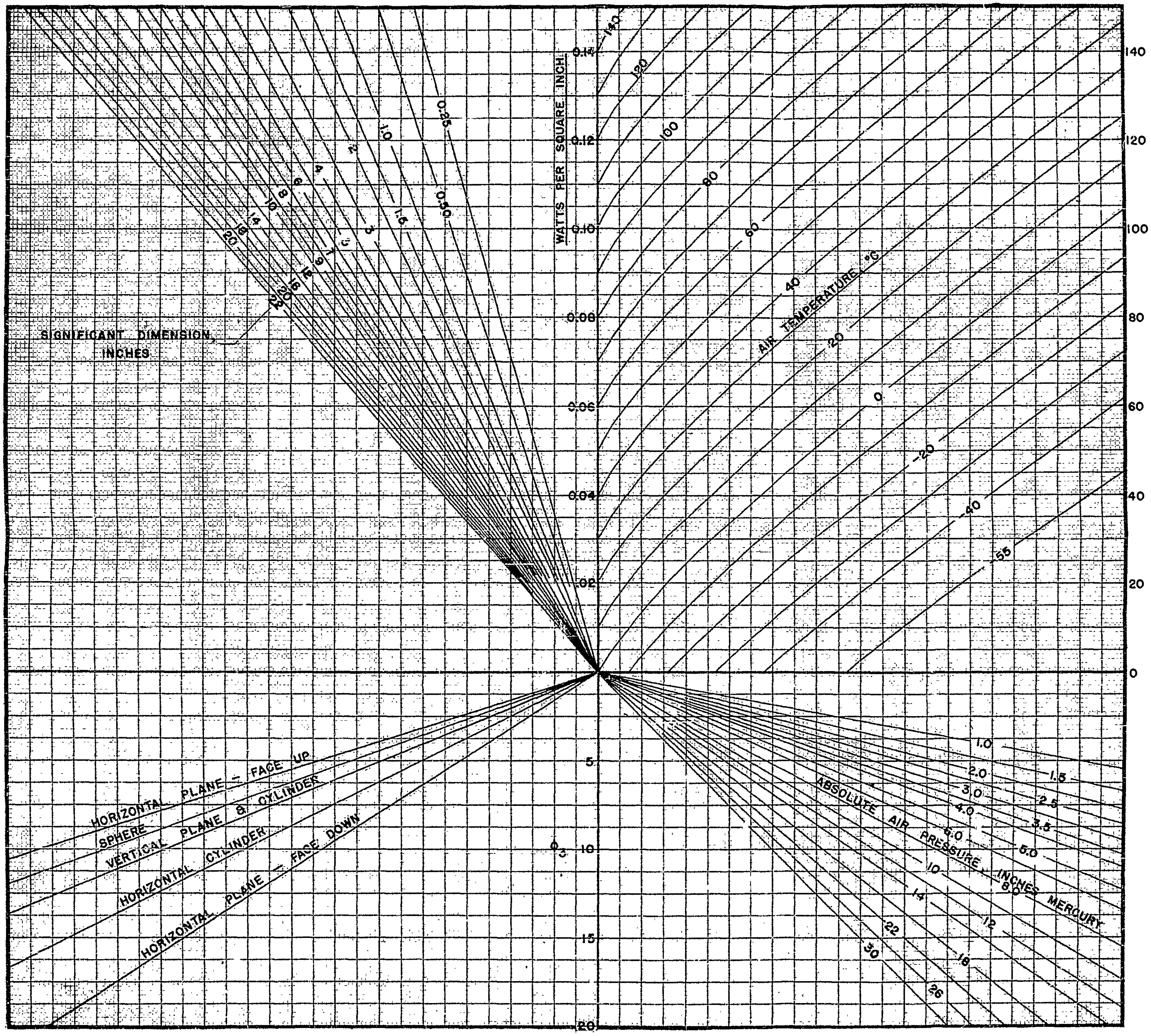
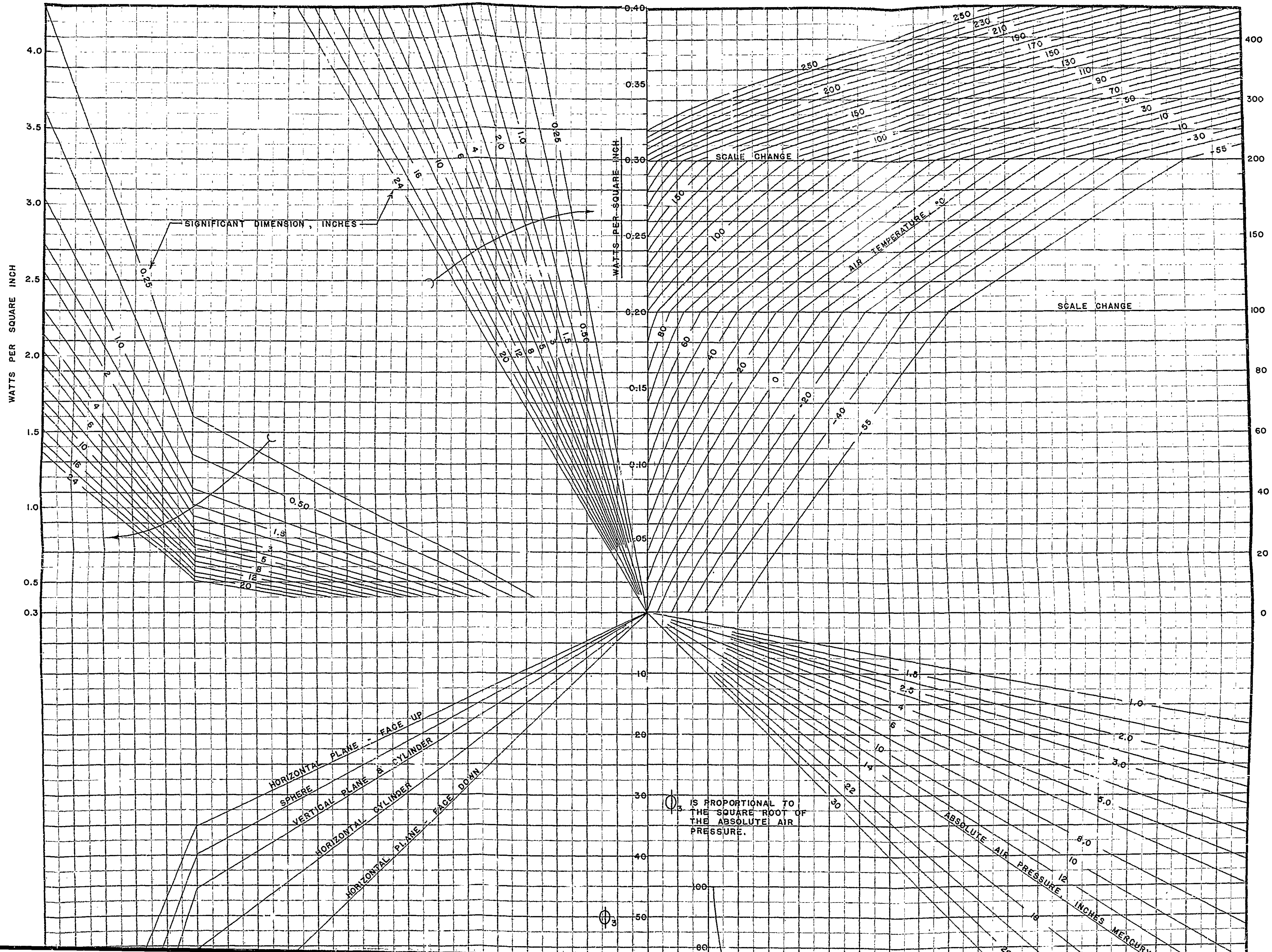
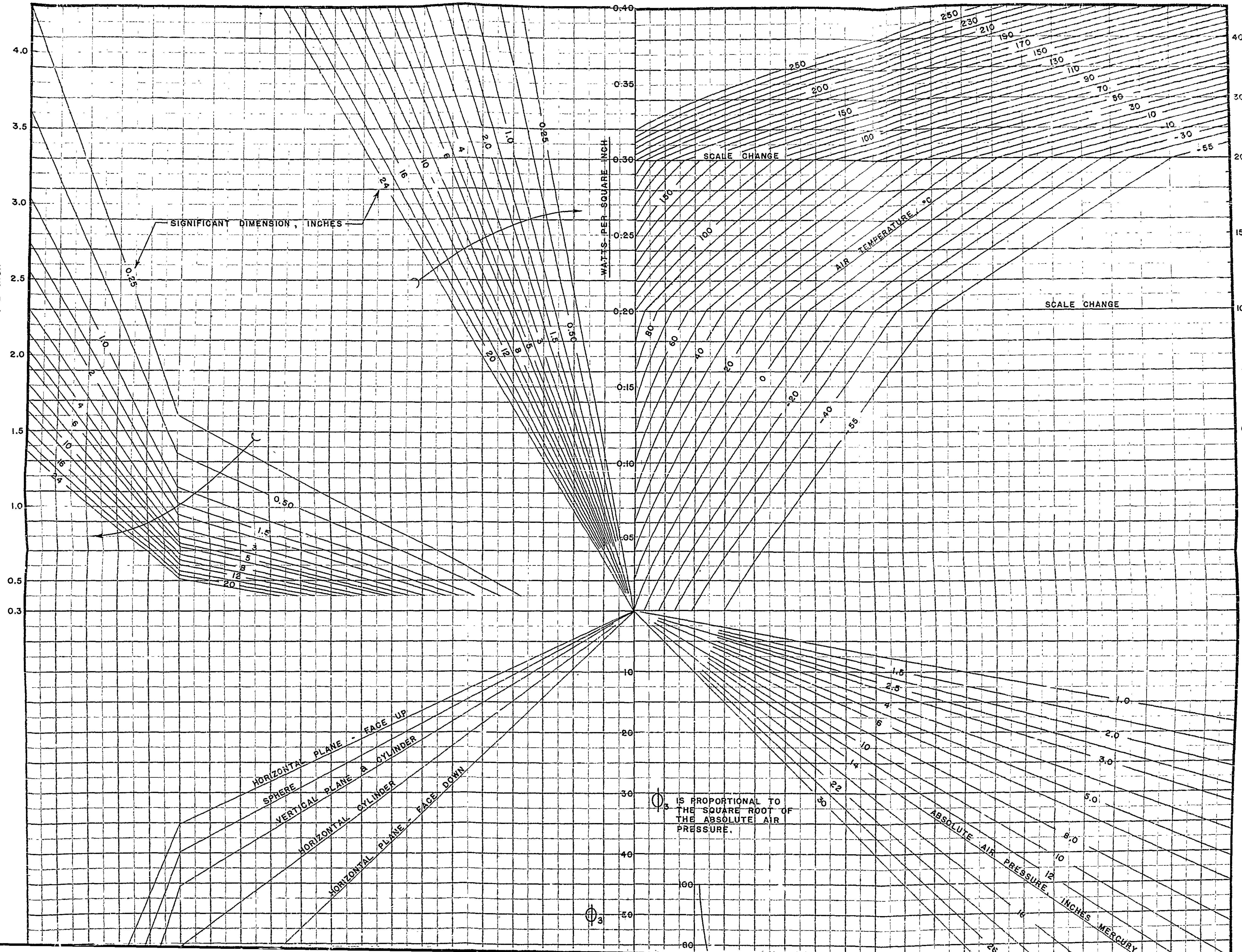


