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STUDY, HIGH PERFORMANCE CABIN COOLING UNIT

ARTHUR P. LANE

STRATOS DIVISION
FAIRCHILD ENGINE & AIRPLANE CORPORATION

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FOREWORD

This report was prepared by Arthur P. Lane of the Stratos Division, Fairchild Engine and Aircraft Company, Bay Shore, New York, on Air Force Contract AF 33(600)-23989, under supplementary Project 664-803-4, "Study of High Performance Cooling Unit". The High Performance Cooling Unit Study was conducted under the direction of Mr. H. K. Ziebarth, Mechanical Branch, Equipment Laboratory, acting as project engineer.

Acknowledgment is made to the following Stratos Division personnel who cooperated in the study: Messrs. D. O. Moeller, Chief Engineer; J. Makowski, Chief Research Engineer; R. B. Barclay; G. K. Fischer; I. D. Miller; V. L. Whitney, Research Engineers.

ABSTRACT

A study of the problems involved in designing a high performance air cycle cabin conditioning system for jet airplanes is discussed. The performance data used in the study pertain to bomber airplanes in present use. As a result of the study, a high performance cooling system has been selected, based on the specified operating conditions. The system is a regenerative bootstrap incorporating a split regenerative heat exchanger.

The prediction of the capabilities of the system necessitated the development of an analytical method for determining the performance of a heat exchanger when moisture is condensed from the cooled air. Such a method is given together with a description of the test program undertaken to assure the reliability of the analytical method.

PUBLICATION REVIEW

The publication of this report does not constitute approval by the Air Force of the findings or the conclusions contained therein. It is published only for the exchange and stimulation of ideas.

FOR THE COMMANDER:

S. T. SMITH
Colonel, USAF
Chief, Equipment Laboratory
Directorate of Laboratories

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SYMBOLS

A_c, A_h	Heat transfer surface, for cooling side, for bleed side - sq ft
A_f	Flow area - sq ft
A_s	Heat transfer or diffusion surface - sq ft
C_d	Nozzle discharge coefficient - dimensionless
c_p	Specific heat at constant pressure - BTU/lb^{o}F
c_t	Turbine tip speed - ft/sec
D	Diameter - feet
F	Compressor flow factor - dimensionless
g	Acceleration of gravity - ft/sec^2
G	Unit weight flow - lb/hr sq ft
h, h_c, h_h	Heat transfer coefficient in general, for cooling side, for bleed side - BTU/hr sq ft ^{o}F
h_e, h_{e_c}, h_{e_h}	Effective film coefficient in general, for cooling side, for bleed side - BTU/hr sq ft ^{o}F
H	Enthalpy - BTU/lb
H_c	Condensation rate - lbsH_2O/lb of fluid flowing
H_i	Bleed air inlet humidity to cooling package - gr/lb
j_d	Modulus for diffusion - dimensionless
j_h	Modulus for heat transfer - dimensionless
J	Mechanical equivalent of heat - 778 ft lb/BTU
k	Thermal conductivity - BTU/hr sq ft ^{o}F/ft
k_d	Diffusivity - sq ft/hr
k_g	Diffusion coefficient - mol/hr sq ft atm
l	Fin length - feet
M_m	Mean molecular weight of vapor and inert gas - lb/mol
M_v	Molecular weight of vapor - lb/mol
N	Rotational speed - rpm
N_d	Number of mols diffused
Nm	Mach number - dimensionless
Nu	Nusselt number = $\frac{hD}{k}$ - heat transfer modulus - dimensionless
P, P_1, P_2	Partial pressures of vapor in general, at point 1, at point 2 - atm
P_c	Partial pressure of vapor at condensate film - atm
P_g, P'_g	Partial pressure of the gas in the gas body and at the condensate film - atm
P_{g_f}	Log mean pressure difference between P_g and P'_g - atm
Pr	Prandtl number = $\frac{c_p \mu}{k}$ - a fluid properties modulus - dimensionless
P_t	Total pressure of system - atm

SYMBOLS

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P_{g_f}	Log mean pressure difference between P_g and P'_g - atm
Pr	Prandtl number = $\frac{c_p \mu}{k}$ - a fluid properties modulus - dimensionless
P_t	Total pressure of system - atm

P_v	Partial pressure of vapor in gas stream - atm
Δp	Pressure difference of diffusing vapor between P_v and P_c - atm
q	Heat flow for interval - BTU/hr
Q	Total heat flow - BTU/hr
Re	Reynolds number - GD/μ - dimensionless
Sc	Schmidt number - $\mu/k_d\rho$ - dimensionless
t_1, t_2	Temperature at pts 1 and 2 - ^oF
t_s	Wall surface temperature - ^oF
t_w	Temperature of cooling flow - ^oF
Δt	Temperature difference - ^oF
T_g, T_c	Temperature of gas and condensate - ^oF
U	Overall coefficient of heat transfer - BTU/hr sq ft ^oF
U_{eff}	Effective overall coefficient of heat transfer - BTU/hr sq ft ^oF
V_o	Theoretical spouting velocity - ft/sec
w, w_b, w_c	Flow rate - in general, bleed, cooling flow - lb/min
W_s	Specific humidity - gr/lb
W_c	Condensed moisture - gr/lb
W_{sp}	Sprayed moisture - gr/lb
λ	Latent heat of vaporization - BTU/lb
μ	Viscosity - lb/ft hr
ρ	Density - lb/cu ft
η_x	Heat exchanger effectiveness, dry - dimensionless
η_{wet}	Heat exchanger temperature ratio parameter, wet - dimensionless
δ	Fin thickness - feet
η_o	Total surface fin efficiency - dimensionless
η_f	Finned surface fin efficiency - dimensionless
ϕ	Flow ratio - dimensionless
η_m	Mechanical efficiency - dimensionless
η_{sep}	Separator efficiency - dimensionless
η_{sp}	Spray efficiency - dimensionless
η_t	Turbine efficiency - dimensionless
η_c	Blower efficiency - dimensionless
π_c, π_t	Pressure ratio, compressor, turbine, - dimensionless
$\surd$	Velocity ratio - dimensionless

INTRODUCTION

The object of this report is to conduct an analytical and, where necessary, an experimental engineering study of air cycle cooling units, meeting the following requirements:

(1) High performance capability.
(2) Weight, bulk, and cooling air drag of the unit to be comparable to units presently used on bomber type airplanes.
(3) Total engine bleed air flow to be less than that required by the units presently used on bomber type airplanes.

The use of cooling units employing means for the removal of ambient moisture has been advanced as a method of developing a high performance cooling unit. The removal of moisture from the cabin air circuit will allow low turbine outlet air temperatures, since there can be no possibility of ice blocking the turbine discharge duct. Under these conditions cooling capacity could be increased.

Other benefits from the removal of water include the reduction of fog dispersing into the cabin from the inlet ducts. In many instances, under high humidity conditions, fine particles of moisture condense during expansion of bleed air in the cooling turbine. This vapor passes into the cabin as a fine fog, re-evaporating when conditions permit. Another advantage from the limitation of excess cabin humidity is the reduction of the danger of fogging of the cockpit windows.

The study that follows reviews some of the fundamentals behind aircraft air conditioning and compares the various systems that appear to have the best chance of being successful for the conditions specified. A selection of the most satisfactory system is made, based on study and test results, and a performance schedule is determined.

SECTION I

PROBLEMS INVOLVED IN HIGH PERFORMANCE COOLING UNIT DESIGN

The factors entering into the determination of the required refrigeration loads for a given flight-altitude schedule are as follows:

(1) Conduction and radiation heat transfer from the atmosphere and sun.
(2) Generation of heat by

(a) cabin occupants
(b) cabin equipment

A typical set of refrigeration requirements as a function of flight-altitude schedules for a present day bomber are shown in Figure 1.

The net cooling capacity of air is found by its change of heat content or enthalpy and is equal to

$$\int_1^2 dH = w \int_1^2 c_p dt = \text{refrigeration capacity} \qquad (1)$$

where

t = temperature air
c_p = specific heat of supply air
w = air weight flow
H = total enthalpy of air

(Points 1, 2, = conditions at supply inlet and cabin respectively.)

If proper mean values of specific heat are available, equation (1) can be simplified to:

$$\Delta H = w \, \overline{c_p} \, \Delta t \qquad (2)$$

where

$\overline{c_p}$ = mean specific heat of air.

Equations (1) or (2) can be used to determine required supply air temperature and air flow for a given refrigeration capacity.

HUMIDITY OPERATION

When moisture, present in the supply air as a vapor, condenses and is removed before introduction to the cabin, the heat of condensation equal to the latent heat of evaporation of water, is liberated. This enthalpy change serves to increase the "corrected" temperature of the refrigerated air. The "corrected" temperature is defined as the actual air temperature entering the cabin.

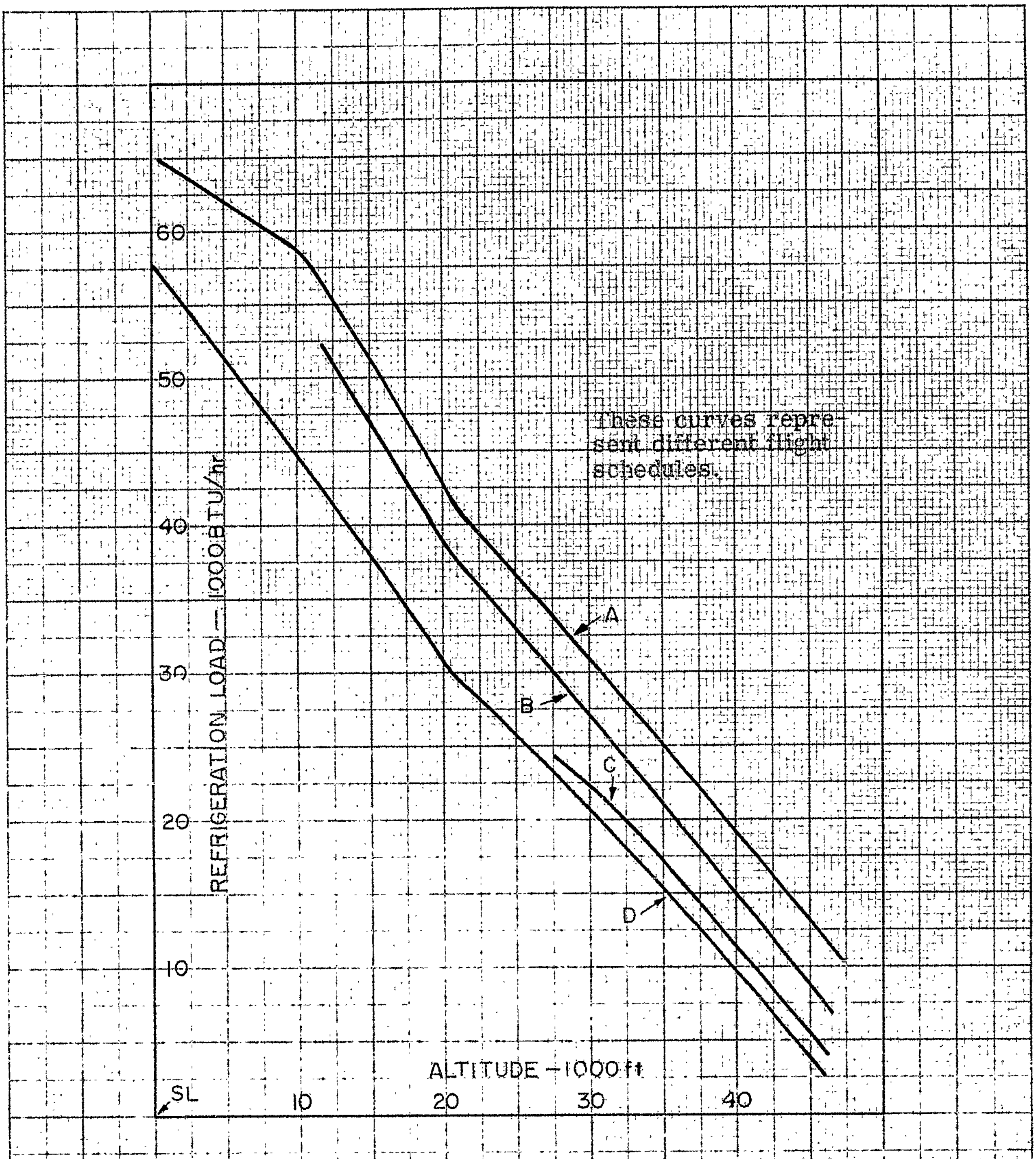


Figure 1. Cabin Refrigeration Loads (Maximum Values)

If, however, the condensed vapor is allowed to re-evaporate in the cabin, a second enthalpy change equal to the first occurs, thus reducing the "corrected" air temperature to its original value, called the "dry air" rated temperature.

For the case of zero re-evaporation, the "corrected" temperature, as a function of "dry air" temperature and ambient humidity, is given in Figure 2.

It can be seen that high ambient humidity operations, when large amounts of moisture are condensed and separated from refrigerated air, impose severe penalties on air conditioning systems, since the cooling capacity is reduced by the elimination of the cooling effect of the separated moisture. Also, the temperature at the turbine discharge must be maintained above freezing to prevent ice from clogging the system. This is usually accomplished by diverting a portion of the high temperature bleed air directly to the turbine discharge. Limiting turbine discharge temperatures in this way obviates the use of more efficient cooling systems which are capable of discharging air at extremely low temperatures, since no use can be made of the low temperatures if freezing is to be avoided.

There are additional features of high humidity operation that are considered objectionable above and beyond the penalty of reduced cooling capacity. Entrained moisture enters the cabin as a fine mist or fog, disappearing as it re-evaporates. In instances of high cabin humidities, the fog may spread far into the cabin before disappearing. Also, when a plane that is thoroughly chilled from high altitude flight suddenly makes a rapid descent, the relatively high moisture content present at the low altitudes condenses and fogs up the cool cockpit surfaces, sometimes completely obscuring the canopy and instruments. Danger of such conditions is great, being especially critical at the low altitudes when the plane is preparing to land.

A moisture separator, inserted at the discharge from the expansion turbine can remove enough moisture to preclude excessive fogging. However, the maximum moisture that can be removed is limited by the dew point (corresponding to the discharge temperature) and limitations of moisture separation. For cases of high ambient humidities, the cabin inlet humidity may still be high enough to cause some fogging but to a lesser extent than previously noted. The means of limiting turbine discharge temperatures to temperatures greater than freezing usually required the use of an anti-icing by-pass valve and controls. The by-pass valve controls the amount of high temperature bleed air that is throttled directly to the turbine discharge.

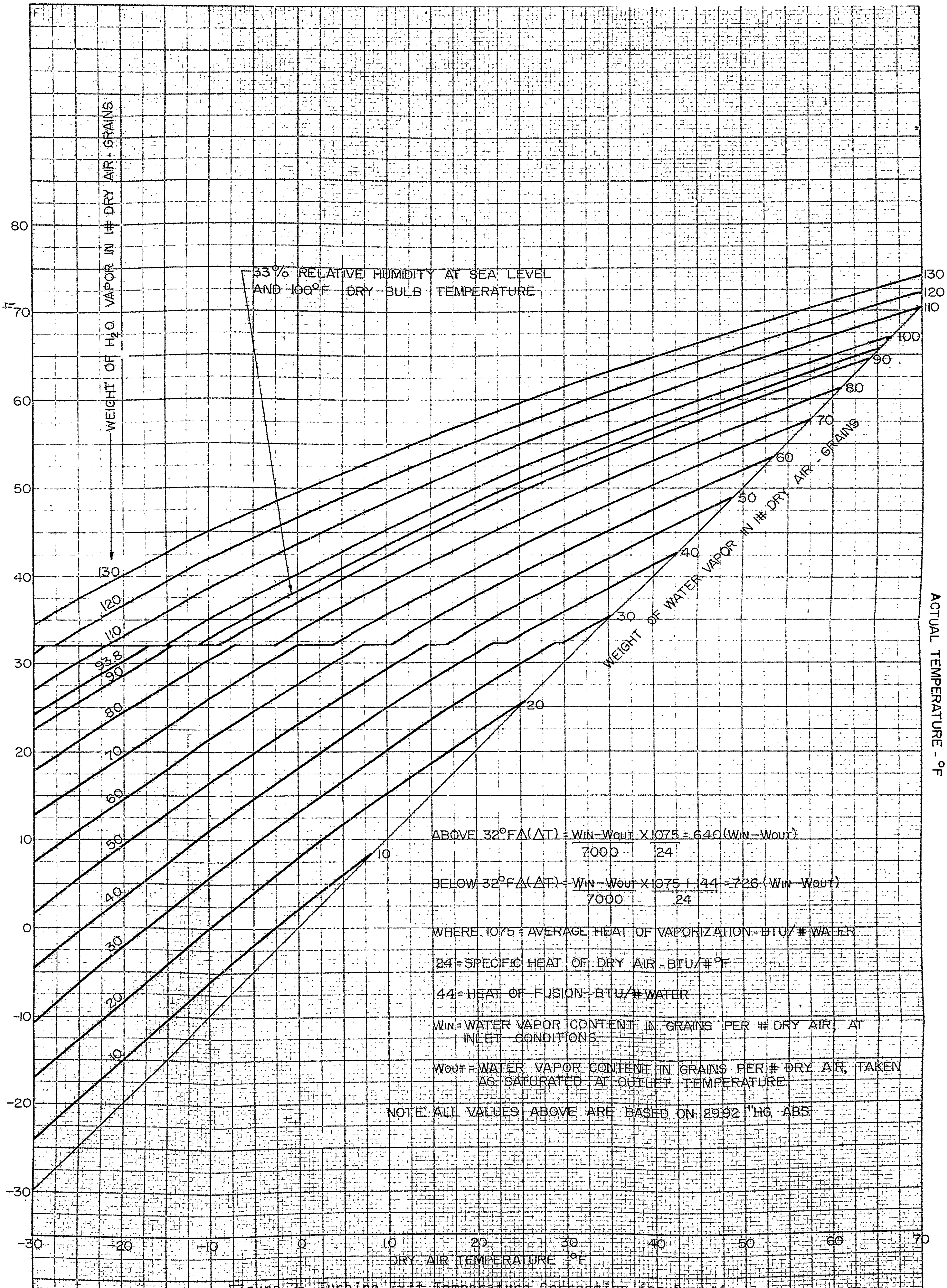


Figure 2. Turbine Exit Temperature Correction for Dry Air

MOISTURE REMOVAL METHODS

The known methods of moisture separation have been reviewed thoroughly in the literature. A list of the important types that might be applicable are given below.

(1) Adsorption drying

(a) liquid
(b) solid

(2) Compression drying
(3) Refrigeration drying

(1) Adsorption drying.

There are a considerable number of liquids and solids known to adsorb water effectively. The metal hydrides and acetylides 1/, such as calcium hydride and calcium carbide react very well with moisture. However, poor regeneration eliminates this group from consideration. The metal salts and oxides, such as calcium chloride and calcium oxide are likewise discarded for poor regeneration qualities.

The use of sulfuric acid and the polyhydric alcohols is questionable since gravitational flow of these fluids cannot be relied upon. Therefore, the pumping problems for these fluids become too complex to warrant consideration for their use. The use of liquid adsorbents also introduce the problem of carryover with the air stream.

The remaining solid adsorbents 1/, such as silica-gel, activated aluminum oxide and anhydrous calcium sulfate, possess fair adsorbing qualities. Silica-gel is examined further since it offers the best adsorbing qualities of the foregoing materials. There are, however, several problems involved by the use of silica-gel which are described as follows:

(a) During the adsorbing action, heat is liberated in an amount equal to the latent heat of condensation plus the heat of wetting. The generated heat increases the temperature of the bed and the dehumidified air considerably. Provision must be made for an additional heat exchanger to remove this heat for satisfactory refrigeration.

(b) The flow resistance across the silica-gel bed is considerable when considering reasonable bed dimensions. The additional pressure drop, resulting from this, reduces the overall cooling capacity of the air cycle unit.

(c) Silica-gel cannot adsorb moisture indefinitely but must be regenerated frequently to reactivate the material. This operation may be performed by exposing it to hot air, during which process the "bed" increases in temperature and dries out. The power required to provide the necessary hot air is a function of dimensions of the bed and face velocity of the hot air. For

1/ Reference 9

reasonable bed sizes, the power expended may be considerable.