FIGURE VI-2b. CHART FOR CALCULATION OF FREE CONVECTIVE HEAT DISSIPATION IN AIR



WATTS PER SQUARE INCH
SIGNIFICANT DIMENSION, INCHES
WATTS PER SQUARE INCH
SCALE CHANGE
AIR TEMPERATURE, °C
SCALE CHANGE
SURFACE TEMPERATURE, °C
HORIZONTAL PLANE - FACE UP
SPHERE
VERTICAL PLANE & CYLINDER
HORIZONTAL CYLINDER
HORIZONTAL PLANE - FACE DOWN
Φ_3 IS PROPORTIONAL TO THE SQUARE ROOT OF THE ABSOLUTE AIR PRESSURE.
ABSOLUTE AIR PRESSURE, INCHES MERCURY
Φ_3



WATTS PER SQUARE INCH
SIGNIFICANT DIMENSION, INCHES
WATTS PER SQUARE INCH
SCALE CHANGE
AIR TEMPERATURE, °C
SCALE CHANGE
SURFACE TEMPERATURE, °C
HORIZONTAL PLANE - FACE UP
SPHERE
VERTICAL PLANE & CYLINDER
HORIZONTAL CYLINDER
HORIZONTAL PLANE - FACE DOWN
ϕ_3 IS PROPORTIONAL TO THE SQUARE ROOT OF THE ABSOLUTE AIR PRESSURE.
ABSOLUTE AIR PRESSURE, INCHES MERCURY
ϕ_3