(d) Provision must be made for at least two dehumidifiers, associated ducting, suitable inlet and exit controls for both regeneration and dehumidifying air, to permit continuous operation. The mechanical and control problems involved must also be considered.

(e) The regenerative hot air may be obtained from the air discharging from the blower heat exchanger of an air cycle cooling unit. The amount of air available for this purpose is a function of the turbine and blower characteristics, and the pressure drops of the heat exchanger and the dehumidifier. Any limitation of weight flow due to the increased flow resistance of the dehumidifier subtracts from the performance of the cooling unit.

(f) Immediately following regeneration, the bed temperature may be considerably higher than the temperature of the moist air at the dehumidifier inlet. To prevent any reheating of the moist air, an auxiliary heat exchanger is necessary to cool the bed.

(g) There are limitations of the linear air velocities across the bed that may be tolerated before abrasion problems occur. For the maximum humidity encountered, face velocities across the bed increase, introducing possible mechanical and abrasion problems.

(h) The presence of oil vapors in the moist air offer the distinct possibility of carbonizing the silica-gel during regeneration. This is especially critical at high temperatures of regeneration.

(i) The total weight of the unit including all beds, ducts, controls, possible mechanical drives, and auxiliary heat exchangers, is considerable.

(j) While silica-gel may be used for long intervals of time, replacement of the gel is required at indefinite periods. Inclusion of entrained moisture at the inlet to the bed promotes breaking up of the silica-gel particles and hastens eventual replacement.

Based on the foregoing discussions, the use of silica-gel has been rejected from any further consideration.

(2) Compression Drying

For a given moisture content, the saturation temperature increases as pressure is increased. Specifically, a greater amount of moisture can be removed from air at a given temperature by increasing its pressure. For example consider the following hypothetical problem:

Air at 29.92"Hga abs and a specific humidity of 110 gr/lb is at a 90°F dry bulb temperature. The relative humidity is 52%. Air at 60°F dry bulb temperature and the same pressure is saturated at a specific humidity of 77 gr/lb. Therefore, cooling the air containing 110 gr/lb from 90°F to 60°F at 29.92"Hg abs will condense 33 gr/lb (110-77). Air compressed to 150"Hg abs and then cooled to 60°F dry bulb temperature is saturated at 15 gr/lb condensing 95 gr/lb (110-15). Thus it can be seen that nearly a threefold advantage in condensing potential can be obtained for a fivefold increase in pressure.

However the relative advantage of compression drying decreases as the final temperature decreases. Under the conditions cited, cooling to 33°F would show an advantage of only 27% for compression drying.

The above example illustrates the principle behind compression drying. Practically speaking, when air is compressed it is usually done under conditions where the temperature of the air also increases. This temperature rise is so large as to reduce or even prevent condensation despite the increase of pressure. Obviously isothermal compression is desired. Cooling the high pressure air with ambient cooling air offers one means of approaching this. This type of cooling is limited in aircraft application. For, with high aircraft speeds, the cooling air temperatures are much greater than ambient temperatures.

The increase in cooling air temperature over ambient temperature due to airplane speed, known as ram temperature rise, is given in the following equation:

$$\Delta t \text{ ram} = \frac{V^2}{2(g)(J)(c_p)} \qquad (3)$$

where V = true air speed - fps

Despite the above limitation, the possibilities of compression drying cannot be ignored when selecting a moisture removal system. Air cycle cooling systems lend themselves to this operation, since high pressure air is available in the cycle.

(3) Refrigeration Drying

It is well known that cooling humid air condenses its moisture content, once the saturation temperature is passed.

The specific humidity of air is given by the Carrier equation:

$$W_s = \frac{(4354) \quad P_s}{P - P_s}$$

where W_s = specific humidity - gr/lb dry air
P_s = partial pressure of vapor in air - "Hg abs
P = total pressure of air vapor mixture - "Hg abs

The specific humidity corresponding to saturated conditions can be found by using the above equation. For any temperature, the equilibrium vapor pressures P_s can be found 1/. When the air temperature falls, this vapor pressure reduces rapidly and moisture in excess of saturation condenses. For example, consider the following problem:

> Air at 29.92"Hg abs and a specific humidity of 110 gr/lb is at a 90°F dry bulb temperature. The relative humidity is 52%. If the air is cooled to 40°F at constant pressure, the air becomes saturated with 37 gr/lb. Therefore 110-37 = 73 gr/lb condenses from the air vapor mixture. Obviously a limitation to this method of moisture removal occurs when there is freezing of the condensed moisture and standard moisture separator equipment fails.

Figure 3 shows saturation lines as a function of dry bulb temperature for various pressures. Inspection of Figure 3 indicates that air at a given dry bulb temperature is capable of containing a decreased amount of water as the pressure is increased.

It is apparent that by combining the desirable attributes of the moisture removal principles of compression and refrigeration drying, an excellent moisture removal system is possible.

HEAT EXCHANGER PROBLEMS

In order to estimate the cooling system performance, it is necessary that the performance of the individual components be known. The regenerative heat exchanger, used in a high performance cooling system, is operating under conditions that are unlike the conditions encountered in the more conventional cooling systems. The major difference is that moisture condenses from the hot air side while the cooling air side may be either nearly dry or have moisture evaporating in the air stream.

1./ Reference 8

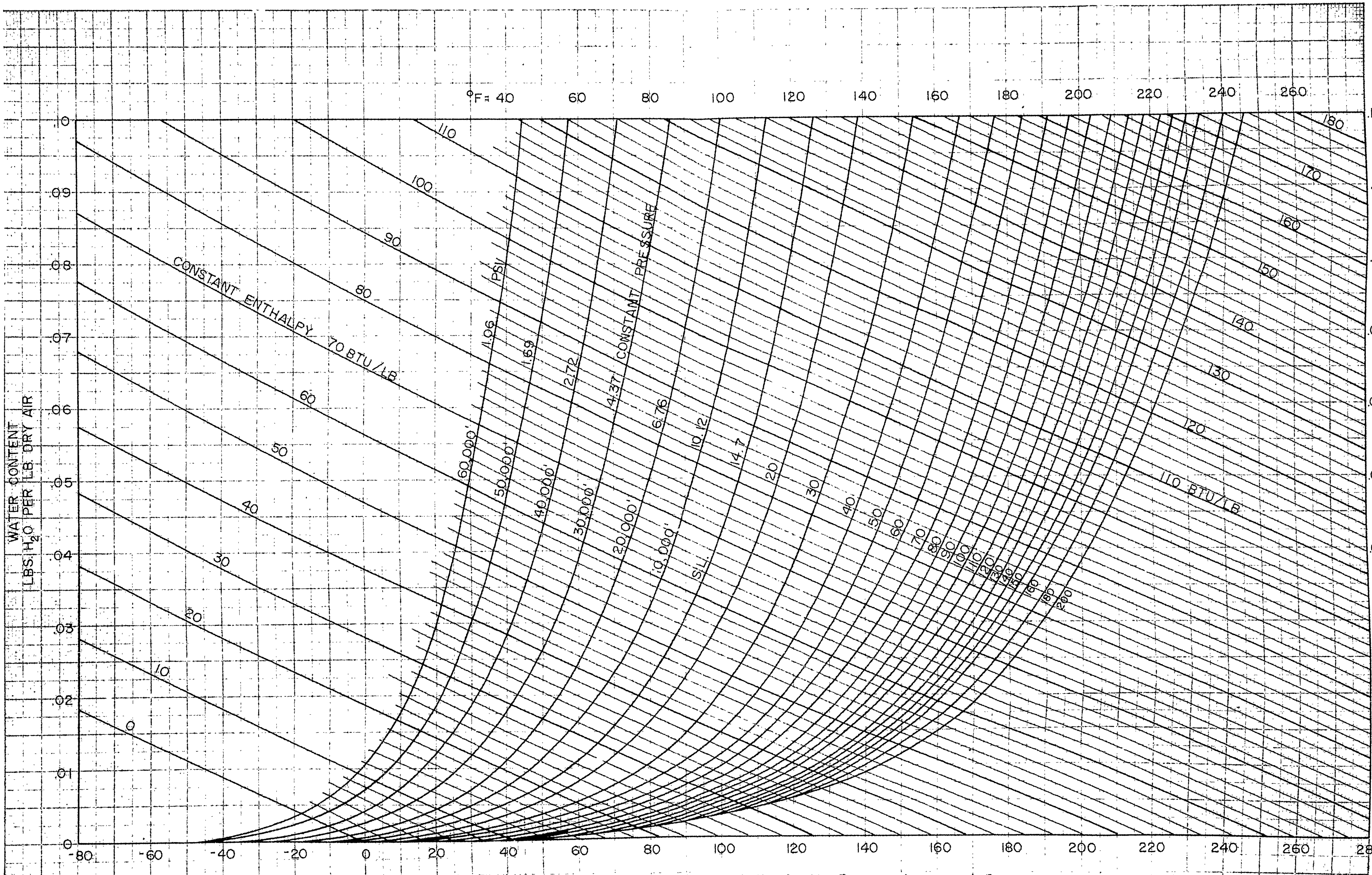


Figure 3. Enthalpy and Specific Humidity of Saturated Air Vs Temperature and Pressure

For "dry" air to air systems, heat exchanger performance may be conveniently expressed in terms of an effectiveness, 1/ 2/ 3/ defined here as:

$$\eta_x = \frac{t_{h\ in} - t_{h\ out}}{t_{h\,in} - t_{c\,in}}$$

where the subscripts h, c, in, out, refer to the hot fluid and cooling fluid into and leaving the heat exchanger, respectively. The effectiveness of a dry heat exchanger is a function of the size of the exchanger, the type of heat transfer surface, and the flow rates of the hot and cooling air streams.

When a wet heat exchanger (one in which condensation occurs) is considered, the problem of predicting performance is complicated because performance is also dependent upon the condensation rate. The condensation rate in turn, is a function of inlet humidity, temperature, pressure, and flow rate. Data available in the literature are not readily applicable to the solution of this problem, since much of the work done previously is concerned with operation at pressures and temperatures well below those encountered in jet airplane air conditioning systems.

Analytical methods are developed to calculate "wet" heat exchanger performance conveniently expressed as a temperature difference parameter defined as:

$$\eta_{wet} = \frac{t_{h\ in} - t_{h\ out}}{t_{h\ in} - t_{c\ in}}$$

where the subscripts are the same as above.

The analytical methods used to calculate the temperature difference parameter of the "wet" heat exchanger, as well as the tests that were required to substantiate such methods, are described in detail in the appendices of this report.

1/ Another method in general use for expressing effectiveness involves multiplying the above equation for effectiveness by the ratio of the hot fluid capacity rate [(Flow) x (Specific Heat)] and the minimum capacity rate. Both effectiveness terms are identical only for capacity ratios $\geq$ 1. It is the practice of the writer to employ the simpler of the two methods, illustrated in the text above.

2/ Reference 6, page 10.

3/ For the range of temperatures and pressures involved, effectiveness is independent of the other properties of air.

METHODS FOR INCREASING COOLING SYSTEM CAPACITY

In the desire to develop high performance cooling systems, some of the thermodynamic aspects of current air cycle systems must be considered. For a given air cycle system:

(1) The cooling capacity for unit air flow is a direct function of the overall pressure ratio of the cycle. The greater the pressure ratio, the greater the cooling performance.

(2) The cooling capacity is little affected by an increase in the number of air to air heat exchangers above some maximum figure since a value of overall heat exchanger effectiveness is reached which cannot be exceeded.

(3) No matter where (in the cycle) the condensation of the ambient humidity occurs, the heat liberated by such condensation is practically constant.

(4) Higher component efficiencies could improve package cooling capacity under dry conditions. However component improvement would NOT result in a corresponding increase in cooling capacity where an appreciable amount of humidity exists. For present systems under humid conditions it is necessary to hold turbine discharge temperatures above the freezing point in order to remove moisture.

Use of more efficient components would require a larger proportion of high temperature air to maintain above freezing discharge temperature. Hence, component improvements would not result in lower package outlet temperatures. An exception to this would be a system where moisture removal is independent of turbine discharge temperature.

Investigation of operation of air cycle systems reveals that the performance of a cooling package reduces when encountering high ambient humidity. One reason for this is illustrated in item (3). above. All of the ambient vapor that is condensed after expansion in the air turbine cannot re-evaporate, since a portion of the condensate is removed by a moisture separator. If, however, the condensed moisture is allowed to cool other portions of the cycle, some regain of the lost cooling capacity is possible. The evaporation of condensed water is particularly effective when applied as a water spray into the cooling air stream of a heat exchanger. Forthis case re-evaporation, both at the inlet to the heat exchanger and along the fluid path in the heat exchanger, serves to reduce the cooling air temperature, increasing the rate of heat transfer.

Another reason for reduced cooling capacity is the operation of the anti-icing by-pass, which prevents freezing at the turbine discharge and consequently reduces cooling capacity. One means of eliminating this is to design for maximum condensation upstream from the turbine. Turbine discharge temperatures can now become as low as possible with the overall pressure ratio, making unnecessary the use of the anti-icing by-pass. One method for accomplishing this is as follows:

A supplementary heat exchanger, known as a split regenerator, can be incorporated in the cooling package located immediately upstream of the cooling turbine inlet. This split regenerator, utilizing a portion of the cold air from the cooling turbine discharge as its coolant flow, will effectively cool the cabin flow to insure that sufficient condensation and separation of moisture occurs upstream from the turbine. Relatively dry air expands through the turbine and leaves at very low temperatures, limited only by the turbine pressure ratio and efficiency. However, only a portion of this very cold air is now available for cooling the cabin, since some of the flow has been diverted back through the split regenerator. The cabin inlet temperature must be low enough to maintain at least the same cooling capacity as a unit in which all of the turbine discharge goes to the cabin, and with a lower total bleed air flow, if any advantage is to be realized from the system.

The advantage of the split regenerator over a simple regenerator is that the use of a split regenerator permits the spraying of condensed moisture back into the cooling air stream. For an ordinary regenerator such spraying is impossible since the cooling air enters the regenerator at temperatures below freezing. By dividing the regenerator into twin series exchangers, whereby the cooling air essentially goes through two passes, the operation of a water injection system is possible. The condensed moisture can be sprayed into the regenerator cooling air before the second pass after the cooling air has been sufficiently heated in the first pass.

The use of a split regenerator is undesirable from the viewpoint of weight and volume considerations, since the incorporation to an existing cooling package represents an increase in weight and size. However, credit should be taken when using a regenerative heat exchanger by an amount equal to the reduction in turbine exit duct size.

Another configuration, employing but a single pass regenerative heat exchanger, would represent some weight saving over split regenerator operation. Air leaving the cooled compartment at compartment equilibrium temperatures is used as regenerative cooling air for a single regenerator. A water spray system would be particularly effective since compartment temperatures are of the order of 70-80^{o}F, and there is no danger of freezing. This configuration is attractive for a number of reasons:

(1) Compartment air, containing a considerable amount of cooling capacity that is normally dumped overboard, would be utilized.

(2) A single regenerator rather than a heavier split type can be used in conjunction with a moisture spray system.

(3) The regenerative heat exchanger, using cabin exhaust air for cooling air, operates at a flow ratio (cooling air flow: cabin air flow) of approximately 1:1. The split regenerator operates at a flow ratio less than 1, probably on the order of 0.5 - 0.6:1.

There are disadvantages in cabin exhaust cooling:

(1) At sea level flight the compartment must be maintained at pressure greater than atmospheric, so that flow can be directed through the regenerator. This positive pressure differential is estimated to be 2 in. Hg for design flow rates.

(2) Ground cooling would be questionable if accessibility to the compartment is desired.

(3) At some conditions (high altitude, low speed) the cabin temperature could be higher than the ram temperature. Unless the cabin exhaust air by-passes the regenerator under these conditions, the temperature of the turbine inlet air would be increased. Providing a by-pass control is, in itself, a disadvantage because of the increased weight and complexity.

(4) There is a considerable delay in attaining design cabin temperatures since there is a thermal lag due to the use of cabin air in the cooling cycle.

The use of single or split regenerators and moisture spray systems reduces the required bleed flow for a given cooling capacity under conditions of high ambient humidity. This economy of operation is maintained throughout a complete range of ambient humidities at the expense of the extra weight of the regenerators and spray systems. When the ambient humidity is low, the economy of operation still exists but in a limited system operating range. This is because any decrease of turbine inlet temperatures resulting from precooling at the regenerator is partly balanced by the reduction of available flow for cabin cooling and the reduced temperature drop across the turbine due to pressure losses in the regenerator.

SECTION II

COOLING SYSTEM CONTROLS

The problem of controlling cabin air conditioning and pressurization systems consists of two distinct parts: cabin pressure controls, and air conditioning package controls.

The first part, cabin pressure controls, is independent of the cooling system. It consists of the cabin pressure regulating valve and safety valves.

The second part, to be discussed here, is the air conditioning system controls. These controls may be performed wholly by conventional separate control devices, or in part by internal control methods. Internal control methods are defined as the suitable variation of the flow areas of the stationary parts of the turbo machinery components.

The controls for a high performance cooling system, whether "internal and/or separate", should perform the following functions:

(1) Control cabin temperature.

(2) Control cabin humidity.

(3) Control cabin air flow.

(4) Protect system from danger of improper operation or damage due to icing.

(5) Protect unit from excessive speed.

(6) Protect jet engines from excessive bleed and the cabin from excessive air flow.

A brief discussion of the control of regenerative cooling systems by means of separate control devices follows in the order listed above. Following this, the various types of internal control methods and the specific application of such methods to typical systems and their related problems will be examined.

SEPARATE CONTROLS

(1) Cabin Temperature Control.

The primary purpose of the air conditioning unit is to maintain cabin pressure and temperature. The control of temperature by separate control components is usually made by modulating the unit discharge temperatures to satisfy cabin requirements.

The by-passing of bleed air around the turbine, or the entire cooling package, and mixing it with turbine exhaust is one method in common use. In a regenerative system, the by-passing of air from points upstream of the unit will supply to the cabin humid air 1/. To minimize the amount of water introduced with by-pass air, the use of a small amount of the hottest available air is advantageous, which is in line with the method used with some conventional units.

(2) Cabin Humidity Control.

Conventional cooling systems do not provide for humidity control but supply the cabin with air of the same specific humidity as in the surrounding atmosphere. In the event a water separator is used downstream of the package the moisture supplied to the cabin is decreased. However, control to a specific value of cabin humidity is not attempted.

Similarly, with a high capacity cooling system, the only humidity control device required is one limiting the maximum humidity entering the cabin to a value lower than that obtained by conventional cooling systems.

(3) Cabin Flow Control.

Normally, pressure in the cabin is maintained by a cabin pressure regulator. Operation is based on the assumption that more air than necessary to cover leakage is supplied to the cabin and the excess is dumped out by the cabin pressure regulator.

However, if the amount of air supplied to the cabin is less than the cabin leakage at required cabin pressure, the cabin pressure regulating valve will be closed and cabin pressure will balance at some equilibrium pressure, which will be between supply and ambient pressure, resulting in lower than cabin pressure schedule requirements.

The air conditioning unit, aside from cooling requirements, must supply at least enough air to cover leakage. In the cases where leaks are smaller than the required ventilation flow, the minimum required flow is dictated by ventilation requirements.

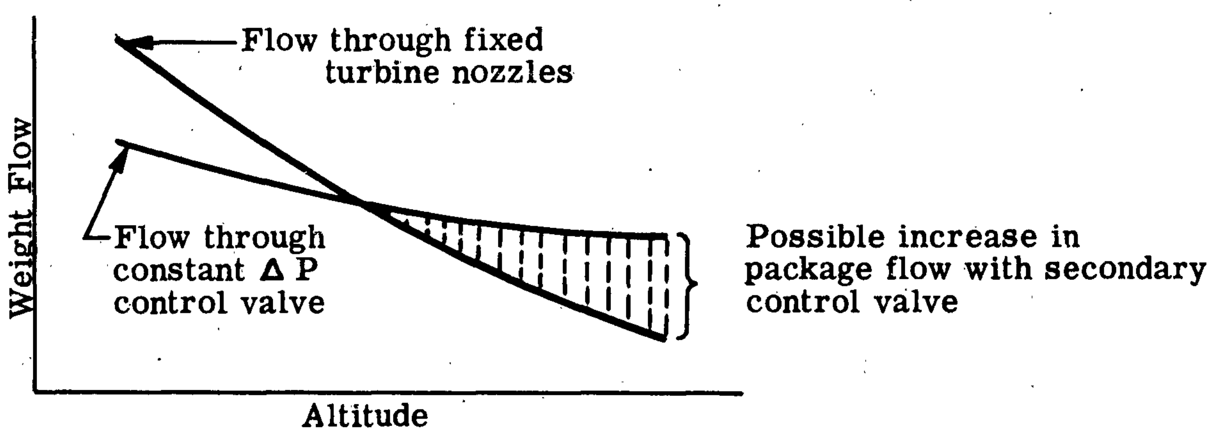
1/ This will depend upon ambient humidity.

Ventilation is defined here as the minimum amount of fresh air required by cabin occupants. Usually no special controls are required to assure sufficient air flow.

Cooling packages in use at the present time incorporate certain types of separate flow control devices, either individually or in combination.

In a fixed turbine nozzle area cooling unit, control of cabin flow is possible by the use of a throttle valve. Throttling to conserve bleed air flow at low altitudes reduces cooling capacity of the unit. The limitations of such throttling control occur when cooling requirements cannot be met.

Another method of separate flow control in use consists of the combination of a primary and a secondary flow control valve. The primary flow control valve incorporates a venturi and a throttle valve. The throttle valve actuation comes from the sensing of the pressure difference between the inlet and the throat of the venturi. When a constant pressure difference is maintained, the throttle valve position will be such that the maximum package weight flow will be proportional to the square root of the inlet density. Inspection of the curve below indicates that there is some altitude where the nozzles become the controlling factor, since the flow capability of the nozzles is less than that of the flow control valve.



If the flow required by the cabin exceeds the flow that can be maintained by the nozzles (above the altitude where the nozzles control the flow), a secondary flow valve can be used which by-passes the turbine and permits a flow as dictated by the primary flow control.

(4) Anti-icing Control.

In conventional systems which use no water separator, protection against icing is not required. During varied conditions of operation the ice which is deposited in the turbine exit duct melts later and does not impair safety of the unit. Icing, however, reduces capacity by building up turbine back pressure. As far as the cabin is concerned, particles of ice are blown into it and melt. Also, with increased temperature, slugs of water and wet ice are blown into the cabin from the ducts. In the case of a water separator which may be damaged when icing occurs (for example, collapse of the filtering section due to excess pressure drop), anti-icing controls are used to maintain the temperature upstream of the separator above freezing. The anti-icing valve usually by-passes warm air from the turbine or package inlet to the turbine exhaust, providing a mixed air temperature of about 35°F.

Instead of an anti-icing valve, a screen at the turbine exit can be used which will maintain a temperature of 32°F or the dew point temperature, whichever is lower. The self-controlling action of the screen is a result of the plugging of air passages with ice and increasing turbine back pressure until equilibrium is reached. For example, if the dew point of the air is higher than 32°F, ice forms on the screen and increases the turbine back pressure until the turbine exit temperature is greater than 32°F, at which time no more ice is formed. For dew points lower than 32°F, the ice builds up turbine back pressure, thus increasing the turbine exit temperature to the dew point temperature where no more ice formation takes place.

In a high performance cooling system, as considered in this report, icing downstream of the turbine is negligible due to the low humidity of the turbine inlet air. However, upstream from the turbine freezing may occur in the regenerator.

Although complete icing of the regenerator may similarly be prevented by the self controlling action, as discussed above, additional protection against icing in the regenerator is provided. It is accomplished by modulating the regenerative air flow to maintain the temperature of regenerator walls in the coldest place at above 32°F.

(5) Speed Control.

Air cycle machines operate at a speed resulting from a balance between turbine and compressor power. Experience has shown that for satisfactory design of fixed nozzle area air cycle machines, the speed variation with altitude and engine setting is generally confined to a limited area, and no provision for speed control is required.

(6) Flow Limiting.

A sonic venturi is commonly used to protect the jet engine compressor from bleeding excessive amounts of air and also to prevent the supplying of excessive amount of air to the cabin when, for example, the temperature by-pass valve is wide open.

INTERNAL CONTROLS

Before proceeding with the discussion of internal controls, a brief description of internal controls is desirable. Described hereafter are:

(1) Variable diffuser vanes

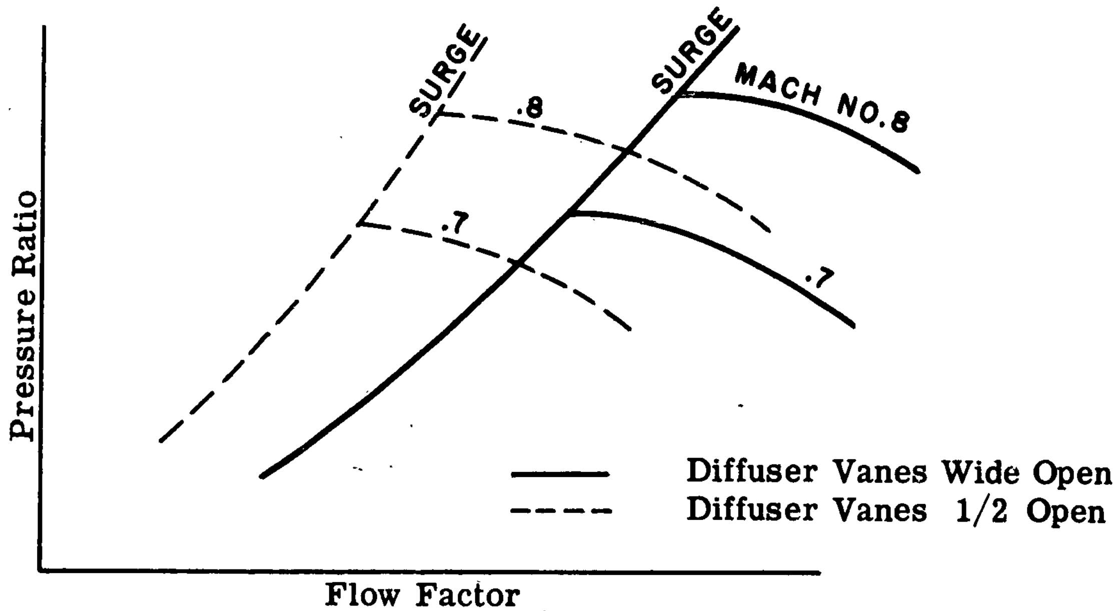
(2) Prewhirl guide vanes

(3) Variable nozzle area

(1) Variable Diffuser Vanes.

There exists a means of governing or controlling the operation of a blower by employing movable diffuser vanes. The diffuser vanes are moved to vary the inlet angle and the area. Generally all vanes are pivoted and move together.

The effect of variable diffuser control on the performance of a typical blower can be illustrated by the figure below.

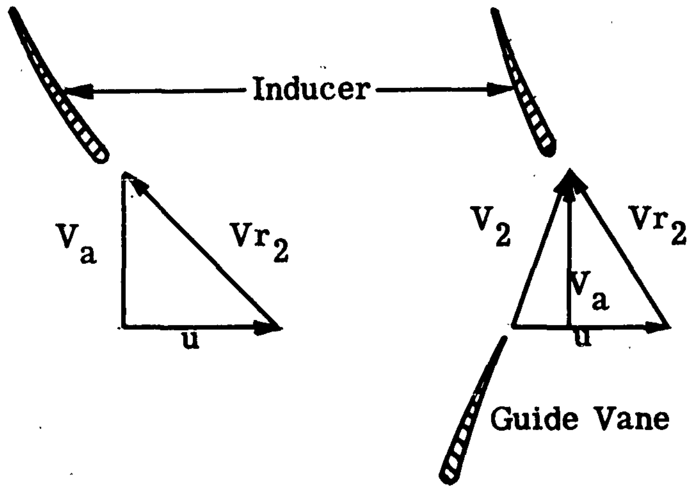


Operating Characteristics of Blower with Variable Diffuser Vanes

From the figure it can be observed that the surge point is moved toward the left (region of lower flow) as the diffuser vanes are closed; hence a particular application for this type of control is to extend the satisfactory performance of a blower into regions of low flow.

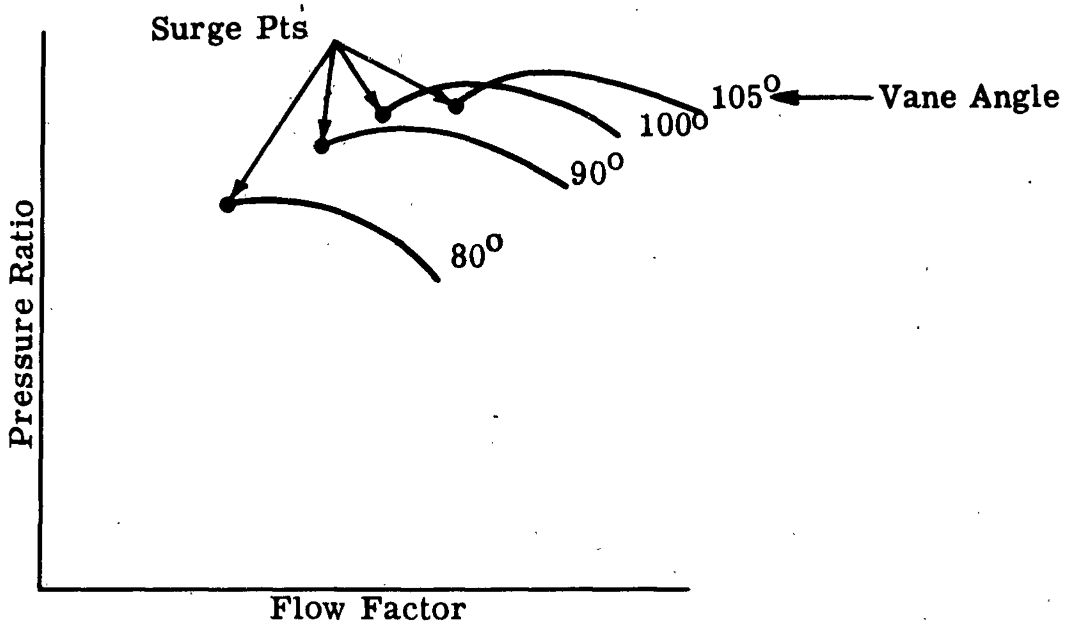
(2) Prewhirl Guide Vanes.

Another means of controlling the operation of a blower involves the use of prewhirl guide vanes. For this case, movable impeller inlet guide vanes are employed to prerotate the air as it enters the impeller. Such prewhirl may be in the direction of or opposite to the rotation of the impeller. For this particular application prewhirl in the direction of rotation is desired. It is depicted in the figure below.



No Prewhirl With Prewhirl

As can be seen from the above figure, a reduced relative velocity, Vr_2, is obtained without reducing the axial velocity, V_a, by prerotation at the impeller inlet. The air is given a whirl component of velocity in the direction of rotation, and a new velocity triangle is shown above. Since prewhirl in the direction of rotation reduces the change of angular momentum through the impeller, work capacity reduces. The reduction in the relative air velocity corresponds to a comparable decrease in impeller blade inlet Mach No., which is generally a desirable feature toward the improvement of blower operation. Compressor delivery pressure reduces slightly, due to the increased pressure drop occuring at the inlet guide vanes. The effect of vane position on a typical blower operating at constant mean impeller Mach No. and inlet conditions is illustrated in the following figure.



Operating Characteristics of Blower with Prewhirl Guide Vanes at Constant Inlet Mach Number.

Notice that satisfactory compressor performance is extended into the region of lower flow by a reduction of the vane angle.

(3) Variable Nozzle Area.

The flow through the turbine is directly proportional to the available turbine nozzle area. In many instances the effect of bleed air, ram air, and cooling load schedules result in greater bleed turbine flow than is necessary. A method to conserve bleed flow has been devised whereby the turbine nozzles are moved to vary the area and inlet angle. In this way economy of operation is assured since only the flow that is required is used. It should be remembered that there are specific limitations as to the maximum nozzle area range permitted. In some cases, if nozzles have to be almost closed, efficiency of the turbine is reduced considerably. This fact is illustrated in the figure on page 22, where the efficiency of a typical variable area turbine is shown for a constant pressure ratio.

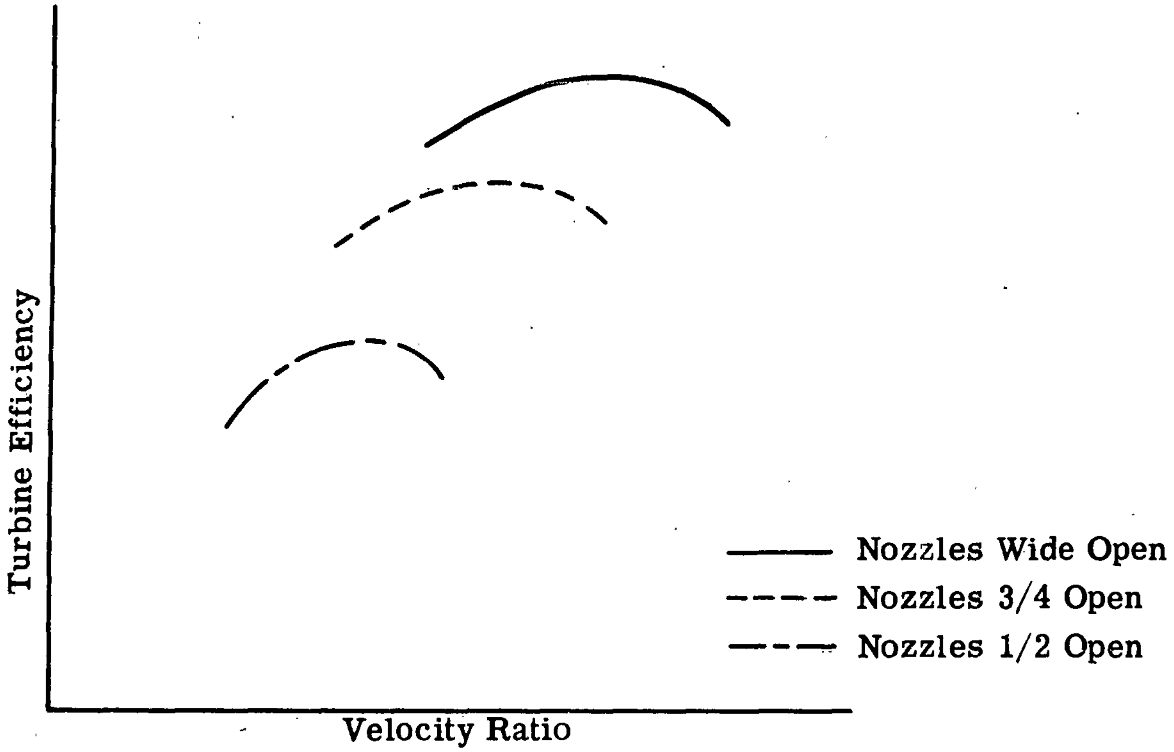
In general, the use of variable area turbine nozzles has proven attractive when a cooling package must operate under conditions when the airplane refrigeration requirements are relatively constant over a wide altitude range. The air flow required at the high altitude (where the bleed pressure is low) generally dictates the selection of a nozzle area, which, when fixed, permits excess flow at low altitudes (where bleed pressures are usually high). To conserve bleed flow at the low altitudes, the fixed nozzle unit could include a throttling valve. The limitation of throttling control occurs when a large degree of throttling reduces the unit cooling capacity to such an extent as to make it impossible to satisfy the cooling requirement with an economical bleed air flow.

In contrast the use of a variable area nozzle would reduce the flow at low altitudes without reducing the available turbine inlet pressure. The economy gained is large when the range of nozzle area to be covered is moderate (up to 7:1 approximately). The use of variable nozzle area control is limited by the reduction of turbine efficiency resulting from large area variations. When the reduced performance approaches that of a separate throttle valve control, the added complexity of variable nozzle area control is not justified.

There is a further problem imposed by a variable area turbine on a simple cooling system. At low altitudes when the bleed pressure increases, it is usual for the ram pressure to increase also, increasing the blower inlet density. Turbine power has increased because of increased pressure ratio, but the turbine air flow has remained at a relatively constant value because of the variable area nozzles. Since there is no such flow restriction on the blower, the usual effect on a centrifugal blower is to overload the turbine to a reduced speed.

Slowing down of the unit has a marked effect on the turbine and system performance, since the decreased value of velocity ratio (turbine tip speed divided by theoretical spouting velocity) results in a decreased turbine efficiency. The effect of reducing both nozzle area and speed on the turbine efficiency is such as to make for poor unit performance outside of a rather limited range of operating conditions.

Variable nozzle area turbines employed in bootstrap units may introduce other control problems. There has been experienced great difficulty in keeping even a fixed turbine nozzle area bootstrap unit from having the blower operate at very low flow coefficients. When variable turbine nozzle area is introduced, this difficulty is increased, and blower surge characteristics must be considered. At low altitudes the bleed pressure increases. For relatively constant cooling load requirements, the weight flow increases but at a much slower rate. This results in the blower operating at low flow factors and high pressure ratios. Inspection of blower performance curves reveals the blower may surge 1/. To correct this problem of surging, internal control of the blower by any of the methods previously discussed is indicated. This naturally increases the complexity of the variable nozzle area bootstrap system. Before specifying this control, the economy of bleed flow gained should be weighed against the increased complication of the cooling unit.



Variable Area Turbine Performance

1/ This will be a function of individual blower characteristics.

SECTION III

DISCUSSION OF HIGH PERFORMANCE COOLING SYSTEMS

A study of several air cycle air conditioning systems was undertaken to determine the possibility of satisfying the following conditions:

(1) Air entering the cabin should contain as little moisture as possible and for the conditions of this problem a maximum amount of 50 gr/lb dry air has been specified. Even under these conditions of reduced inlet humidity there will probably be some mist introduced into the cabin and on occasion fogging of the cockpit surfaces but to a lesser extent than previously noted (Section I, Humidity Operation). There are several reasons for specifying the maximum moisture content of the air entering the cabin as 50 gr/lb dry air. A maximum figure less than 50 gr/lb would require lower temperatures at the water separator. If a maximum figure is selected so that the air would have to be cooled below 32° F at the water separator inlet, the danger of freezing and subsequent blocking of the water separator would exist. Tests conducted as a part of this study indicate that air containing only 50 grains of water per pound of dry air is not likely to cause the turbine discharge duct to become blocked with ice. It is recognized, however, that complete tests on duct icing, as well as on wet heat exchanger performance, constitute, in themselves, adequate material for a very exhaustive study. The tests conducted under this study were intended to permit the selection of a high performance cooling system, and the results are, to this extent, adequate to permit tolerating the 50 gr/lb dry air, without fear of performance penalty, at very low turbine exit temperatures.

(2) The total bleed air flow must not exceed the bleed flow required by current typical cooling systems of the same cooling capacity. This condition is imposed because the disadvantages inherent in bleeding larger amounts of air contradict the definition of a high performance system.

(3) The unit selected shall not have excessive bulk, weight and cooling air drag. No specific limitations can be arbitrarily set for any of these characteristics since such limitations are by far dependent upon the type of aircraft employed.

The following systems have been considered as having the best chance of meeting the conditions described on the previous page. In all cases, provision has been made for spraying the condensed moisture into the cooling air stream of the appropriate heat exchanger:

(1) Simple regenerative with split regenerator.

(2) Simple regenerative with cabin exhaust cooling.

(3) Bootstrap regenerative with split regenerator.

(4) Bootstrap regenerative with cabin exhaust cooling.

(5) Dual cycle regenerative.

The first system considered is a simple regenerative system shown in a schematic form in Figure 4. The cycle is as follows: engine compressor bleed air enters the first heat exchanger (ram operated). The bleed air then passes through the second heat exchanger (blower operated). The bleed air then passes through the split regenerative heat exchanger. Cooling air for the regenerative heat exchanger consists of a portion of the turbine discharge air. The bleed air at the regenerative heat exchanger exit should be cooled to a temperature low enough to condense most of the ambient humidity so that all but 50 gr/lb dry air maximum remains in the air stream. The water removed is sprayed into the cooling air of the second pass of the split regenerative heat exchanger where, in evaporating, it reduces the effective temperature of the cooling air. The bleed air then expands through the turbine and enters the cabin, with a part of the turbine discharge air going to the regenerative heat exchanger as cooling air. The moisture content of the turbine discharge air must be low enough (50 gr/lb maximum) so that very little ice can be expected. Control to insure adequate moisture removal can be accomplished by placing a valve in the regenerative heat exchanger cooling air line, to maintain the turbine inlet temperature at some prescribed value by regulating the amount of cooling air going through the regenerative heat exchanger.