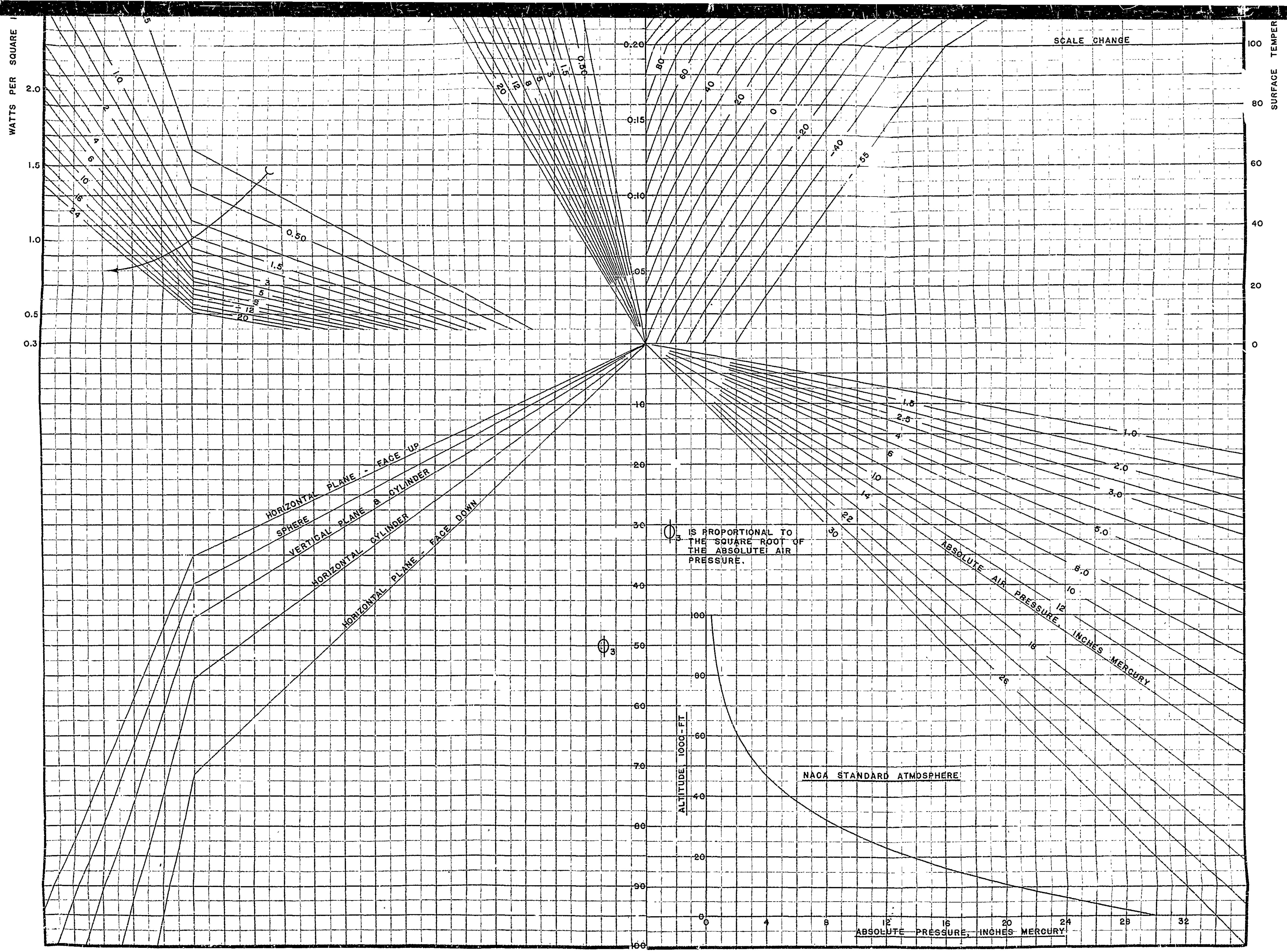
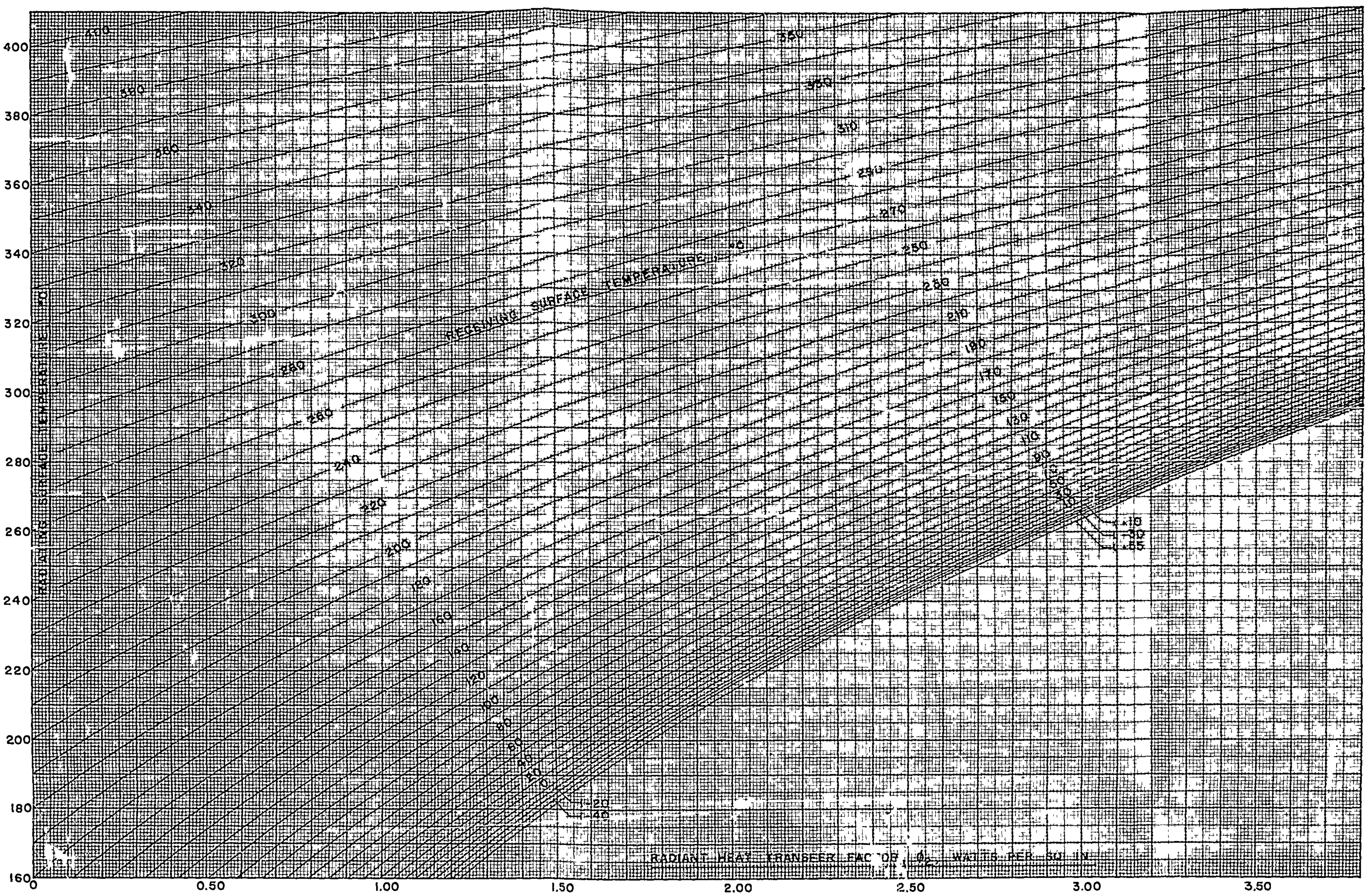


FIGURE VI-2a. CHART FOR CALCULATION OF FREE CONVECTIVE HEAT DISSIPATION IN AIR



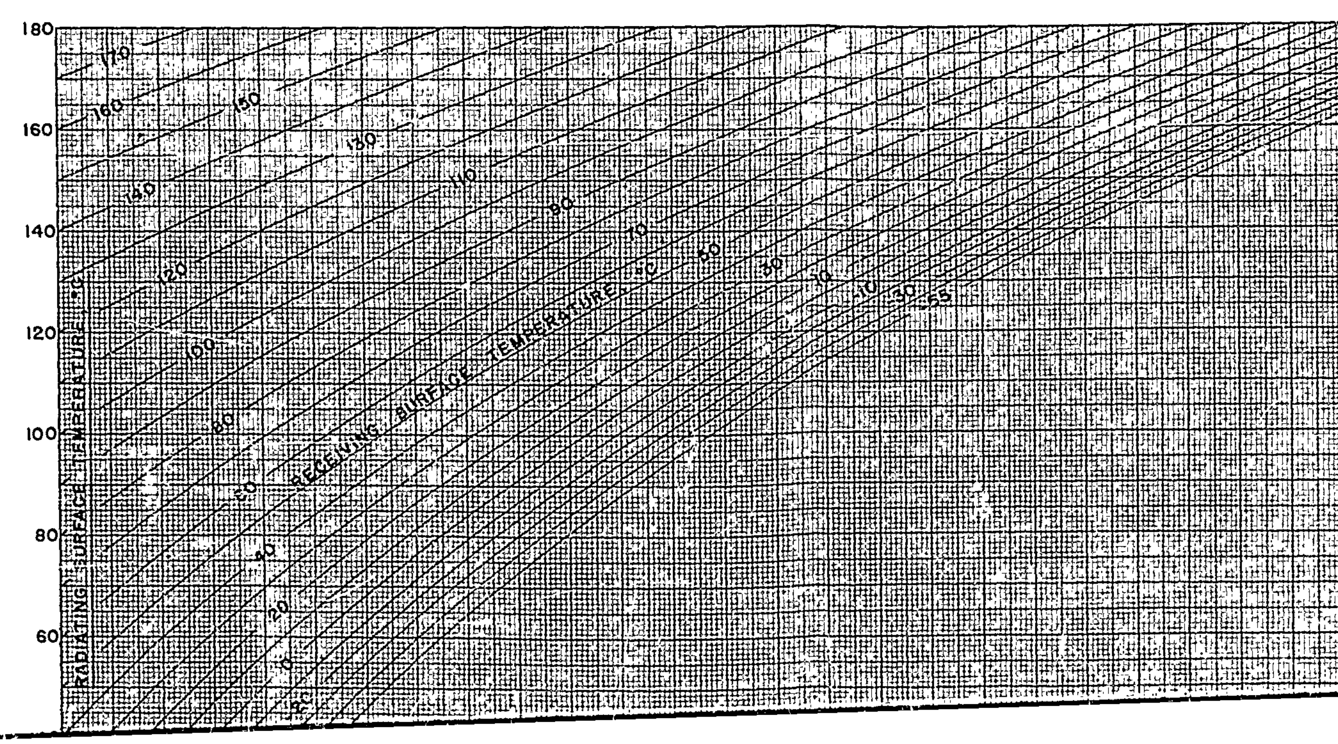
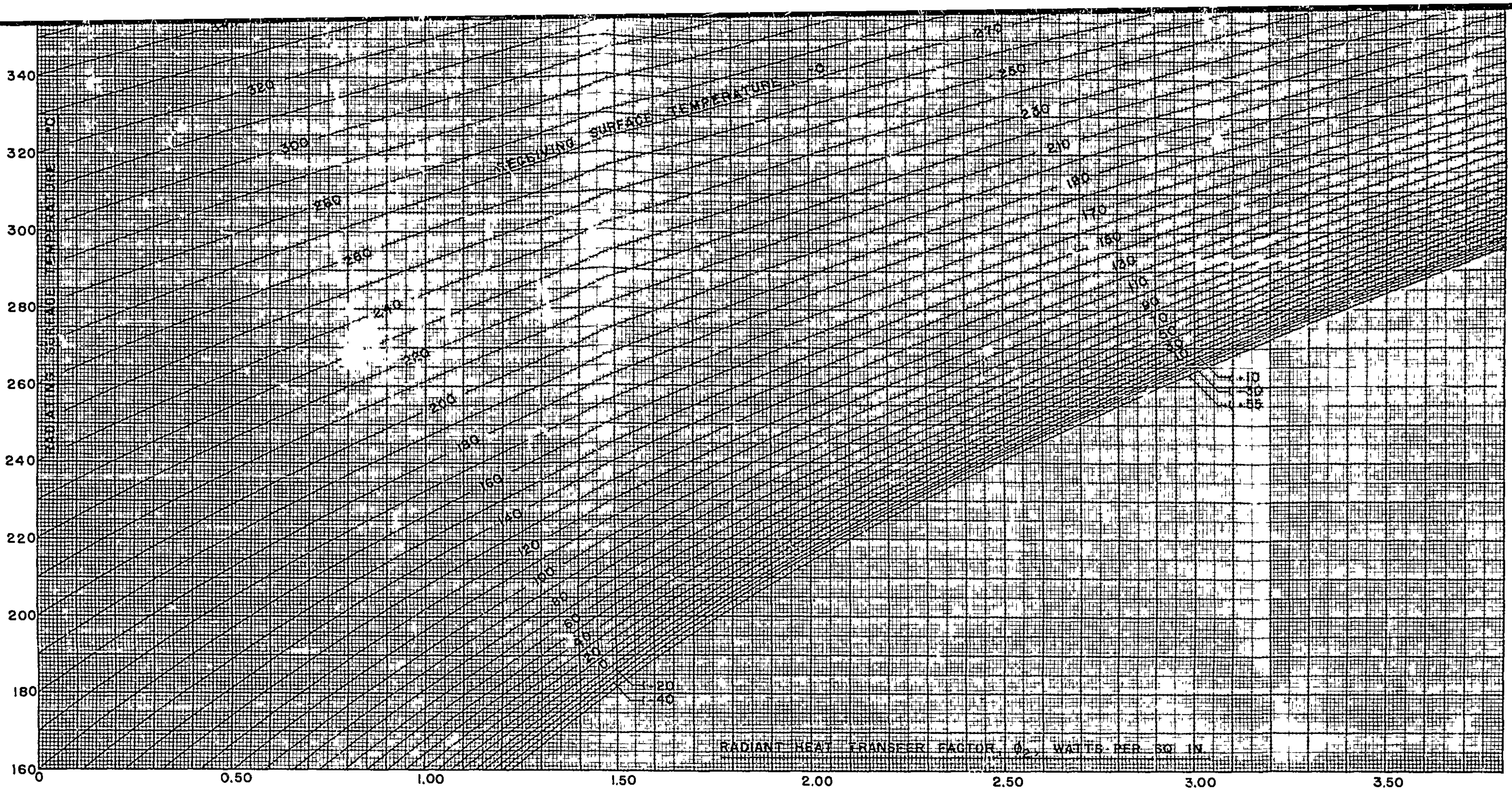