The second system considered, a modification of the first, provides for using cabin exhaust air as cooling air in the regenerative heat exchanger. A schematic of this system is shown in Figure 5. The split regenerator is not required since the compartment exit air is above freezing temperatures, and moisture may be sprayed directly into the cooling air. In this system, compartment air is bled through the regenerator and then ducted overboard.

The third system to be analyzed is the bootstrap regenerative system with split regenerator. A schematic of this system is given in Figure 6. Engine compressor bleed air enters the first heat exchanger (ram operated). The cooled bleed air is compressed by the air turbine driven compressor and passes through a second ram heat exchanger. After most of the heat of compression is removed, the bleed air enters the split regenerator. Cooling air for the split regenerator consists of a portion of the turbine discharge air which is ducted overboard after two passes through the split regenerator. The bleed air must be cooled to a temperature low enough to condense most of the ambient humidity so that all but 50 gr/lb dry air maximum remains in the air stream. The water removed is sprayed into the cooling air at the beginning of its second pass. It is expected that icing at the turbine discharge will not present a problem since the moisture content will be low. A valve in the regenerator cooling air line insures adequate water removal by controlling the regenerative heat exchanger outlet temperature to a desired value.

The fourth system, the bootstrap regenerative with cabin exhaust cooling, is a modification of the third and is shown in schematic form in Figure 7. This system utilizes compartment air for regenerative cooling flow in the same manner as the simple regenerative system with cabin exhaust cooling. The discussion relating to this type of regenerative cooling as presented for the simple system is also applicable to the bootstrap regenerative system.

A fifth system under consideration is a dual air cycle regenerative system. The system is shown in schematic form on Figure 8. In this system, one of the air cycle machines supplies cooling air for the regenerative heat exchanger. An advantage for this system is the fact that the regenerator flow ratio (cooling air flow: bleed air flow) can be more than 1:1, instead of about 0.5:1, as in the split regenerative systems mentioned, resulting in an appreciably higher value of regenerative heat exchanger performance. Also, the turbine discharge that cools the bleed air in the second heat exchanger could be colder than the cabin temperature in those systems using cabin exhaust cooling. However, the weight and space required could constitute a serious disadvantage. For conditions where the moisture content of the air is very low, the discharge of both turbines could be put into the cabin.

While the finned surfaces of the regenerative heat exchanger in the systems considered are in themselves excellent separators of condensed moisture, there is still a question whether the amount of condensate handled can be adequately removed. Therefore a water separator should be specified for these applications only when necessary as is indicated in Figures (4-8).

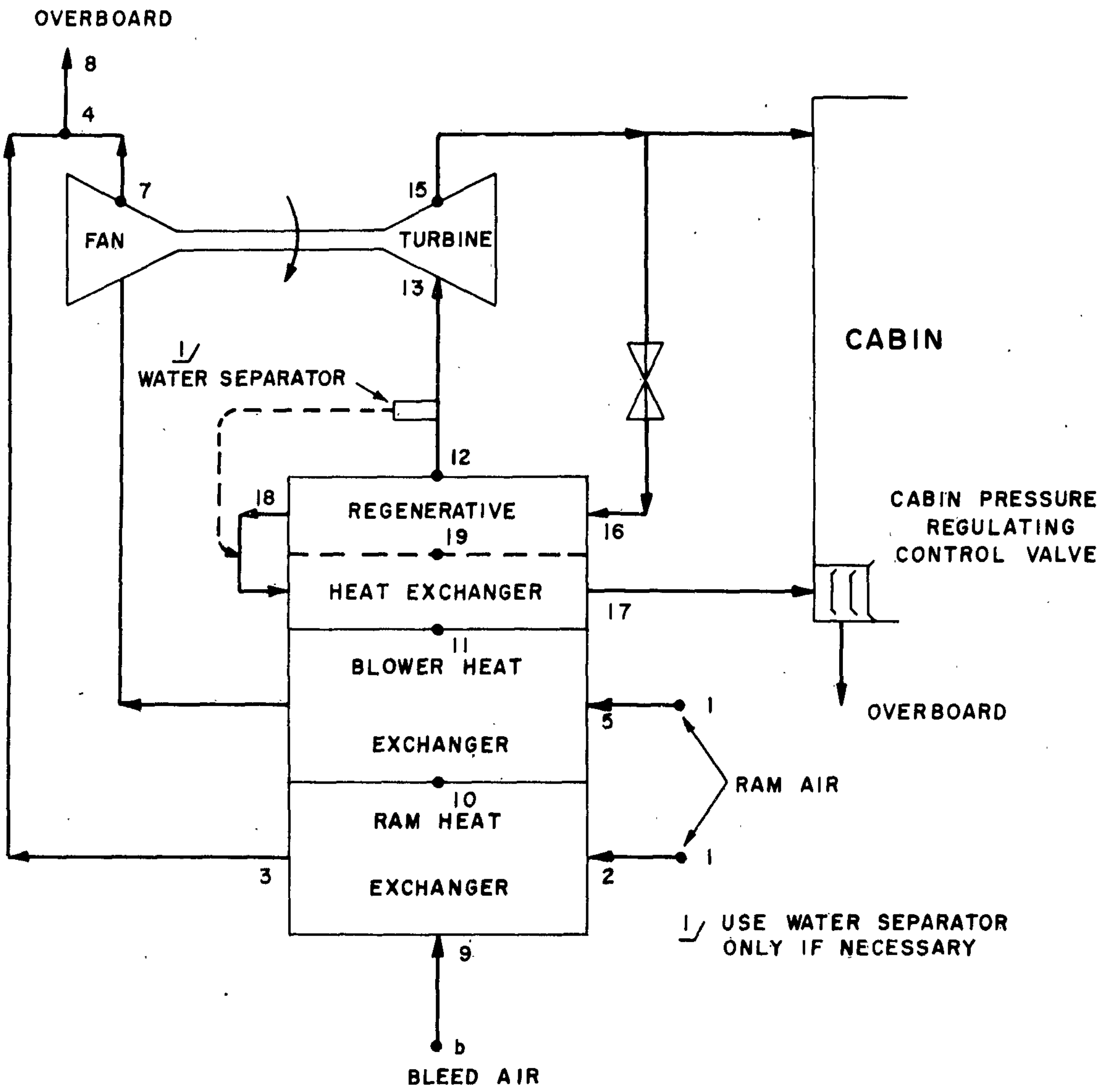


Figure 4. Simple Regenerative Air Cycle with Split Regenerator and Spray Cooling of Regenerative Flow

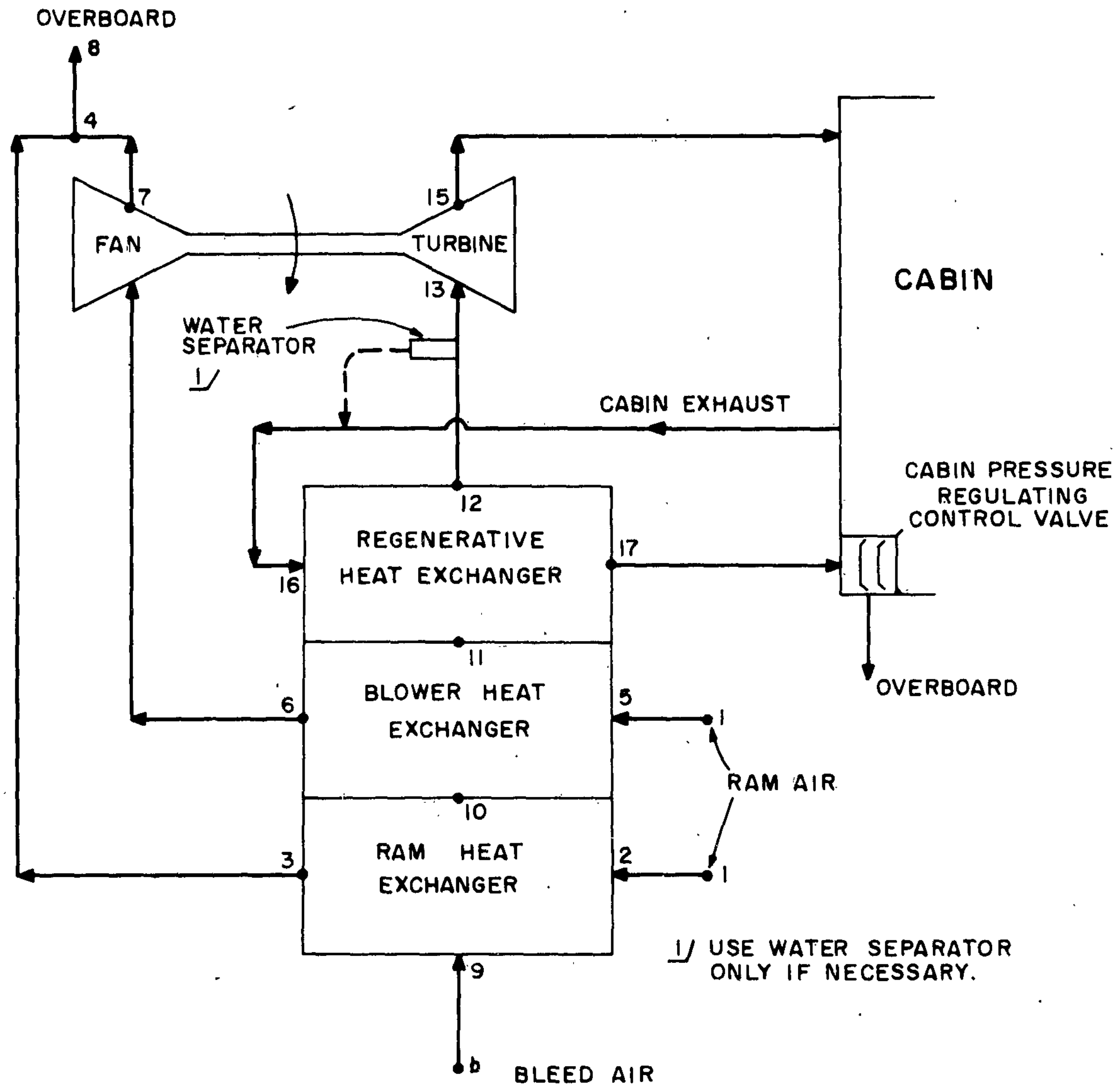


Figure 5. Simple Regenerative Air Cycle with Cabin Exhaust Cooled Regenerator

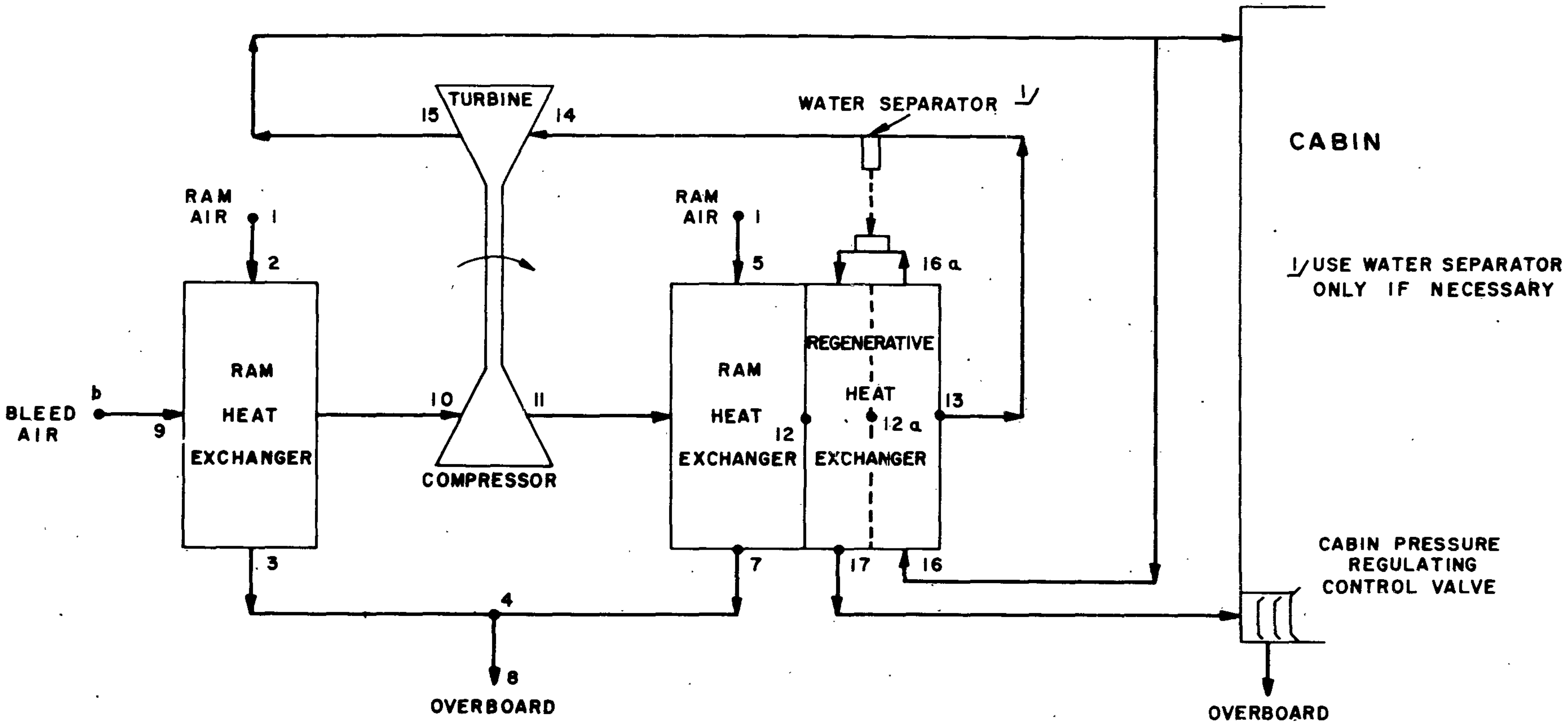


Figure 6. Bootstrap Regenerative Air Cycle with Split Regenerator and Spray Cooling of Regenerator Flow

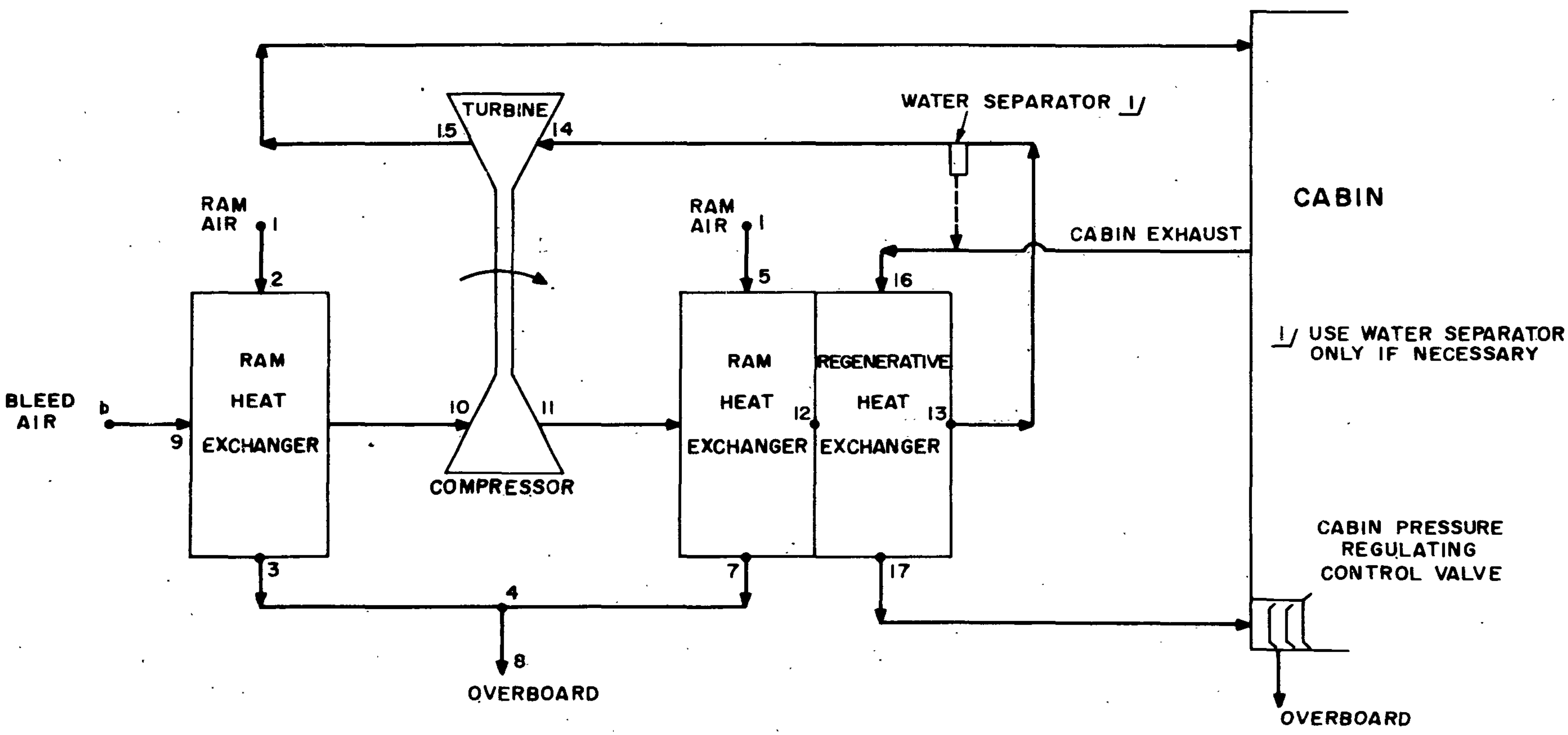


Figure 7. Bootstrap Regenerative Air Cycle with Cabin Exhaust Cooled Regenerator

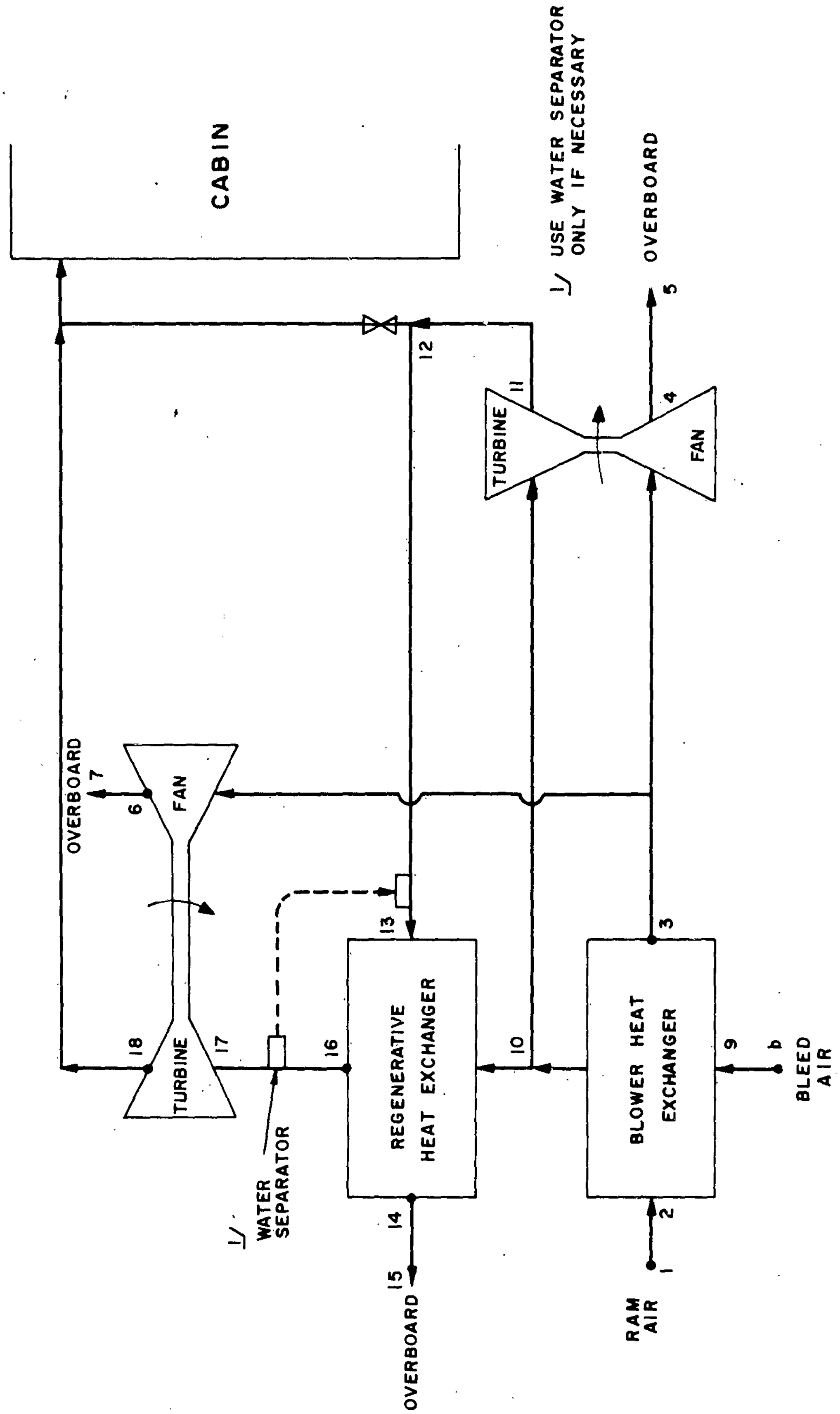


Figure 8. Dual Regenerative Air Cycle

Reasonable ranges of performance data for the components have been assumed. These performance data include the following:

Blower heat exchanger effectiveness.	.79
Ram heat exchanger effectiveness.	.85
Turbine efficiency.	.75 to .80
Compressor efficiency.	.70 to .75

Engine bleed and cooling air conditions per Figures 31, 32, 33.

On the basis of the above performance data, and the prescribed operating conditions it is possible to make a preliminary determination of the suitability of the systems described for use as high performance cooling systems.

The possibility of satisfying the requirement for a maximum of 50 gr/lb of moisture discharged into the compartment must be determined. For this condition the required performances of the moisture separator and the regenerator were calculated. These calculations merely indicate the required regenerator effectiveness for a range of separator efficiencies to limit the moisture content of the cabin air to 50 gr/lb maximum.

While this does not necessarily imply that such performances are possible, these calculations do indicate the relative merits of the system under consideration when operated under conditions of high humidity. Should the values of required effectiveness prove unattainable for the range of possible separator efficiencies, the system can be discarded as being contrary to the limitations imposed. Figure 9 illustrates the range of required regenerator effectiveness values for various values of moisture separator efficiencies for sea level operation at flight Condition A. Inspection reveals that the bootstrap regenerative and the simple regenerative systems with split regenerator, in the order listed, have the best chance for sucessful operation at the given bleed conditions. The remaining systems investigated require effectiveness values that appear to be unattainable under condensing operations despite the fact that each of them operates with flow ratios of at least 1:1. Therefore, because of inadequate moisture removal ability, the dual cycle regenerative and the simple and bootstrap regenerative systems with cabin exhaust cooling are rejected. It should be noted that the reason for the unsuitability of cabin exhaust cooling is that the cabin exhaust temperature of 80° F is too high. The dual cycle system is eliminated because the exit temperature of the turbine that supplies cooling air is very high with the large amount of moisture that condenses in the turbine.

Since the most effective systems are either the simple regenerative or bootstrap regenerative together with a split regenerative heat exchanger, a comparison of the design aspects of the bootstrap system and the simple system is desirable to aid in the selection of the high performance cooling system. The bootstrap cycle, due to its bleed compressor action, has a higher turbine pressure ratio and supplies colder air than a simple system. This represents a saving on the total amount of bleed air flow if the air flow supplied is still larger than the flow required for pressurization. Due to the higher turbine inlet pressures, a smaller nozzle area is required, for the same weight flow as a simple system. In order to maintain proportional design, smaller wheel diameters are used. This increases the rotational speed required to maintain the same blade velocity. While ground cooling is easier to achieve with a simple system, ground cooling is possible but more difficult with the bootstrap. The provision of a separate blower or the use of a jet bleed augmenter are just two possibilities for ground cooling. The addition of a separate blower to provide cooling air on the ground constitutes a weight disadvantage for the bootstrap system, which is partly overcome by the smaller and lighter air cycle machinery of the bootstrap system.

Figure 9 indicates that the bootstrap regenerative system offers the best chance of successfully removing the required amounts of moisture, since the moisture is removed at approximately the same temperature as in a simple regenerative system but at higher pressure.

Because of the superior performance of the bootstrap regenerative system, it appears that it should be the only system considered for additional study for the problem as stated. The bootstrap regenerative system, therefore, will be investigated under various operating conditions with the use of practical component characteristic data.

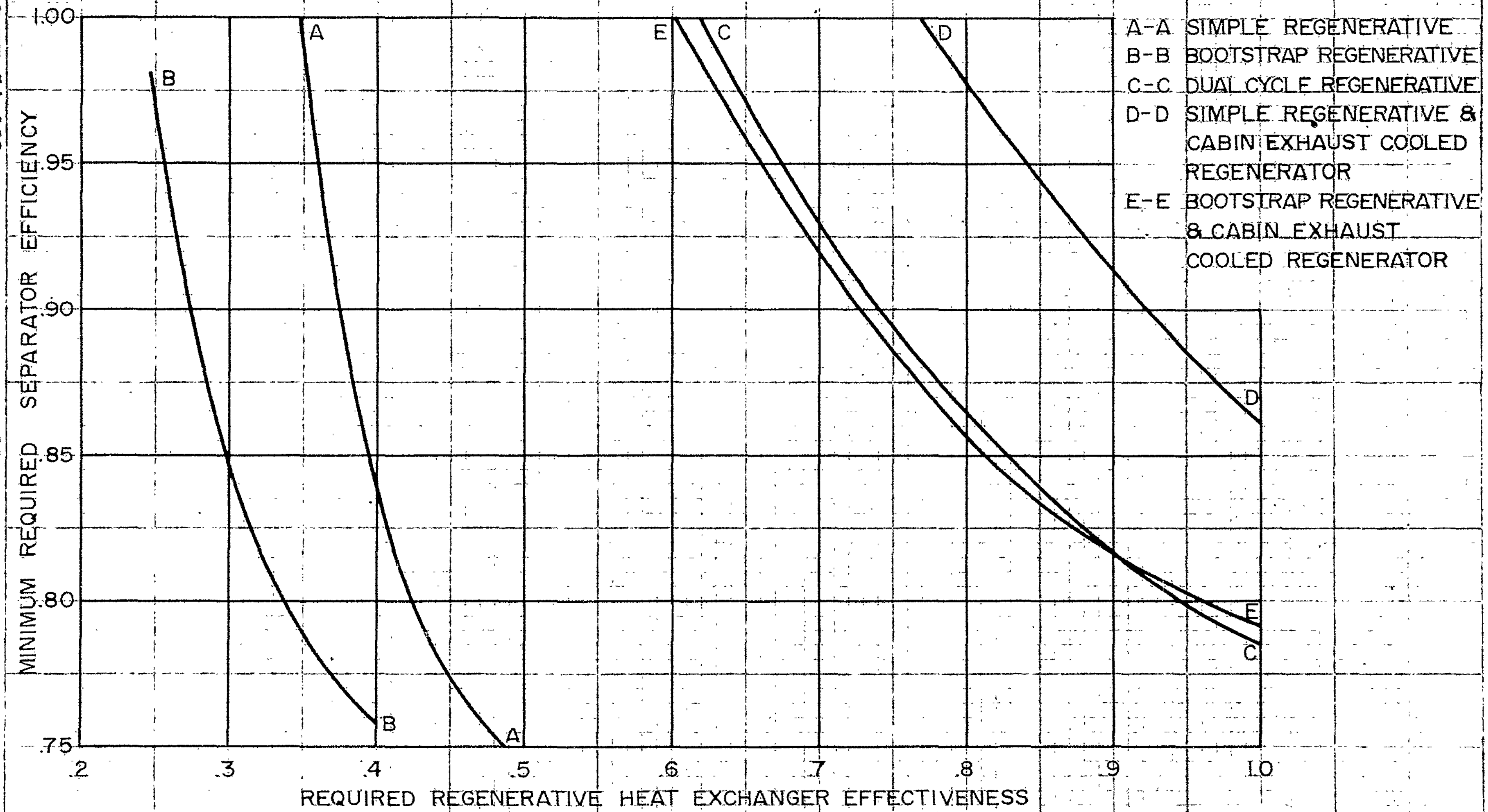


Figure 9. Required Regenerative Heat Exchanger and Water Separator Performance

SECTION IV

DETAIL DESCRIPTION OF THE HIGH PERFORMANCE COOLING UNIT

The bootstrap regenerative system, with split regenerator, and with provision for spray cooling of the regenerative cooling air flow, is considered the system best suited for meeting the objective for a high performance cooling unit based on the operating conditions as specified. These requirements are given in detail in Section III of this report, and are stated here briefly as:

(1) Maximum moisture entering the cabin is 50 gr/lb dry air.

(2) Minimum engine bleed when compared with conventional cooling systems of equal cooling capacity.

(3) Size, weight, and cooling flow drag of the unit shall be such as to be practical for use on present day aircraft.

To provide for ground cooling, some auxiliary means must be included. An auxiliary turbine driven blower or a jet bleed augmenter that operate only on the ground are but a few of the methods available. Their addition of course increases the overall size and weight of the cooling package. For this high performance cooling unit the auxiliary blower is considered.

A schematic drawing of this system 1/ is shown in Figure 10. A general description of the system (bootstrap regenerative with split regenerator) may be found in Section III. It is, therefore, appropriate to continue with a detailed description of the components and controls that comprise the High Performance Cabin Cooling Unit, hereafter referred to as HPCC.

The HPCC consists of two ram operated heat exchangers, a split regenerative heat exchanger, an expansion turbine driving a bleed air compressor, an auxiliary ground cooling blower, a spray system and, if necessary, a moisture separator, together with the required connecting ducts and controls.

All of the heat exchangers are conventional, brazed aluminum extended surface heat exchangers 2/. The two ram operated heat exchangers are identical three-cabin air pass, cross-counter flow

1/ The auxiliary ground cooling blower is not included in the schematic for simplification of presentation.

2/ Reference No. 6.

exchangers. The nominal overall dimensions of each ram heat exchanger are 9 x 10-3/4 x 10-3/4 inches. The split regenerator is essentially two series exchangers. The first section is a two-cabin air pass, cross-counter flow exchanger, while the second is a simple cross-flow exchanger. Their nominal overall dimensions are 6-5/8 x 9-3/4 x 12-3/8 inches and 5-1/2 x 11-7/8 x 12-3/8 inches respectively.

The air cycle machine consists of a full-bladed, fixed area centripetal turbine driving a centrifugal compressor. The turbine has a wheel diameter of 3.8 inches and a nozzle area of 0.2677 square inches. The estimated performance of the turbine is given in Figure 11. Performance calculations discussed in the following pages reveal that the required refrigeration load and pressure scheduling of the airplane are such that the turbine nozzle area, sized at the design point, will successfully control the air flow to the unit for all altitudes and all flight conditions. This factor, of great importance, results in a relatively simple control system, eliminating any need for a variable area nozzle or a secondary flow control valve.

The compressor is a conventional centrifugal type with a wheel diameter of 3.85 inches. The estimated compressor calibration is given in Figure 12. Based on the performance calculations that are discussed in the following pages, compressor operation has been found to be satisfactory for all altitudes and all flight conditions. This factor obviates the need for internal control regulation such as prerotating vanes or variable diffuser vanes.

It is clear from the above discussion that internal control methods for the turbomachinery components of this high performance cooling unit are not warranted.

Referring to Figure 10, the regenerative cooling flow is shown being discharged overboard via the cabin pressure regulating valve. Should this be impracticable, a flow-limiting valve must be provided in this duct to reduce the regenerative cooling flow when the airplane cabin is pressurized. Otherwise no air would flow into the cabin. Since no information of the specific cabin pressure regulating valve geometry is available, the simpler plan is employed.