FIGURE VI-3a. (above)
3b. (left)

CHARTS FOR CALCULATION OF RADIANT HEAT DISSIPATION



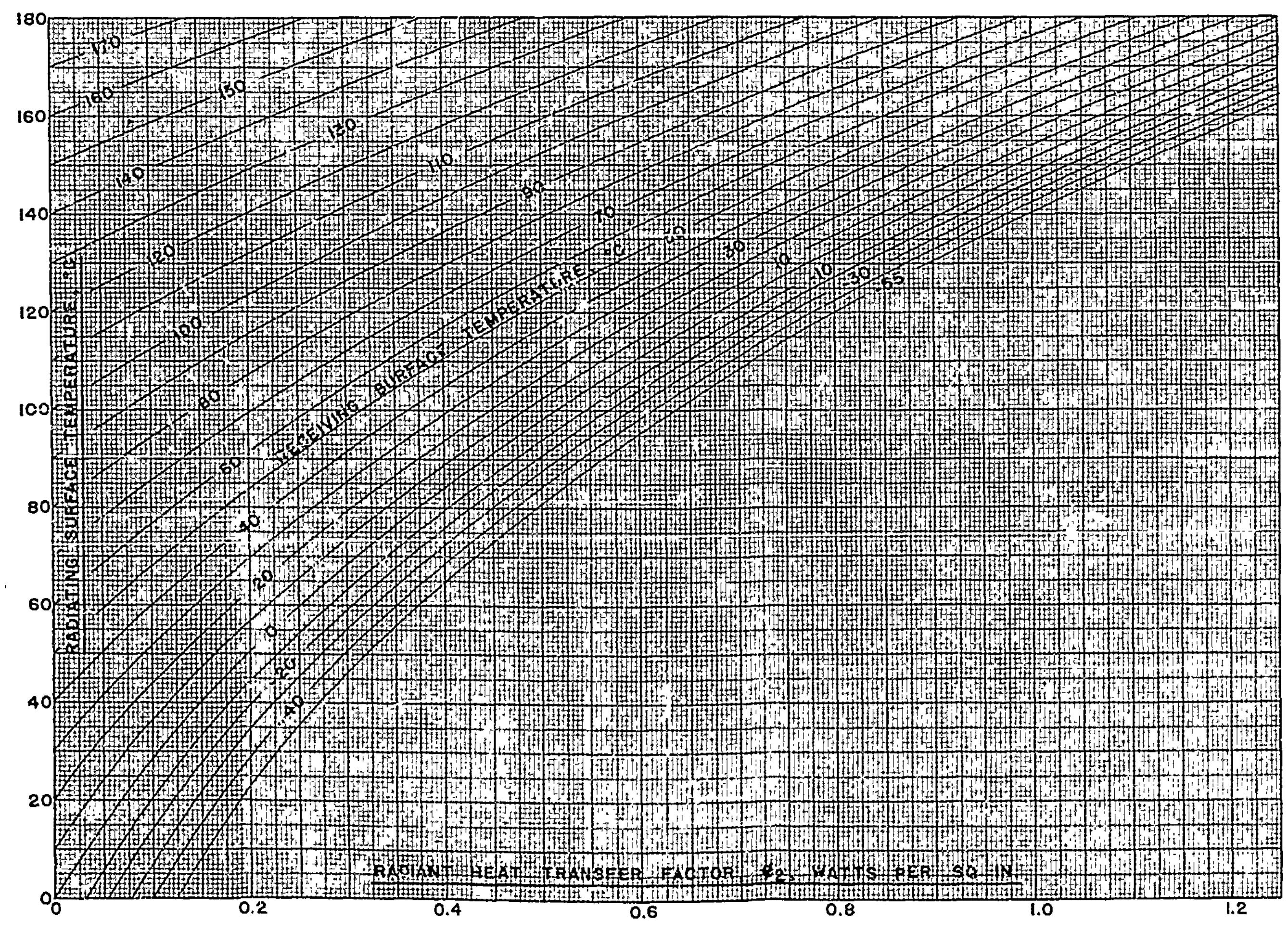


FIGURE VI-3a. (above)
3b. (left)

CHARTS FOR CALCULATION OF RADIANT HEAT DISSIPATION

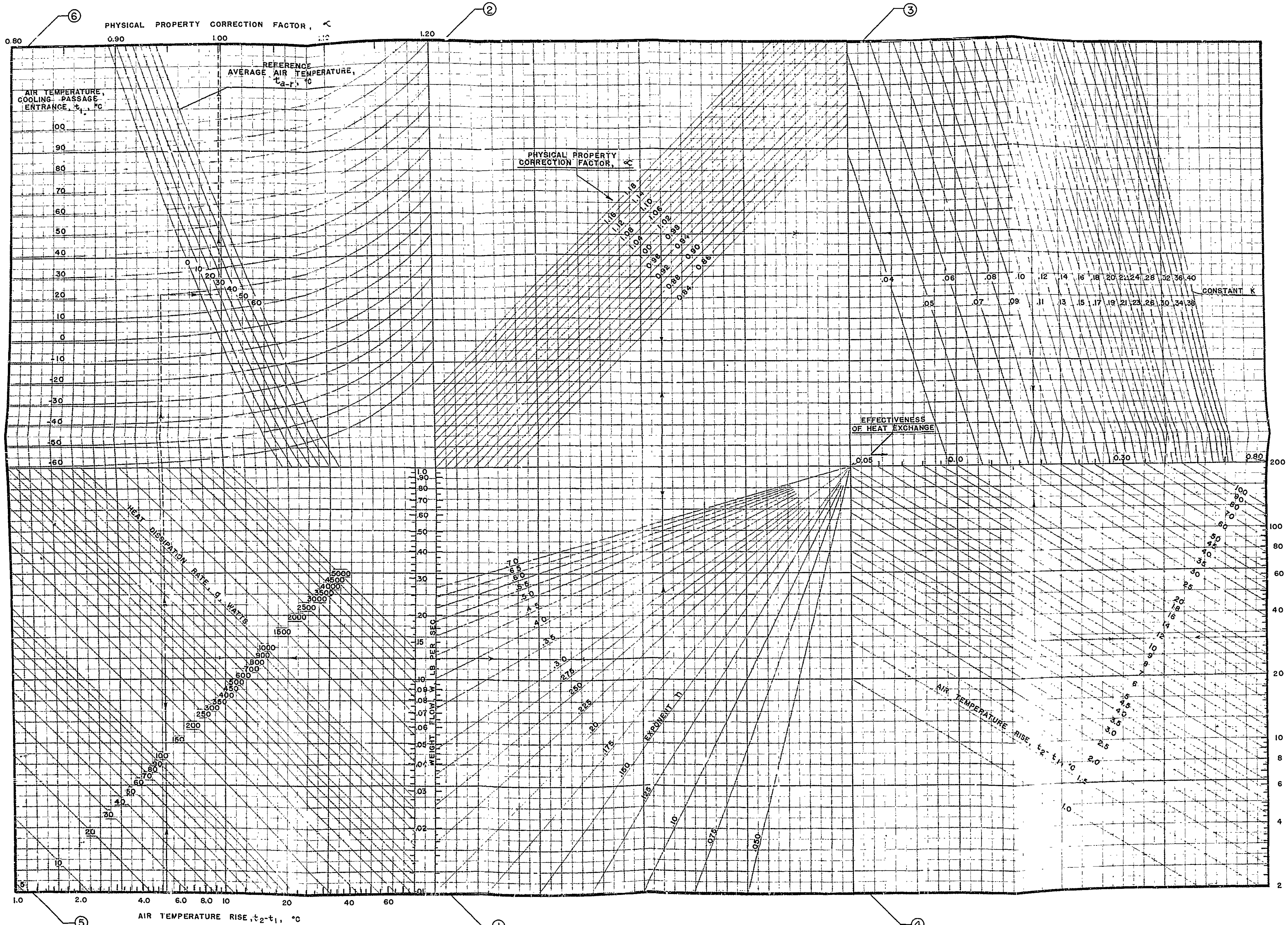


FIGURE VI - 7. HEAT TRANSFER CHART FOR FORCED CONVECTIVE AIR COOLING

AFTR-6579

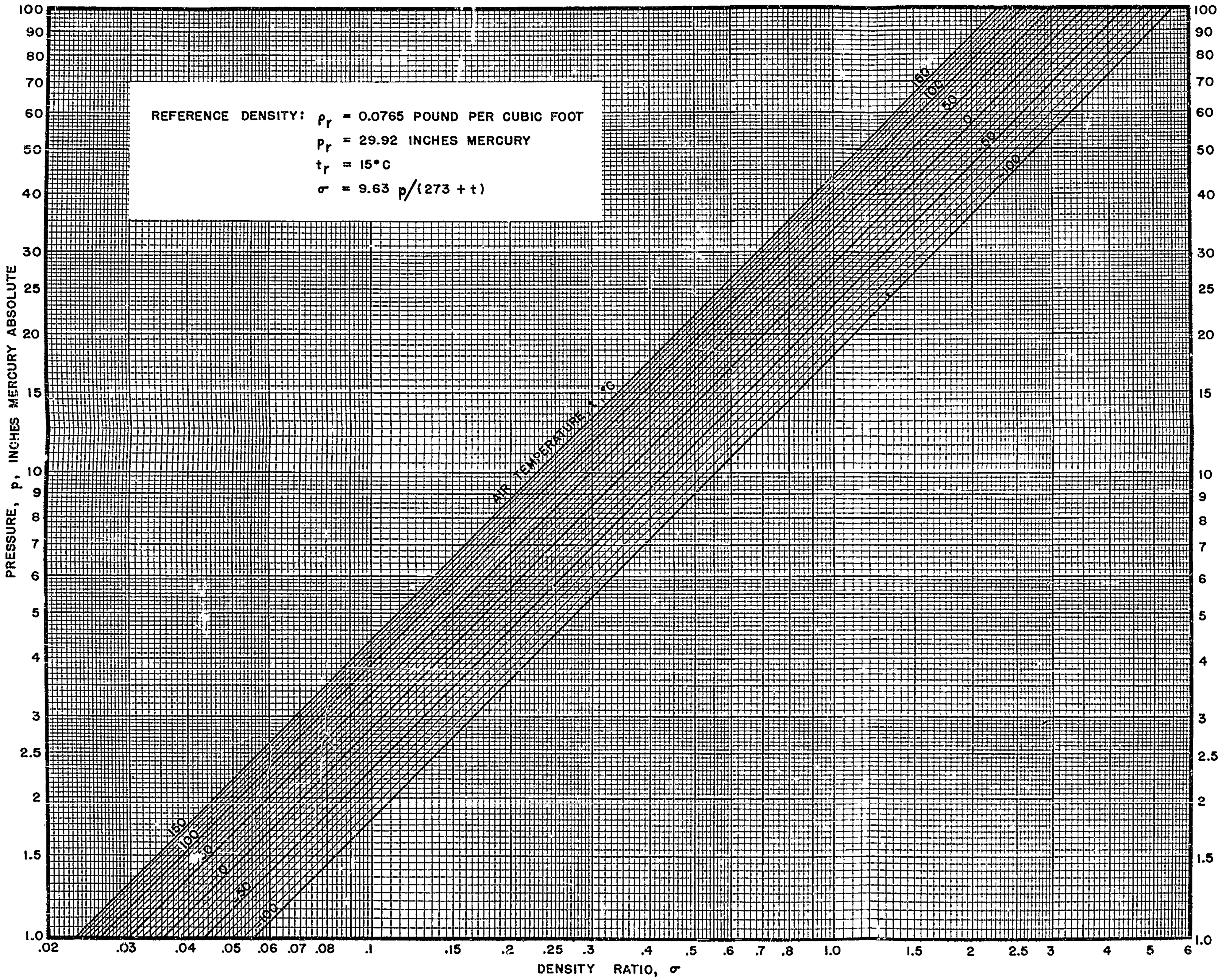