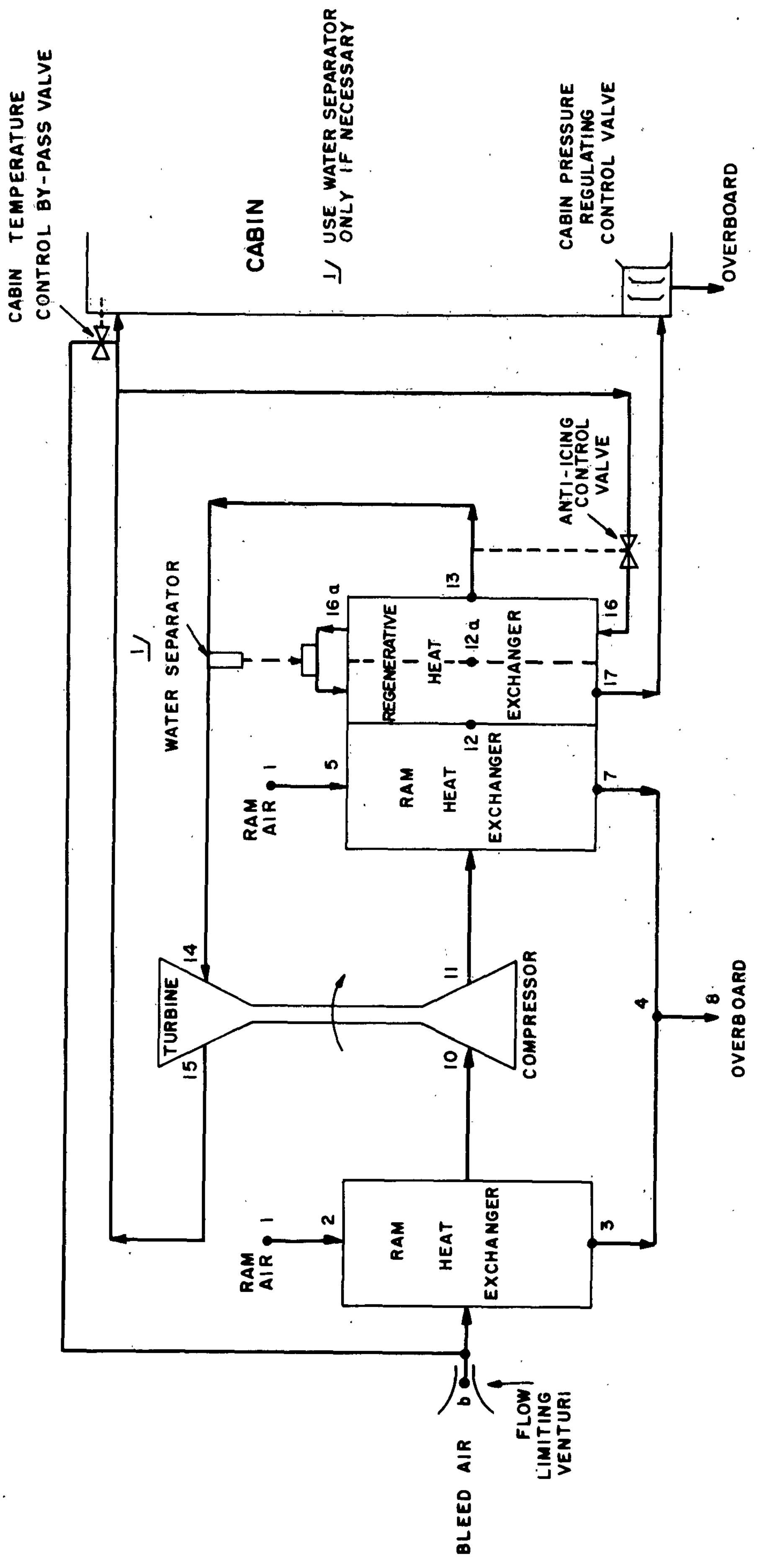


Figure 10. Schematic Diagram of High Performance Cooling Unit with Controls

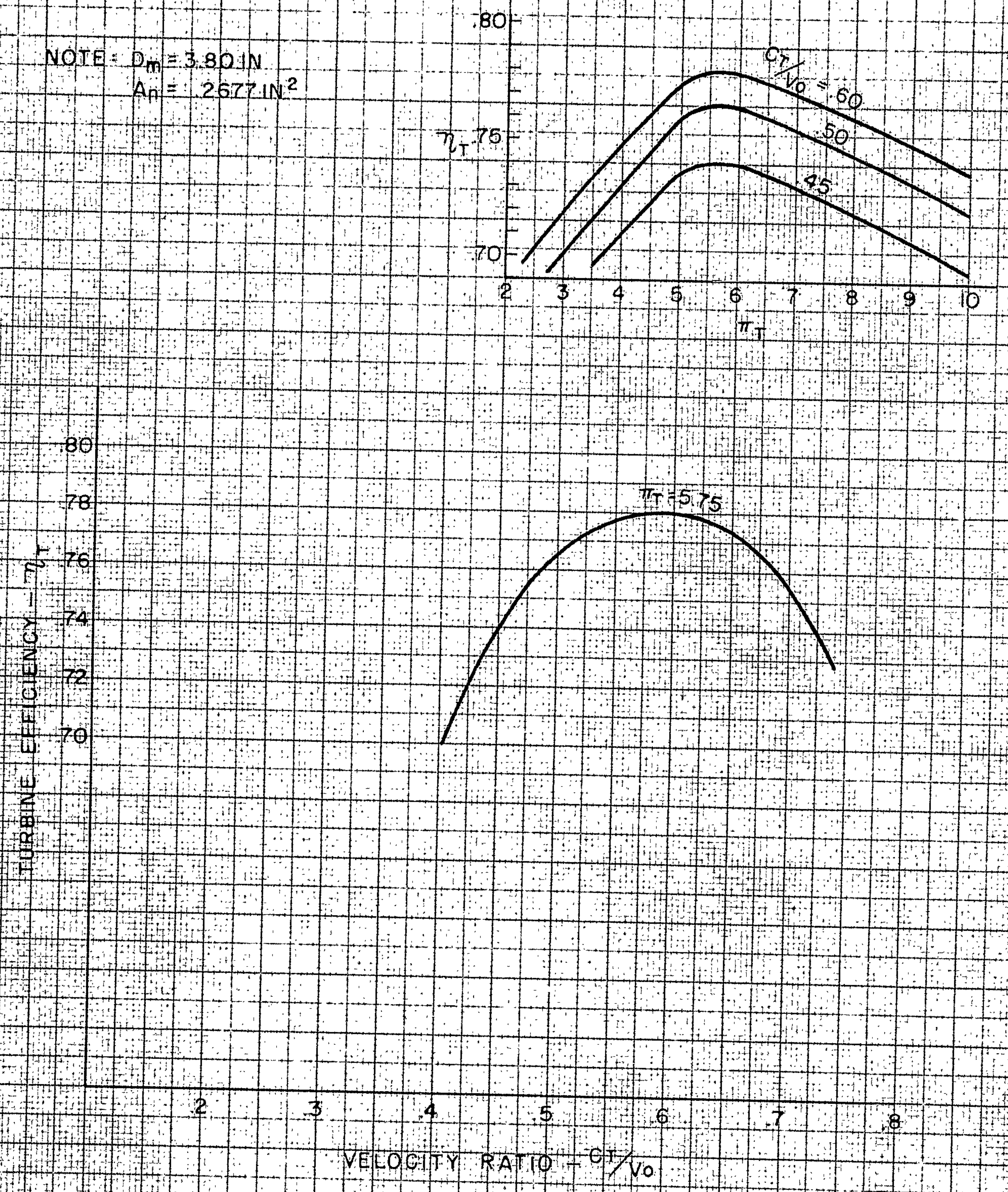


Figure 11. Estimated Turbine Performance

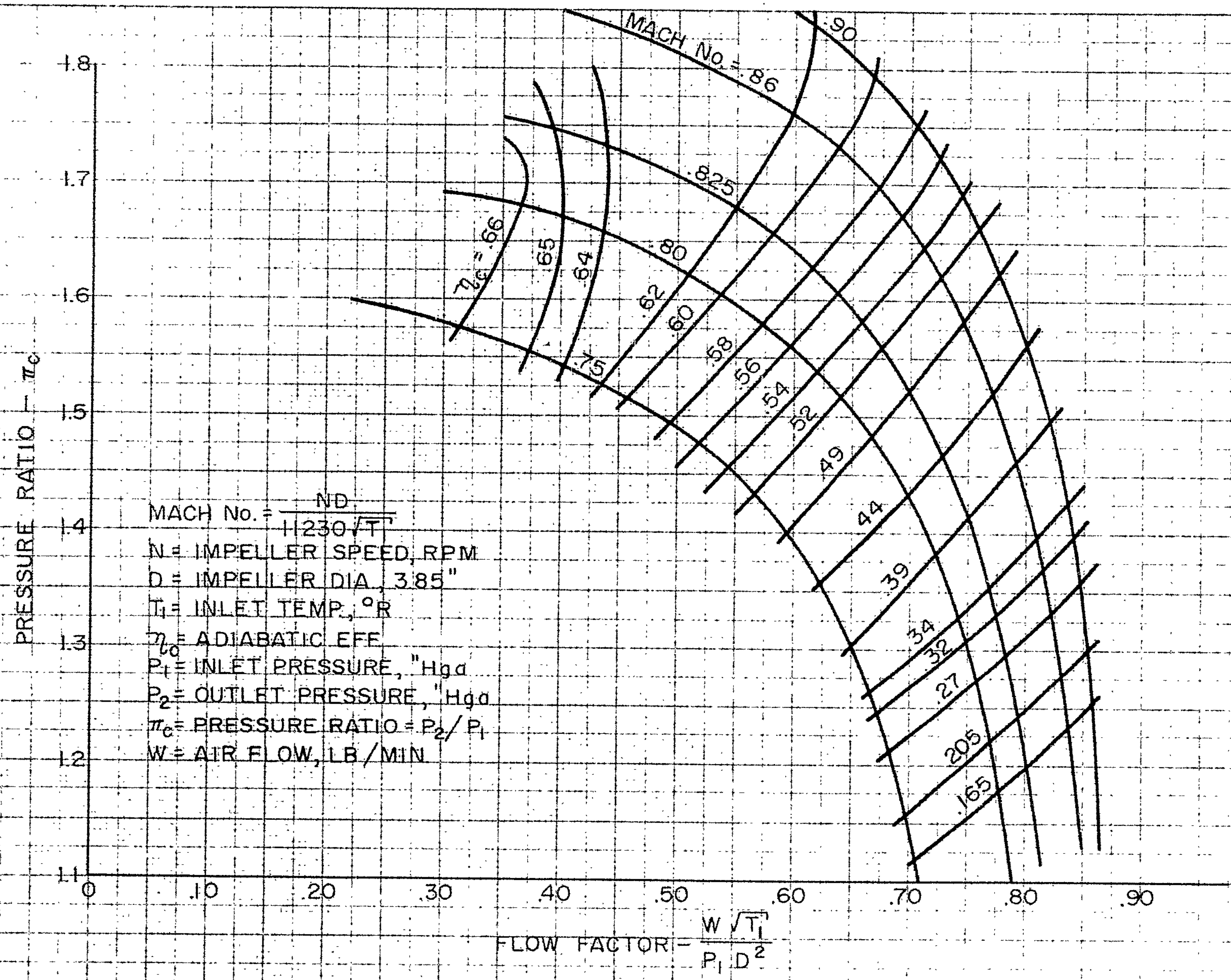


Figure 12. Estimated Compressor Calibration

From the foregoing considerations, regulation of the HPCC by separate control methods is required. The control components are of the same type as are used on conventional cooling systems, and are described as follows:

(1) Cabin temperature is controlled by flow in the hot air duct which by-passes jet engine bleed air directly to the cooling unit discharge duct. The by-pass valve controlling the by-pass flow is responsive to cabin temperatures.

(2) The problem of cabin humidity control has been reduced to that of effectively transmitting no more than 50 gr/lb dry air to the cabin. It is not advisable, in view of the added control complexity, to control cabin humidity any closer. Based on the performance calculations to be discussed in the following pages, a fixed restrictor inserted in the duct system from the turbine discharge to the cabin and an anti-icing control valve in the cooling air flow line of the regenerative heat exchanger insure that no more than 50 gr/lb dry air enters the cabin. The fixed restrictor is required to permit adequate cooling flow to pass through the relatively high resistance of the split regenerator. The anti-icing control valve is responsive to the temperature of the bleed air at the moisture separator. When this temperature falls below 35 ^{o}F, the valve begins to close, reducing the cooling air flow and the overall regenerative effectiveness of the split regenerator. This action increases the bleed air temperature leaving the split regenerator and entering the moisture separator.

(3) Cabin pressure control is accomplished by a conventional cabin pressure regulator, relief and dump valves. Adequate compartment flow for pressurization and ventilation is provided for by the fixed turbine nozzle area which was sized for design conditions. At high altitudes and all flight conditions, satisfactory operation is assured for the following reasons. Referring to the performance tabulation, it can be seen that during high altitude flights the unit operates with an excess of cooling capacity, resulting in low cabin temperatures. Likewise, the temperature at the moisture separator inlet falls below 35 ^{o}F. The package by-pass valve, responding to the low cabin temperatures, opens, permitting hot air to enter the cabin through the cooling unit discharge duct. The anti-icing control valve, trying to maintain the moisture separator inlet temperature above freezing, closes completely, preventing any regenerative action. Thus, both the complete turbine flow and the hot air by-pass flow serve to pressurize and ventilate the cabin. Calculations indicate that, under maximum design cooling loads, the combined flow is adequate for ventilation and pressurization.

(4) There is no need for a speed control device for this unit. The air cycle machine operates at a speed resulting from a balance between turbine and compressor power, as determined by the other system variables, including turbine flow, compressor pressure ratio, turbine expansion ratio, compressor and turbine efficiency. Referring to the performance calculations, it was found that the resultant speed varied with altitude and engine setting in a limited region.

(5) A sonic venturi, located at the inlet to the cooling unit limits the maximum amount of air that can be bled from the engine. This is done to prevent excessive uncontrolled bleed from the jet engine in the event of any damage to the cooling unit or cabin.

The performance of the HPCC was investigated for various altitudes at flight Conditions A and C. The results are tabulated in Table 1. It should be noted that the design yielded excess cooling capacity for all points investigated, with the exception of 25,000 feet altitude at flight Condition C. If further reduction of engine bleed flow is contemplated, by reducing turbine nozzle area, there will be a deficiency in cooling capacity at that operating point. For this reason, the excess of cooling capacity at all other points was tolerated.

Based on pressurization schedules, a critical point exists at 45,000 feet altitude for flight Condition C. For this condition the turbine flow is 5.95 lbs/min. This is less than the prescribed minimum flow for maintaining cabin pressurization equal to 7.8 lbs/min (see Figure 13). However at this condition, due to the excess cooling capacity, cabin temperatures fall below the design value of 70°F. The temperature control valve opens to permit hot air to by-pass the unit and thus raise the cabin temperatures. Calculations indicate that approximately 3.40 lbs/min of hot air by-pass the cooling unit in this way. The total cabin flow then becomes 9.35 lbs/min.

In order to indicate more clearly the inherent advantages and disadvantages of the HPCC, a comparison with a simple system of approximately the same dry rated cooling capacity is desirable. The simple system selected for comparison includes an air cycle machine and a blower operated heat exchanger. The air cycle machine consists of a turbine driving a blower. Air is bled from the jet engine and is first cooled in the heat exchanger. Cooling air from the heat exchanger enters the turbine driven blower and is discharged overboard. The bleed air, somewhat cooled, expands in the turbine, reducing its temperature substantially below ambient. The cold bleed air is then directed into the cabin. A moisture separator is included at the turbine discharge to remove any moisture that is condensed during the expansion.

The comparison of both systems will now be undertaken for flight Condition A, considering the following features:

(1) Cooling capacity.

(2) Engine bleed air.

(3) Aircraft drag and cooling air flow.

(4) Humidity control.

(5) Size and weight.

(6) Pressurization and ventilation.

(7) Ground cooling.

(1) The cooling capacity of the HPCC is much greater than the simple system. This is evidenced in Figure 14, where cooling capacity versus altitude is compared for both systems.

Comparison between the two systems demonstrates the greater capacity of the bootstrap regenerative unit at the low altitudes, where the maximum humidity encountered is high. As the altitude increases and the ambient humidity reduces, this difference diminishes. It is still, however, a sizable factor at 35,000 feet, but this is due essentially to the increased cooling capacity of bootstrap operation over simple system operation, which exists up to moderate altitudes. At very high altitudes, the effect of a reduced speed of the bootstrap unit decreases its overall efficiency, resulting in a cooling capacity comparable to the simple system.

(2) The curves, showing both engine bleed and cabin flow as a function of altitude for both systems are given in Figure 15. It can be seen that the HPCC requires less bleed than the simple system. This reduction is as much as 11 lbs/min at sea level. There are two curves shown for the HPCC, engine bleed and cabin flow. The difference between the two is the regenerative cooling flow.

(3) The drag due to the cooling air requirements of the heat exchangers has been estimated for both systems. Evaluation of this power loss is difficult to make unless certain information concerning system location aboard the aircraft is known. Since this is not the case, these assumptions will be made in the calculation of drag.

(a) 85% ram inlet duct efficiency.

(b) Zero thrust recovery by jet effect of the discharging cooling air.

The cooling air flow and resultant drag vs altitude are shown in Figures 16 and 17 for both systems. The HPCC with two ram heat exchangers requires approximately 80% more cooling air flow than the simple system with one blower heat exchanger. Likewise, the bootstrap unit has, in the same order of magnitude, higher drag than the simple system.

(4) The HPCC, specifically designed to operate satisfactorily even under adverse high humidity conditions, limits the cabin inlet moisture content to 39 gr/lb max (at sea level).

The simple system introduces considerably more moisture to the cabin, depending on the efficiency of the moisture separator employed. Estimates place this value in the order of 120 gr/lb (at sea level).

(5) The size and weight of the HPCC is, of course, much larger than the conventional system, requiring an additional heat exchanger, a split regenerator, and an auxiliary blower for ground cooling. The estimated weight breakdown of both systems are given in Table 2.

(6) Required pressurization and ventilation requirements are satisfied for all flight conditions and altitudes. At the worst possible condition (highest altitude and lowest bleed pressure) 45,000 feet at flight Condition C, satisfactory operation is assured by the cabin temperature control by-pass valve. At very high altitudes the cabin temperatures are so low as to cause the cabin temperature control valve to open. This permits hot engine bleed to by-pass the cooling unit, increasing the cabin temperature. The total flow entering the cabin satisfies the pressurization schedule.

(7) For ground cooling, an auxiliary blower is included in the HPCC. The blower is powered by a suitable drive from the cooling turbine to pass cooling air through the ram heat exchanger. On the other hand, the simple system with a blower operated heat exchanger requires no additional components for ground operation. Besides the additional weight of the secondary blower (10 lbs) there are other disadvantages resulting from the added controls for satisfactory operation.

TABLE I

PERFORMANCE TABULATION OF BOOTSTRAP REGENERATIVE AIR CYCLE WITH SPLIT REGENERATOR, SPRAY COOLING OF REGENERATOR FLOW

1.	Flight Condition	A	A	A	A	A	C	C
2.	Altitude (feet)	SL	5800	10000	30000	45000	25000	45000
3.	Airplane Speed (MPH)	540	600	640	604	573	480	478
4.	Ambient Humidity(gr/lb)	184	184	132	16	9	27.5	9
5.	Bleed Pressure ("Hga)	182.6	158.8	142	68.21	36.44	63.12	27.49
6.	Bleed Temperature (^{O}F)	588	575	561	469	405	395	300
7.	Ram Heat Exch. Bleed Inlet Press. ("Hga)	179.26	155.78	139.38	67.194	36.06	62.36	27.286
8.	Ram Heat Exch. Bleed Exit Press. ("Hga)	173.73	150.78	134.86	65.22	35.242	60.81	26.794
9.	Ram Heat Exch. Effect.	.932	.952	.9675	.9695	.960	.968	.9565
10.	Ram Heat Exch. Bleed Exit Temp. (^{O}F)	181.6	163.5	150.8	69.6	11.4	62.1	-7
11.	Compressor Pressure Ratio	1.71	1.728	1.725	1.61	1.33	1.50	1.175
12.	Compressor Flow Factor	.5213	.5251	.5192	.4522	.3641	.4183	.3188
13.	Compressor Efficiency (%)	62.9	62.8	62.8	63.15	56.3	61.4	51.5
14.	Ram Heat Exch. Bleed Inlet Temp. (^{O}F)	348.8	329.7	313.1	190.6	81.8	165.5	34.1
15.	Ram Heat Exch. Bleed Inlet Press. ("Hga)	297.1	260.5	232.6	105	46.87	91.2	31.48
16.	Ram Heat Exch. Bleed Exit Press. ("Hga)	294.3	258	230.34	104	46.4	90.34	31.152
17.	Ram Heat Exch. Effect.	.933	.953	.9685	.970	.961	.9685	.9585
18.	Ram Heat Exch. Bleed Exit Temp. (^{O}F)	165.2	151.5	142.5	61	-1.3	54.7	-18.7
19.	Second Cooling Pass Split Regen. Temp. Difference Parameter	.584	.576	.562	.217*/	--#/	.219*/	

*/ Anti-icing Control Valve partially closed for this condition.

#/ Anti-icing Control Valve completely closed for this condition.

TABLE I (cont)

20.	Second Cooling Pass Split Regen. Bleed Exit Temp. (°F)	141.3	135.5	122	60	-1.3	53.6	-18.7
21.	Second Cooling Pass Split Regen. Bleed Exit Press. ("Hga)	293.76	257.59	229.9	103.78	46.3	90.14	31.07
22.	First Cooling Pass Split Regen. Temp. Difference Parameter	.346	.336	.324	.18	--	.166	--
23.	First Cooling Pass Split Regen. Bleed Exit Temp. (°F)	48.1	47	40	32	-1.3	32	-18.7
24.	First Cooling Pass Split Regen. Bleed Exit Press. ("Hga)	293.32	257.12	229.52	103.58	46.18	89.97	30.99
25.	Moisture Condensed (gr/lb)	179.1	178.5	127.3	8	9	19.5	9
26.	Separator Efficiency (%)	81	81	81	--	--	--	--
27.	First Cooling Pass Regen. Cooling Air Inlet Temp. (°F)	-128.6	-128.1	-131.7	-95.6	--	-77.8	--
28.	First Cooling Pass Regen. Cooling Air Exit Temp. (°F)	124.2	123.7	106	56.4	--	50.6	--
29.	Spray Efficiency	.727	.727	.70	--	--	--	--
30.	Second Cooling Pass Regen. Cooling Air Exit Temp. (°F)	153.6	142.3	135.4	60.7	--	54	--
31.	Turbine Inlet Press. ("Hga)	292.86	256.7	229.13	103.35	46.04	89.76	30.87
32.	Turbine Inlet Temp. (°F)	48.1	47	40	32	-1.3	32	-18.7
33.	Turbine Exit Press. ("Hga)	31.406	26.57	26.44	23.21	18	24.79	17.74

TABLE I (cont)

34. Turbine Press. Ratio	9.325	9.66	8.665	4.454	2.56	3.62	1.74
35. Turbine Efficiency (%)	74.25	73.8	75.1	75.25	70.1	73.1	67.85
36. Turbine Horsepower	53.19	46.74	40.96	13.77	3.73	10.29	1.468
37. Compressor Horsepower	50.33	44.37	38.72	13.06	3.50	9.69	1.388
38. Turbine Nozzle Area (in^2)	.2677	.2677	.2677	.2677	.2677	.2677	.2677
39. Turbine Exit Temp. (°F) (dry air basis - full cold)	-128.6	-128.1	-131.7	-95.6	-76.6	-77.8	-62.1
40. Cabin Air Weight Flow (#/min)	21.5	20.8	20	14.52	8.76	12.16	5.95
41. Regenerative Cooling Air Weight Flow (#/min)	31.5	26.2	22	4.48	0	4.34	0
42. Total Bleed Weight Flow (#/min)	53	47	42	19	8.76	16.5	5.95
43. Ram Air Inlet Pressure ("Hga)	39.18	34.127	30.73	13.52	6.857	14.187	6.054
44. Ram Air Inlet Temp. (°F)	152	142.7	137	57	-5	51.1	-21
45. Total Ram Cooling Air Flow (#/min)	341	328	312	138	70	121	54.9
46. Turbine, Compressor Speed (rpm)	61,250	60,817	60,112	52.620	39,895	48,990	31,594
47. Cabin Temp. (°F)	80	80	80	70	70	75	70
48. Cabin Air Inlet Humidity (gr/lb)	39	39.4	28.9	16	9	27.5	9
49. Cooling Capacity BTU/hr	72,730	70,180	66,600	33,800	18,780	6980	11,400
50. Excess Cooling Capacity (%) Above Design Conditions	11.9	11.4	13	12	43	1.04	100

TABLE 2

ESTIMATED WEIGHT BREAKDOWN COMPARISON BETWEEN HPCC AND SIMPLE COOLING SYSTEM

Item	No. Req'd	HPCC	Simple System
Blower Heat Exchanger	1	-	20
Ram Heat Exchanger	2	30	-
Split Regenerator	1	24	-
Air Cycle Machine	1	22	24
Moisture Separator	1	4	6
Spray System	1	2	-
Connecting Ducts	-	6	3
Cabin Temp. Control Valve	1	2.5	2.5
Flow Limiting Venturi	1	2	2
Anti-Icing Control Valve	1	1.5	1.5
Shut-Off Valve	1	2	2
Auxiliary Ground Cooling Blower, Drive, and Controls	1	10	-
Total		106 lbs.	61 lbs.