FIGURE V-8. DENSITY RATIO CHART FOR AIR

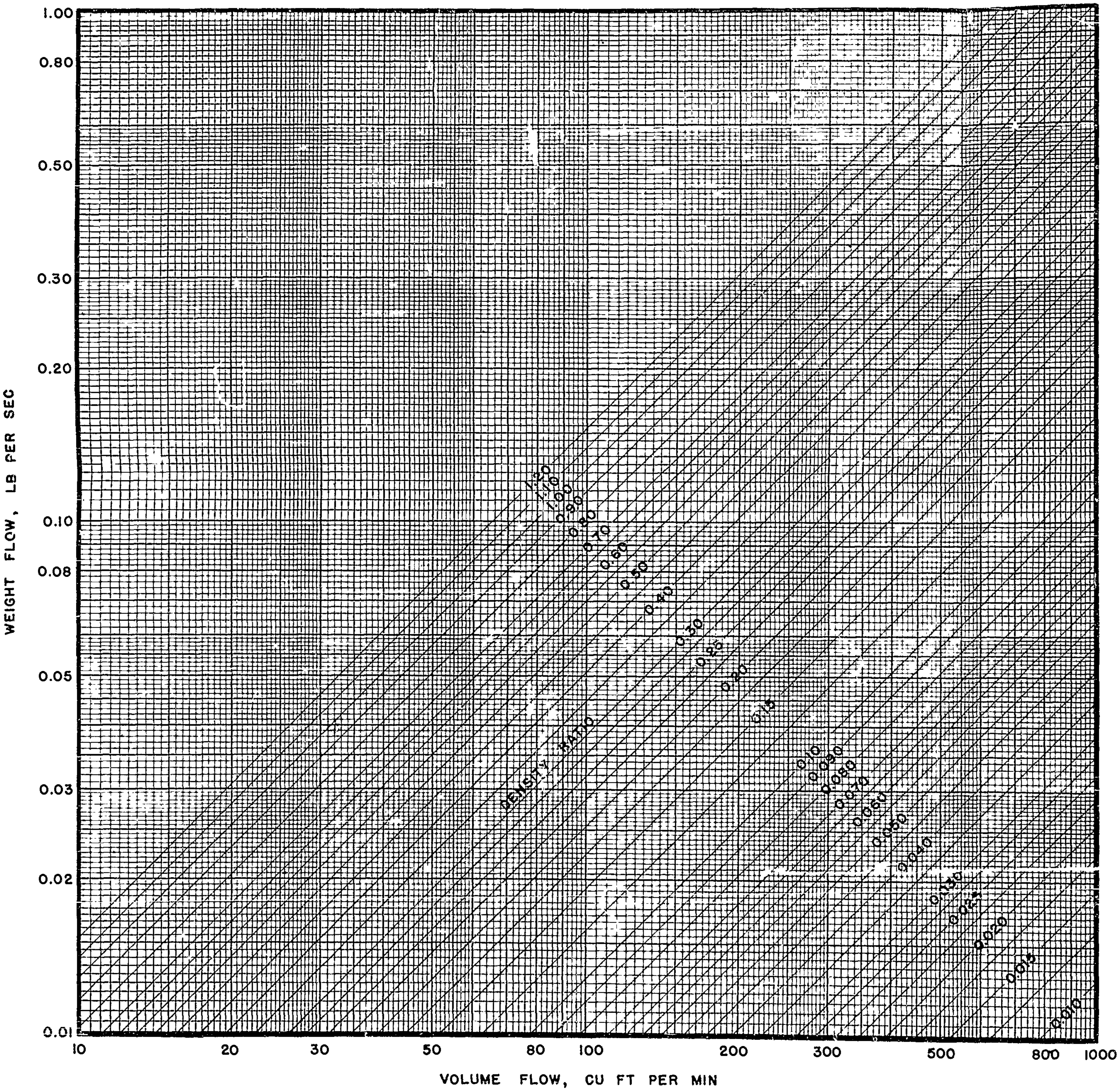


FIGURE V-9. VOLUME-WEIGHT FLOW CONVERSION CHART FOR AIR

AFTR-6579

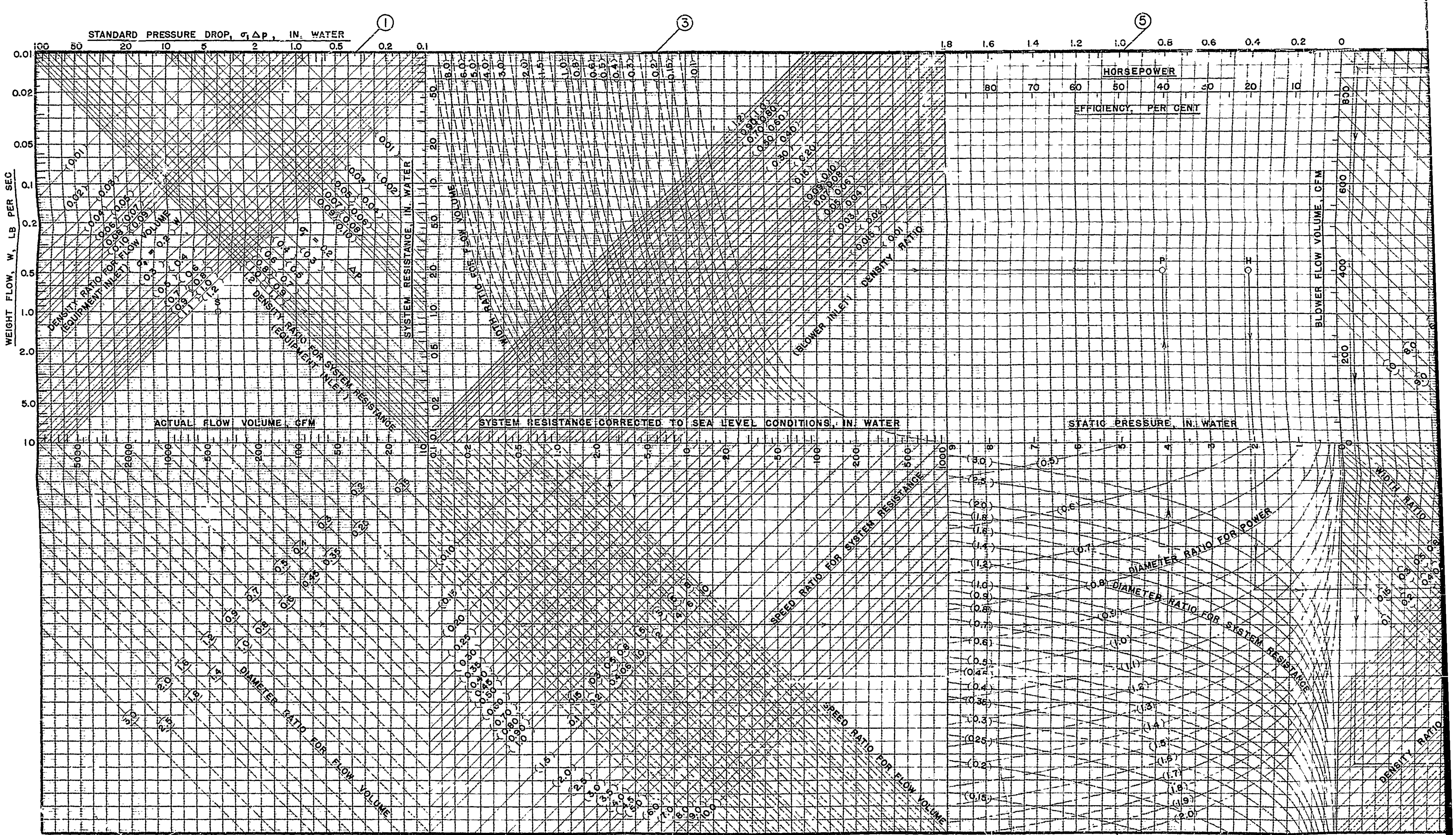


FIGURE V-12. WORKING CHART FOR BLOWER PERFORMANCE ANALYSIS AND DESIGN

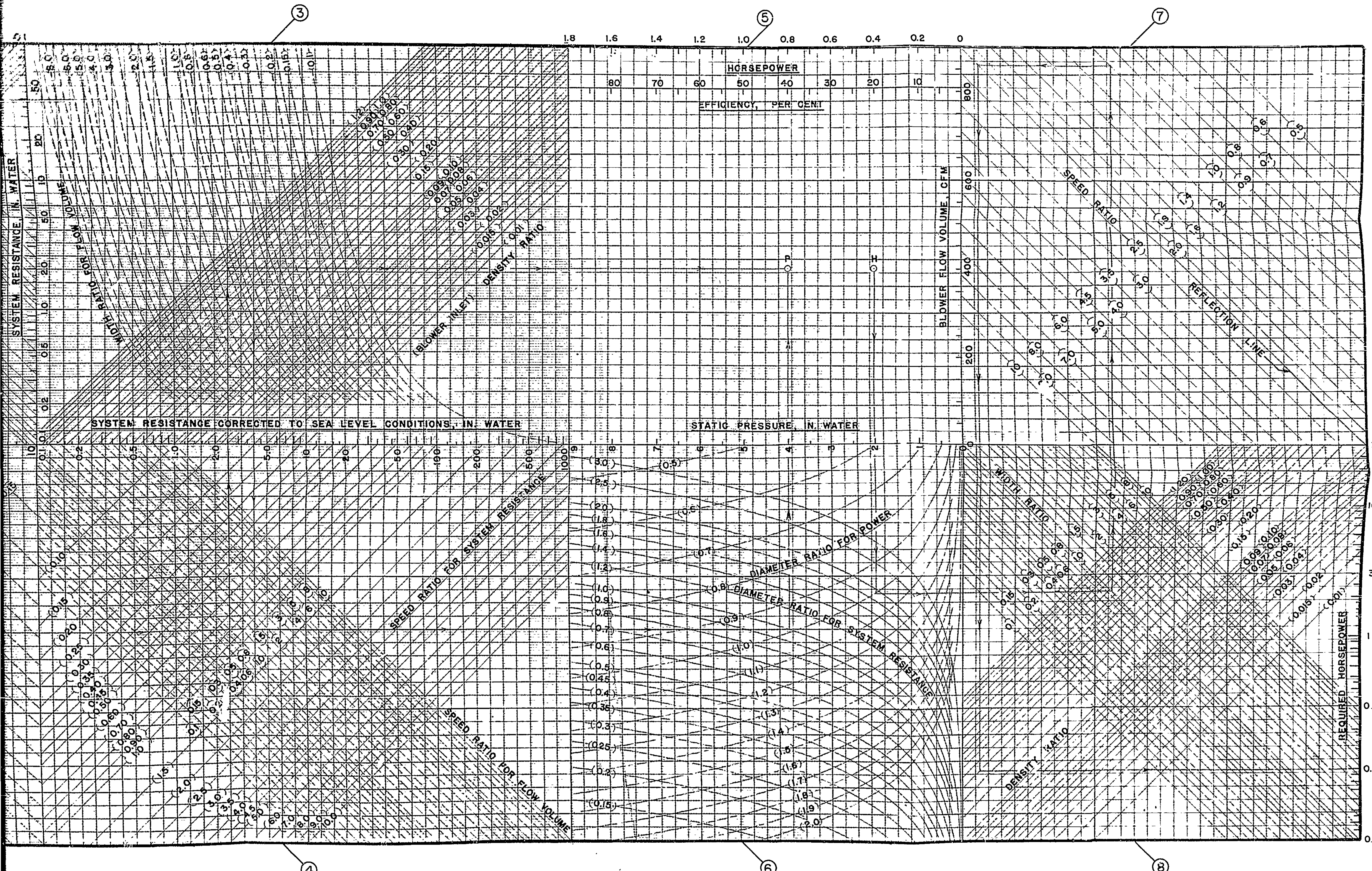


FIGURE V-12. WORKING CHART FOR BLOWER PERFORMANCE ANALYSIS AND DESIGN

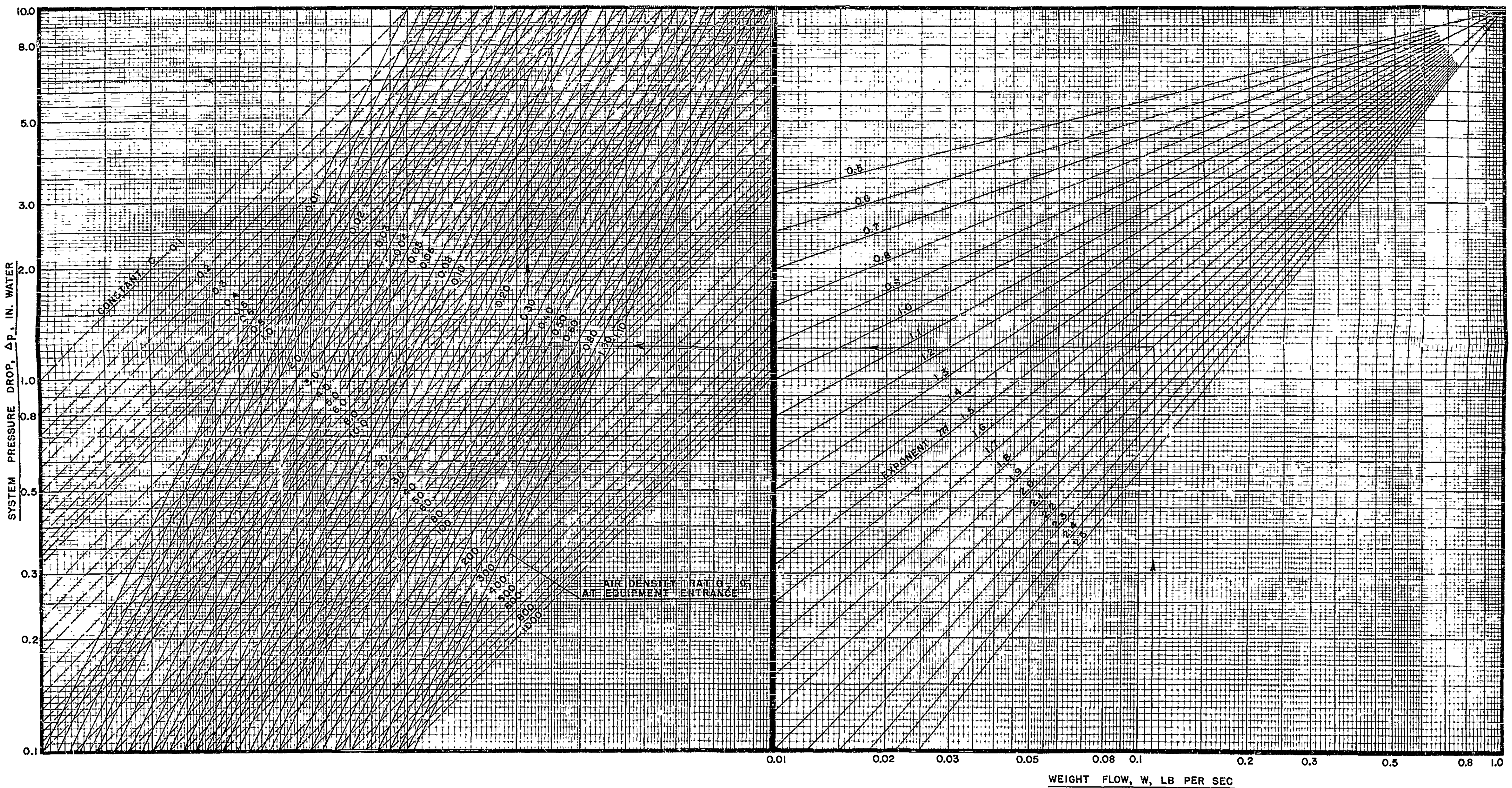
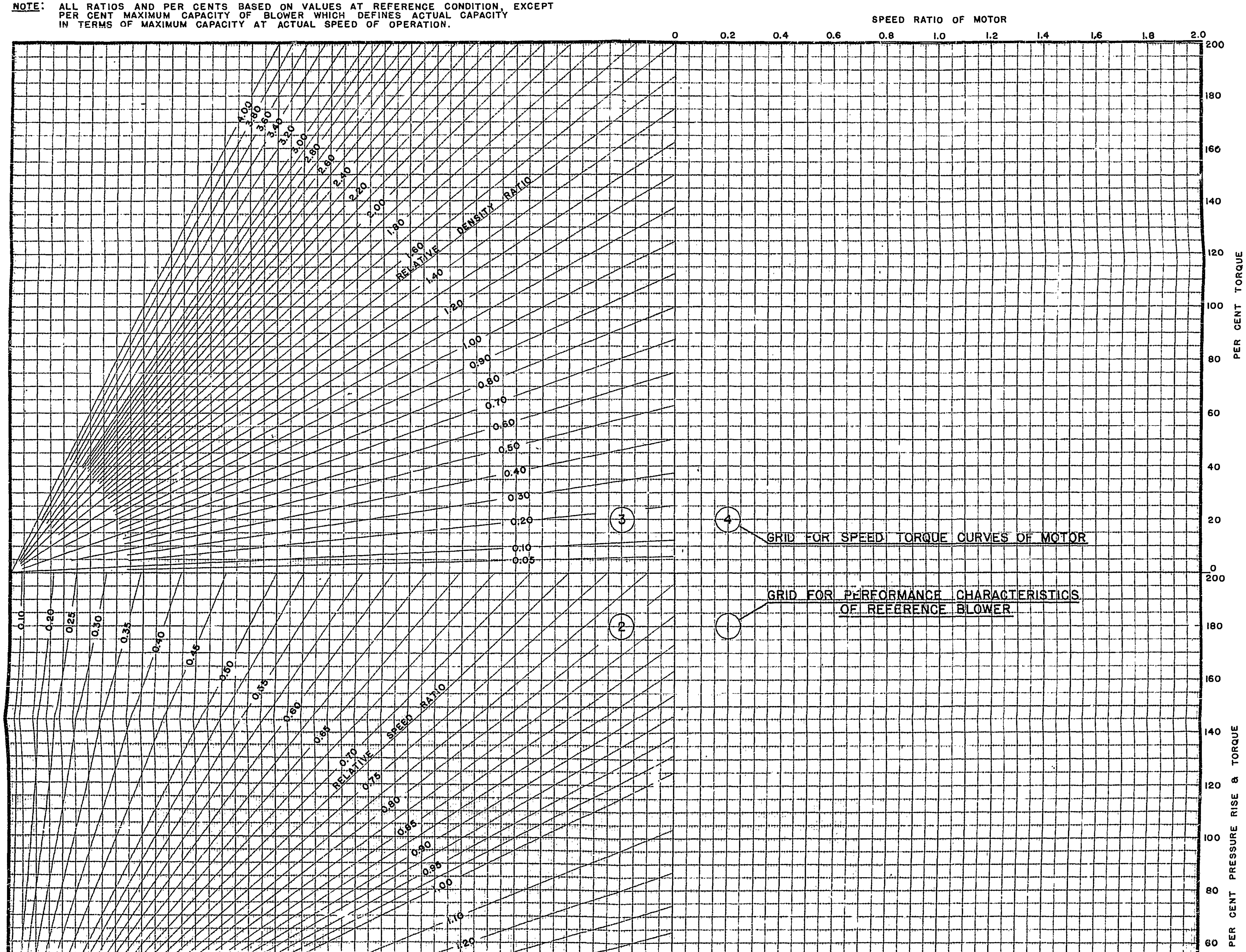