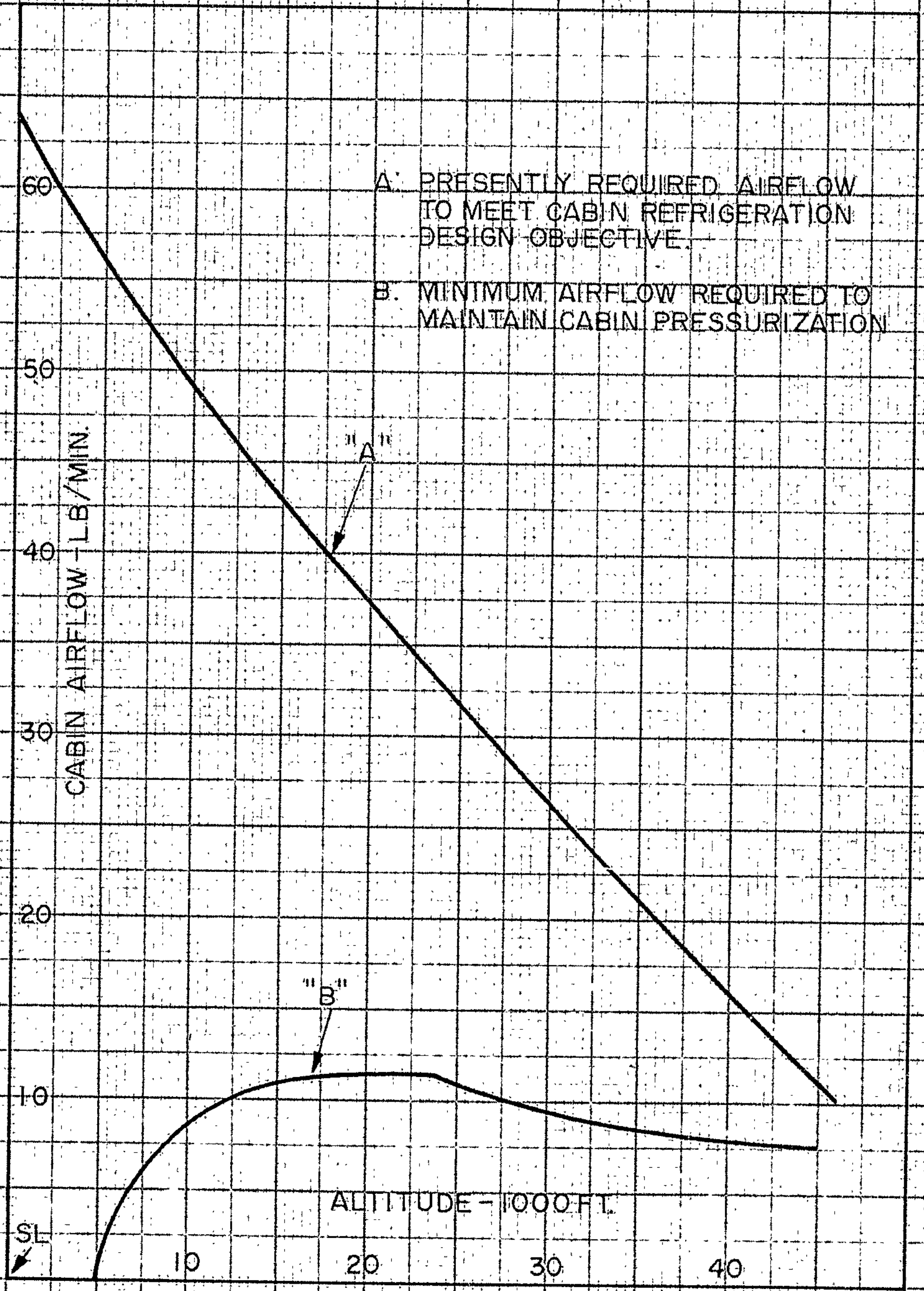


Figure 13. Existing Cabin Air Flow Requirements

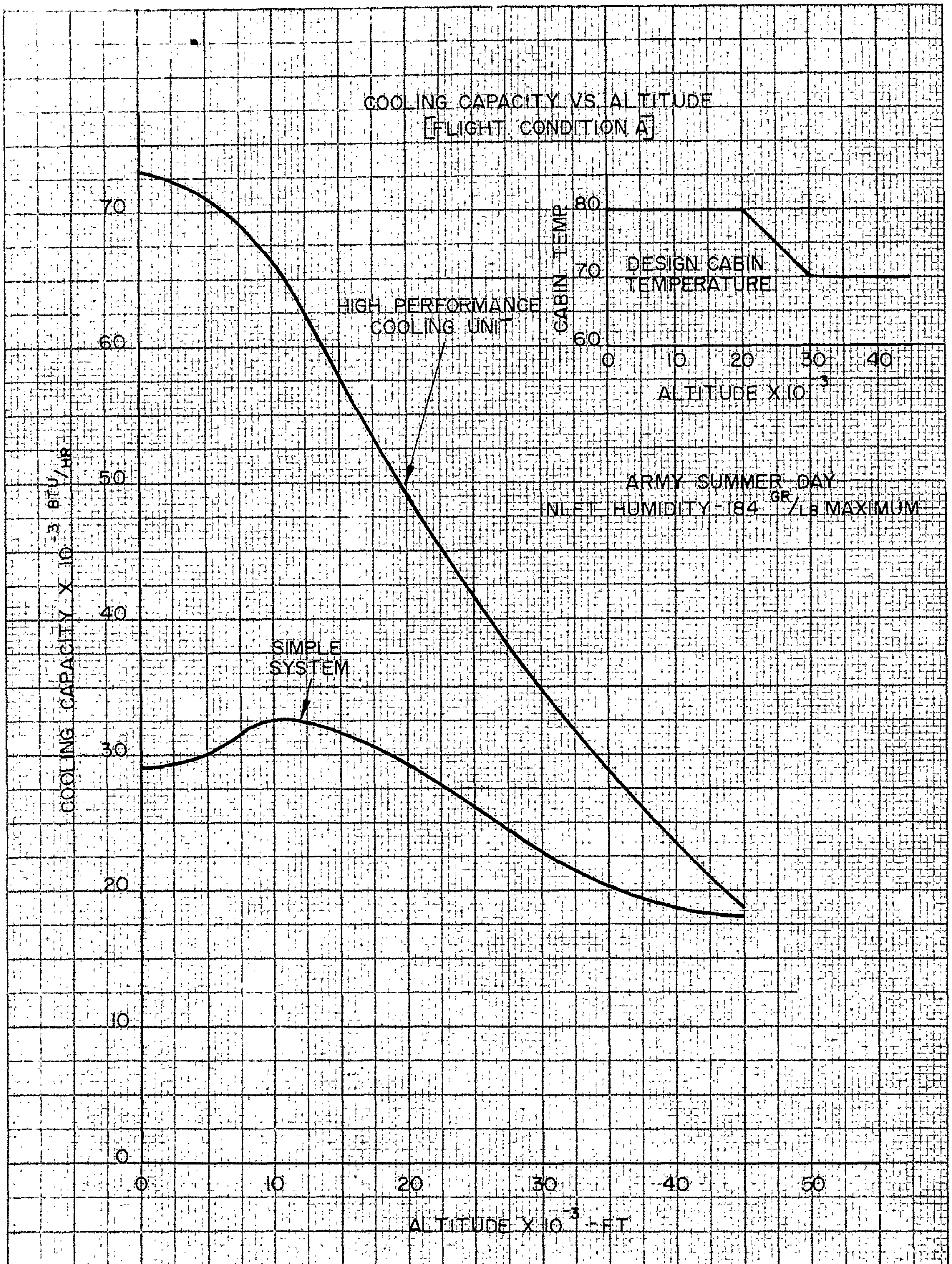


Figure 14. Comparison of Estimated Performance of High Performance Cooling Unit with Simple Cooling System

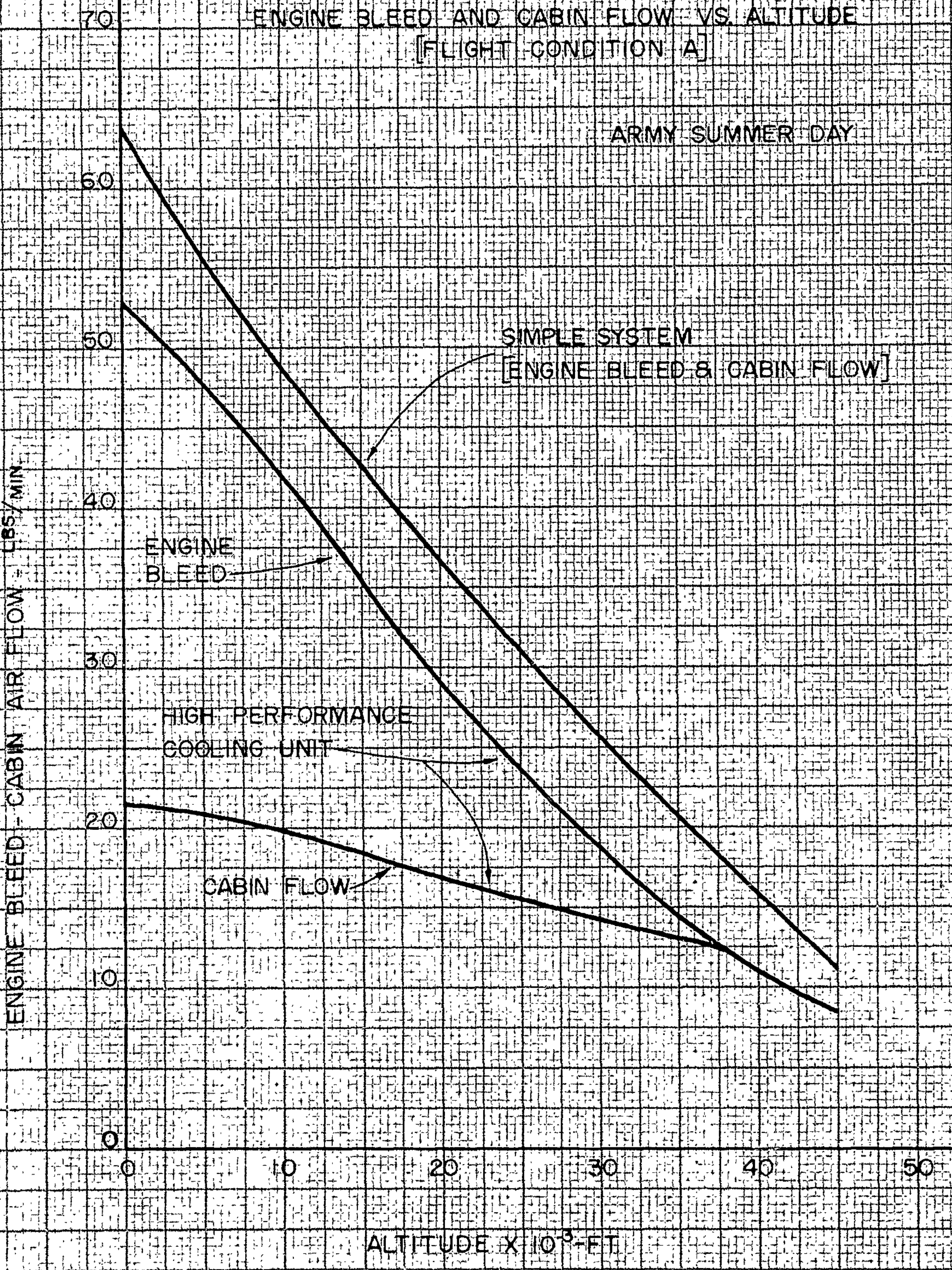


Figure 15. Comparison of Estimated Performance of High Performance Cooling Unit with Simple Cooling System

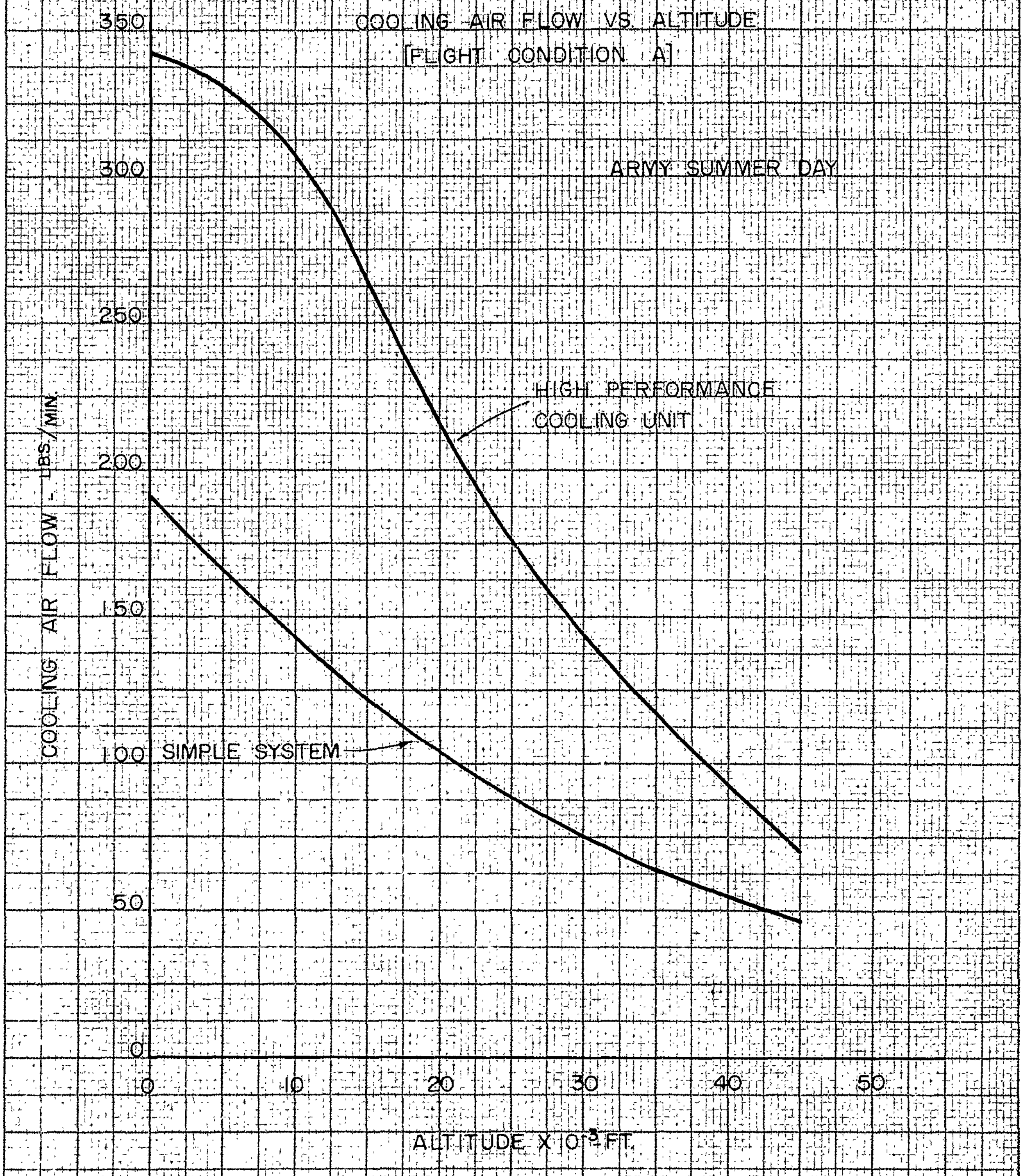


Figure 16. Comparison of Estimated Performance of High Performance Cooling Unit with Simple Cooling System

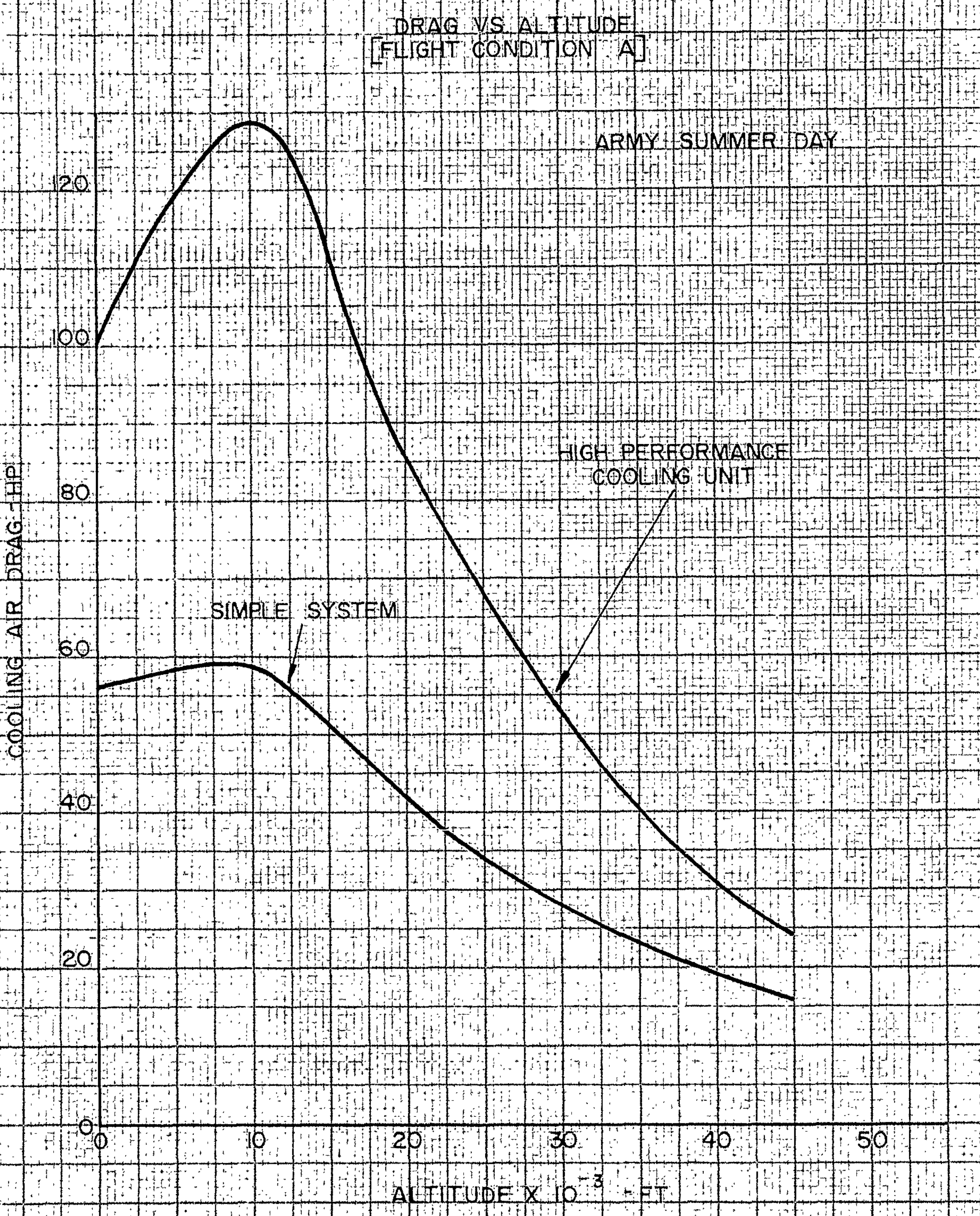


Figure 17. Comparison of Estimated Performance of High Performance Cooling Unit with Simple Cooling System

SUMMARY

Many cooling systems used in present day airplanes do not deliver maximum cooling capacity during high ambient humidity conditions. A reason that full cooling capacity cannot be realized is that turbine discharge temperatures must be maintained above 32°F to prevent the formation of ice in the discharge duct.

An increase in cooling capacity can be obtained if moisture is condensed and removed from the jet engine bleed air at a point other than at the lowest temperature in the system. For example, removal of moisture before expansion in the turbine section of the air cycle machine, followed by subsequent expansion of the dried bleed air, would permit tolerance of very low turbine discharge temperatures.

The object of this study is to demonstrate the thermodynamic possibilities of air cycle refrigeration systems with regenerative cycle configurations and with features allowing the control of excess air moisture on the high pressure side of the cycle.

This report is concerned with the methods employed in development of high performance cooling systems. Therefore methods that enable the removal of moisture from the jet engine bleed air, such as the incorporation of regenerative heat exchangers using turbine discharge air or cabin exhaust air for the cooling medium, are examined and discussed.

An investigation of literature relating to airplane air conditioning indicated a paucity of information and sometimes conflicting data in the range of pressures and temperatures encountered. Therefore it was considered unwise to extrapolate such information. Consequently in order to determine the performance of a "wet heat exchanger" (a heat exchanger in which condensation of moisture from the air stream is taking place), an exact theoretical analysis was made. Subsequently tests on a typical extended surface heat exchanger were conducted. Such tests substantiated the theoretical analysis. Empirical data derived from these tests were employed in a simplified analysis to determine the performance of a "wet" heat exchanger.

This study considered the following systems as offering the best possibilities for high performance operation:

(1) Simple regenerative with split regenerator.

(2) Simple regenerative with cabin exhaust cooling in the regenerator.

(3) Bootstrap regenerative with split regenerator.

(4) Bootstrap regenerative with cabin exhaust cooling in the regenerator.

(5) Dual cycle regenerative.

The results of the study, applicable for the particular conditions as specified herein, indicate that the most effective system is the bootstrap regenerative system with split regenerator. The study further indicates the system requires less bleed flow than systems of the presently used type over the complete range of operational conditions.

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APPENDIX 1

HEAT EXCHANGER DESIGN

A major difficulty in predicting the performance of the high performance system selected is the scarcity of data on heat exchangers under the conditions of operation encountered. The regenerative heat exchanger will operate with condensation on the cooled air side, evaporation on the cooling air side, and possibly with freezing of the condensed moisture. A search of the literature indicated that most of the information available is not applicable to the solution of this particular problem, since the range of pressures and temperatures encountered in aircraft air conditioning is different than used in the commercial fields. An analytical examination was undertaken, therefore, to enable prediction of "wet" heat exchanger performance under the specific conditions applicable.

DRY HEAT EXCHANGER DESIGN

In order to show the major differences between "dry" and "wet" heat exchanger design, a brief review of dry heat exchanger design techniques or methods is desirable.

A design technique useful for accurately estimating size and performance of air to air heat exchangers, operating non-condensing (dry), is available 1/. First the basic heat exchanger variables are specified or assumed. Then, the application of suitable dimensionless parameters is possible. This permits a rather simple mathematical approach to heat transfer, indicating whether the estimated performance is correct.

Heat exchanger variables include the following:

(1) Fluid terminal temperatures, hot and cold side.
(2) Fluid weight flow, hot and cold side.
(3) Fluid flow friction, hot and cold side.
(4) Surface configuration.

For convective heat transfer, the general relationships have the form containing groups of variables in dimensionless form.