FIGURE VI-9. SYSTEM RESISTANCE CHART FOR FORCED CONVECTIVE AIR COOLING

NOTE: ALL RATIOS AND PER CENTS BASED ON VALUES AT REFERENCE CONDITION, EXCEPT PER CENT MAXIMUM CAPACITY OF BLOWER WHICH DEFINES ACTUAL CAPACITY IN TERMS OF MAXIMUM CAPACITY AT ACTUAL SPEED OF OPERATION.



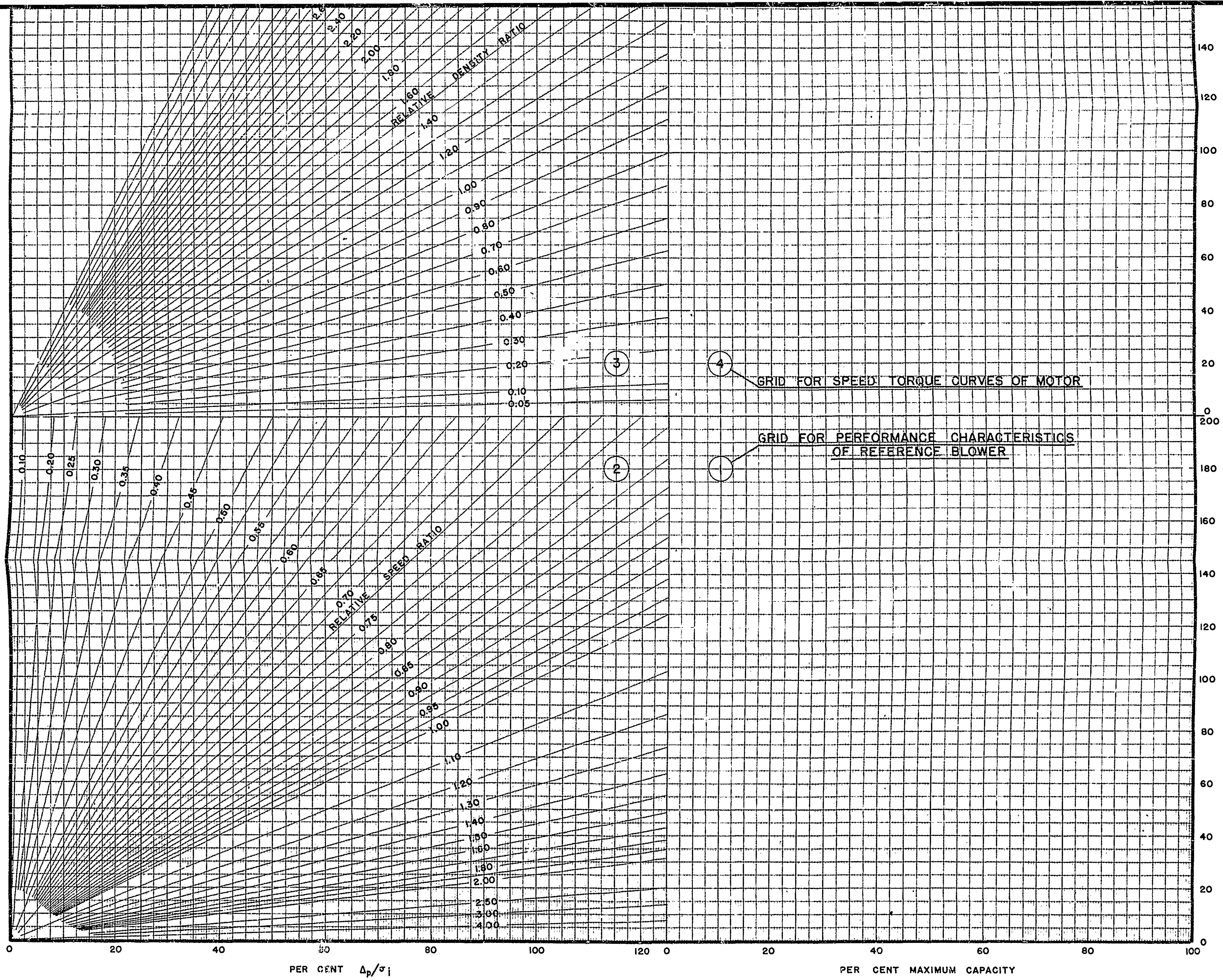


FIGURE V-19. CHART FOR PERFORMANCE ANALYSIS OF BLOWER-MOTOR UNITS

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THE THERMAL EVALUATION OF AIR-COOLED ELECTRONIC EQUIPMENT - AND APPENDIXES I THRU VI, by Walter Robinson and R.H. Zimmerman. Sep '52, 408 pp incl. photos, tbs, diagrs, graphs, dwgs, charts. UNCLASSIFIED

Air-cooled electronic equipments designed for airborne application are classified according to their cooling methods. Basic thermal test methods and techniques of making the necessary measurements are described with emphasis on bench testing.

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DIVISION: Electronics (3)
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1. Electronic equipment - Cooling
2. Fans, Cooling
I. Robinson, Walter
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III. USAF Contr. No. W33-038-ac-14987

Fans
Air Cooled

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Calculation methods for the prediction of altitude performance from bench-test data are presented. The analysis of altitude-chamber test data and calculation methods for their correction are discussed for equipment which cannot be evaluated by bench tests only, and for other equipments for which certain data may be conveniently secured by altitude-chamber test. Methods applicable to various types of equipment for evaluation of nonsteady-state operation are covered in detail. One chapter is devoted to the theory, performance, evaluation and selection of cooling blowers. Throughout the report, charts are presented which allow graphical means to be utilized to a great extent for purposes of analysis. All methods are illustrated by means of examples. Physical property data, air flow theory, blower test methods and details of experimental apparatus are presented.

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