1/ Reference 6

This relationship for the turbulent region is given as follows: 1/

$$\left(\frac{hD}{k}\right) = C_1 \left(\frac{GD}{\mu}\right)^m \left(\frac{c_p \mu}{k}\right)^n \qquad (1)$$

For a gas such as air, where the Prandtl number, $\left(\frac{c_p \mu}{k}\right)$ is for all practical purposes independent of temperature, equation (1) reduces to

$$\frac{hD}{k} = C_2 \left(\frac{GD}{\mu}\right)^m \qquad (2)$$

Solving for the pure film coefficient of heat transfer, h.

$$h = \frac{C_2}{D^{1-m}} \left(\frac{k}{\mu^m}\right) (G)^m \qquad (3)$$

For the ranges of temperature and pressure encountered, the convective film coefficient for a given surface configuration is therefore only a function of the unit weight flow raised to the m power.

$$h = C_4 (G)^m \qquad (4)$$

Curves of h as a function of G, for a given heat transfer surface, are shown in Figure 18. 2/

For extended surfaces, a fin efficiency term, η_o 3/ is incorporated to account for the temperature gradient existing along the fins. The effective film coefficient is given as:

$$h_e = \eta_o h \qquad (5)$$

Curves of the h_e as function of G, for a given heat transfer surface, are shown in Figure 19.

The mathematical basis of this design method lies in the determination of $(UA)_{avail}$ and $(UA)_{req'd}$.

1/ Reference 14, page 167
2/ Reference 27, page 10
3/ The method for calculating η_o is given on page 79.

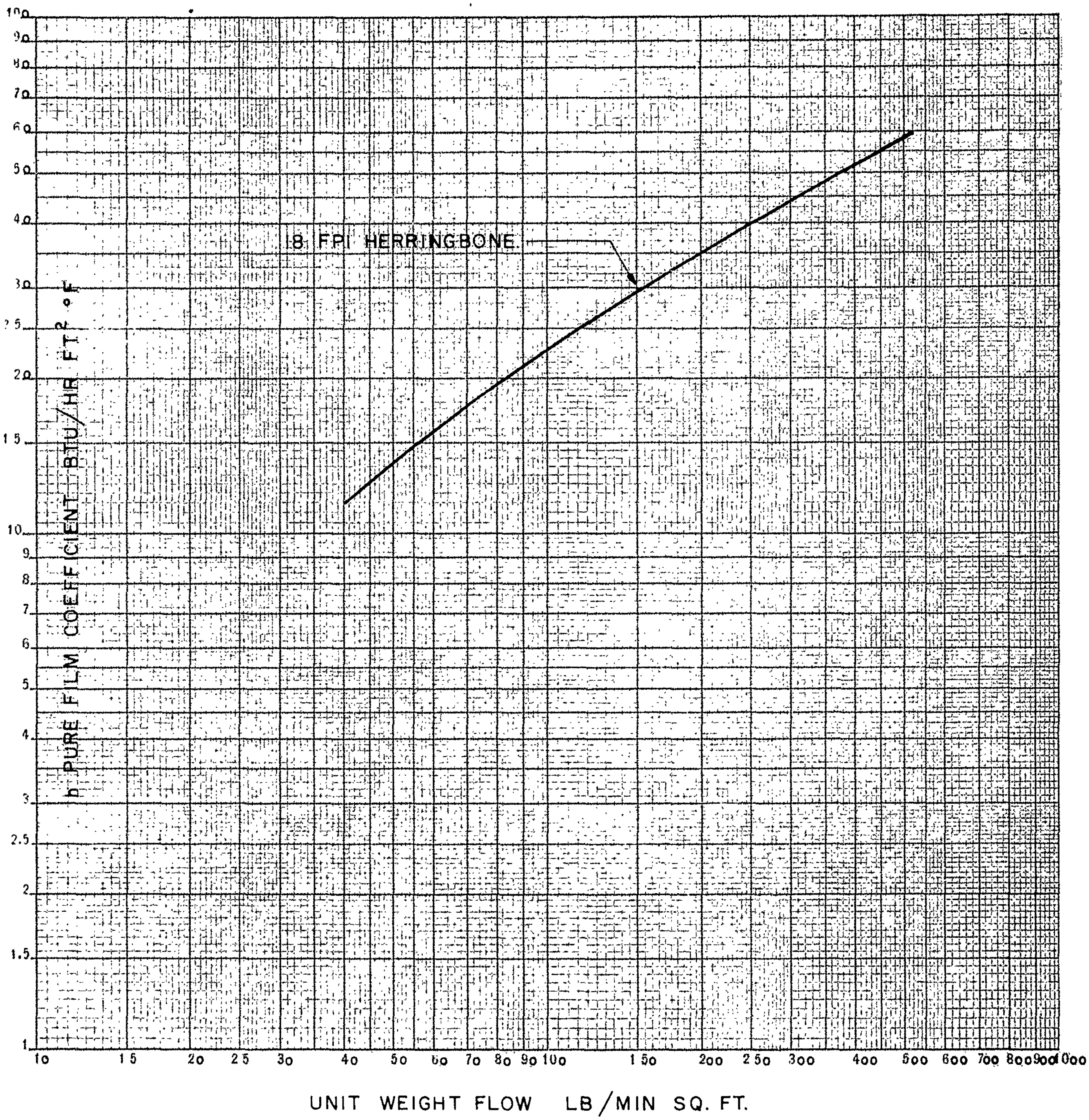
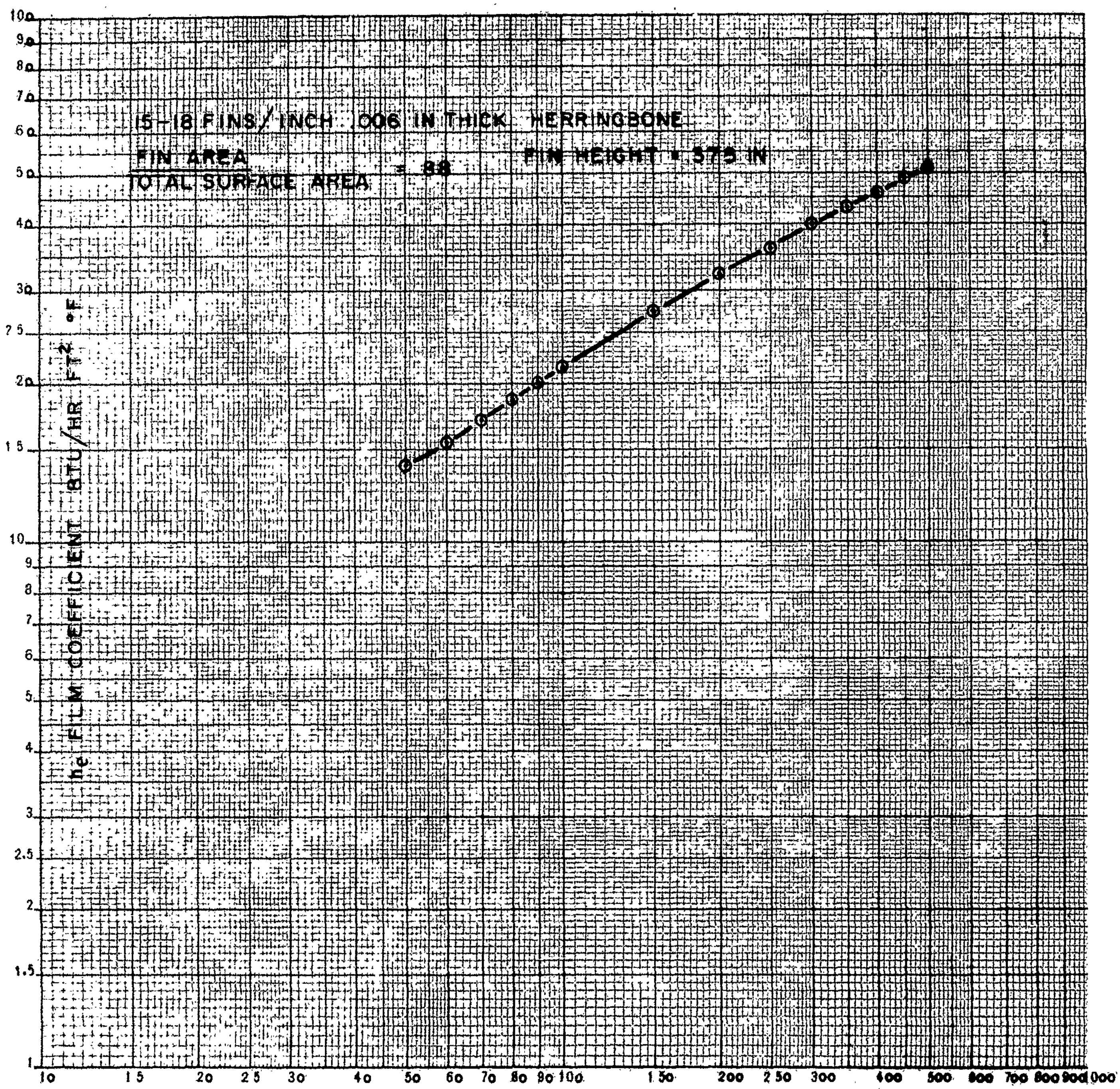


Figure 18. Heat Transfer Coefficients for Brazed Aluminum Surfaces



UNIT WEIGHT FLOW G LB/MIN SQ. FT.

Figure 19. Heat Transfer Coefficient for Brazed Aluminum Extended Surfaces, Dry Air

The reciprocal of $(UA)_{avail}$ is equal to the sum of the reciprocals of the following:

(1) The product of the effective hot side film coefficient and the hot side surface area.
(2) The product of the effective cold side film coefficient and the cold side surface area.
(3) The product of the wall resistance component and the mean wall surface area.

For usual heat exchanger problems, the wall resistance component can be considered negligible when compared to the fluid side resistances. Expressing the above in equation form,

$$\frac{1}{(UA)_{avail}} = \frac{1}{h_{e_h} A_h} + \frac{1}{h_{e_c} A_c}$$

or

$$(UA)_{avail} = \frac{h_{e_h} A_h + h_{e_c} A_c}{(h_{e_h} A_h)\ (h_{e_c} A_c)} \qquad (6)$$

Newtonian heat transfer is assumed. Therefore $(UA)_{req'd}$ is given as:

$$(UA)_{req'd} = \frac{\text{Total Heat Transferred}}{\text{Mean Temperature Difference}} \qquad (7)$$

A balance between $(UA)_{avail}$ and $(UA)_{req'd}$ indicates that the correct performance estimate has been made, or

$$(UA)_{req'd} = (UA)_{avail}$$

"WET" HEAT EXCHANGER DESIGN

The analysis of heat exchanger performance when condensation occurs (wet heat exchanger design) is easily accomplished when considering pure condensable vapor. However where the condensable vapor is mixed with a non-condensing gas (dehumidification), a far more complex situation exists.

For this case, a film of non condensables and vapor forms between the main body of the gas stream and the liquid condensate. The surface temperature of the condensate formed is less than the temperature of the gas stream. Likewise the concentration of vapor in the gas film is less than the vapor concentration in the gas stream.

As evident from the above, the rate of heat transfer from the gas stream to the condensate surface is governed by two forces; sensible heat transfer, due to the temperature gradient, and diffusion or mass transfer resulting from the concentration gradient.

A number of studies 1/ have been made where this phenomenon and the analogous behavior between mass and heat transfer is shown to exist. A brief development of the theory involved is presented herein together with the design technique as recommended by Colburn and Hougen 2/

A dimensionless transformation that is particularly useful in heat transfer work is, for turbulent flow,

$$j_h = \frac{(Nu)}{(Pr)^{1/3}\ (Re)} = \frac{h}{c_p G}\ (Pr)^{2/3} \qquad (8)$$

The importance of using j_h is evident since we avoid calculating k directly (Pr is nearly constant over wide temperature ranges) 3/.

From the conservation of energy

$$Q = w\ c_p\ (t_2 - t_1) = h\ A_s\ \Delta t \qquad (9)$$

Solving for h we obtain

$$h = \frac{w\ c_p\ (t_2 - t_1)}{A_s\ \Delta t} \qquad (10)$$

1/ Reference 1, 2, 3, 4, 10
2/ Reference 2
3/ Reference 14, page 415

Noting that $w = GA_f$ we find

$$\frac{h}{c_p\ G} = \frac{A_f}{A_s} \quad \frac{(t_2 - t_1)}{\Delta t} \tag{11}$$

Substituting (11) into (8), we have

$$j_h = \frac{(t_2 - t_1)}{\Delta t} \left(\frac{A_f}{A_s}\right) \quad (Pr)^{2/3} \tag{12}$$

During condensation, when vapor diffuses from the saturated gas body into the liquid condensate, the mass transfer is unidirectional with mols of vapor transferring from gas to liquid. The differential equation expressing the number of mols transferred is given by

$$dN_d = d\left(\frac{G\ A_f\ \ P}{M_m P_t}\right) = k_g\ \Delta P\ \ dA_s \tag{13}$$

One solution of this differential equation 1/ yields the following equation:

$$\frac{k_g\ \ P_{g_f}}{(G/M_m)} = \frac{P_1 - P_2}{\Delta P} \quad \frac{P_{g_f}}{P_g} \quad \frac{A_f}{A_s} \tag{14}$$

The relationship between heat transfer and fluid friction, as demonstrated in "Reynold's Analogy", and the influence of $\frac{c_p \mu}{k}$ on heat transfer, both may be similarly applied to mass transfer,

In this case, however, skin friction rather than fluid friction applies. A dimensionless modulus relating the fluid properties that affect skin friction is the Schmidt Number, $\mu/\rho k_d$.

Inferring that the influence of $(\mu/\rho k_d)$ on diffusion is comparable to that of $\frac{c_p \mu}{k}$ on heat transfer, the similarity (see equation (12)) is intensified by arbitrarily raising the Schmidt No. to the 2/3 power and multiplying by equation (14).

1/ Reference 5, page 342.

Assuming P_{g_f}/P_g = unity, (14) reduces to

$$j_d = \frac{k_g \; P_{g_f}}{G/M_m} \left(\frac{\mu}{\rho k_d}\right)^{2/3} = \frac{P_1 - P_2}{\delta P} \left(\frac{A_f}{A_s}\right)\left(\frac{\mu}{\rho k_d}\right)^{2/3} \qquad (15)$$

The similarity between heat transfer and mass transfer is evident when we compare equation (12) with equation (15).

From previous experiments 1/, the correlation of the mass transfer transformation j_d with the heat transfer transformation j_h has been demonstrated. Therefore, it is reasonable to equate j_h and j_d, from equation (8) and equation (15) respectively:

$$\left(\frac{\mu}{\rho k_d}\right)^{2/3}\left(\frac{k_g \; P_{g_f}}{G/M_m}\right) = \frac{h}{c_p \; G} \quad (Pr)^{2/3}$$

Solving for the diffusion coefficient, k_g, we obtain

$$k_g = \frac{h \; (Pr)^{2/3}}{c_p \; P_{g_f} \; M_m \left(\frac{\mu}{\rho k_d}\right)^{2/3}} \qquad (16)$$

With high vapor concentrations, the assumption that $P_{g_f}/P_g = 1$ does not hold. For this condition, incremental values of P_{g_f}/P_g must be considered.

Investigation of the relation for sensible heat transfer and mass transfer indicates that both these rates vary markedly throughout the dehumidifier or condenser. In the case of air-vapor mixtures in air cycle refrigeration systems, initially, when the mixture is unsaturated, low film coefficients can be expected. As the mixture cools, and the saturation point is passed, film coefficients increase. Should further condensation reduce the amount of vapor flowing, a slight reduction of the film coefficient is possible. Furthermore, a simple relation between the temperature of the condensing vapor and the cooling stream does not exist, due to the differential rate of change in enthalpy, as temperature varies.

The problem reduces itself toward finding the relationship of heat transfer coefficient and temperature difference with heat transfer surface.

1/ Reference 1, 16

In other words, the following equation must be solved:

$$\int dA = \int_0^Q \frac{dq}{d(U\,\Delta t)} \qquad (17)$$

The prediction of the performance of a given heat exchanger, when condensation occurs, involves the solution of a heat balance applied across the condensate film.

Neglecting subcooling, we can write (18)

$$A_h h_h \,(T_g - T_c) + k_g M_v \,\lambda (P_v - P_c) A_h = h_c A_c (T_c - T_w) = U\, A_h \Delta t$$

The solution of (18) can be made by a step by step, cut and try process. The performance of the heat exchanger must first be estimated. This gives initial and final temperatures of coolant and cabin air flows. For a given interval of T_g and T_w (assumed constant throughout each interval), the condensate temperature T_c is selected such that (18) is satisfied.

The hot air heat transfer surface required to transfer the heat for each interval, n, is found from (17).

$$(A)_n = \frac{q_n}{(U\,\Delta t)_n} \qquad (19)$$

After completing calculations for all the intervals, the summation of $(A)_n$ gives the heat transfer surface required for the performance initially estimated. This process is repeated with more accurate estimations of performance, until the surface area required equals the surface area available.

Where the entering mixture is not saturated, the determination of when condensate is first formed is a problem in itself.

When considering the diffusional process, condensate will be formed when the cooling surface temperature falls below the mixture dew point, even though the mixture as a whole may be unsaturated. This necessarily complicates an already complex cut-and-try method, involving approximation of the condensation rates when the main body of the mixture is as yet unsaturated. To avoid this condition, condensation must be assumed to occur only when the main mixture body temperature falls below the dew point.

If an effective overall coefficient of heat transfer, U_{eff}, for the heat exchanger is desired, it may be found as follows:

$(q/\Delta t)$ is calculated for each interval.

$$U_{eff} = \frac{\Sigma\, q/\Delta t}{\Sigma\, A} \qquad (20)$$

When evaluating the performance of dry air to air heat exchangers (no condensation), it has been convenient to express performance in terms of effectiveness, η_x, defined on page 10. In general, η_x, for a given dry air to air heat exchanger is independent of inital quality and temperature level and is only a function of flow rates.

However, in the case of wet air to air heat exchangers (condensation), the situation is complicated in that a further controlling factor of performance is condensation rate. Therefore performance of a wet air to air heat exchanger is dependent upon initial quality, pressure and temperature only where such properties affect the condensation rate.

If it is possible to fix these properties, then a good idea of the performance capablilities of a given wet air to air heat exchanger is represented by a temperature difference parameter defined previously as

$$\eta_{wet} = \frac{t_{hot\ in} - t_{hot\ out}}{t_{hot\ in} - t_{cold\ in}} \qquad (21)$$

For the condition of fixed fluid inlet temperature and pressure, the influence of condensation on heat exchanger performance can be clearly shown if the inlet humidity is varied for a large range of flow conditions. Then, based on the method as derived herein, the performance of the heat exchanger throughout the range can be determined.

The analytical study encompassed the following range of flows and humidities, since it is in this range that a large portion of aircraft problems are included:

Cabin flow:	10, 15, 20, 25 lb/min
Flow ratio: (for each cabin flow)	0.5, 0.8, 1.0, 1.5
Inlet humidity:	100, 200, 300, 360 gr/lb dry air

The results of the analytical program are shown plotted in Figures 20 through 23, where regenerator performance is shown as a function of flow ratio for different initial humidities. When compared with the dry effectiveness values as shown in Figure 24, it is apparent that there is a marked reduction of the temperature difference parameter with increasing condensation rate. Other factors, including mass flow and flow ratio, have a similar influence on the temperature difference parameter as on effectiveness, but to a lesser degree.

The reason for the reduction of η_{wet} for a wet heat exchanger is as follows: The heat transferred during condensation is largely latent, which does not serve to reduce the condensing mixture temperature. Therefore η_{wet} being a measure of the temperature change of the condensing mixture, reduces as the latent portion of the cooling load increases. η_{wet} would be much lower if it were not for an actual increase of heat capacity resulting from increasing film coefficients. It is apparent that the extent of such action in improving wet heat exchanger performance is small when compared with the reducing effects caused by condensation as discussed above.

Sample calculations illustrating the basic design technique described herein, for a typical run, (15 lb/min cabin air flow, 1:1 flow ratio, inlet moisture content of 300 gr/lb) are presented in Appendix II.

As previously noted, a search of the available literature on the influence of condensation on heat exchanger performance, revealed no new information considered applicable to the problem other than reviews of the fundamentals of mass and diffusional transfer.

Included in the literature covered was a paper describing a design technique for wet heat exchangers 1/. It appears appropriate at this time to discuss this method so as to indicate the basic difference with the proposed design technique of this report.

The method, as described in the reference, proposed various simplifying assumptions to avoid the complexity of diffusional transfer.

(1) The heat transferred, q, is computed on a step by step basis throughout the exchanger. A temperature vs enthalpy plot, of both the cooling and cooled flows, is constructed, illustrating a heat balance based on counterflow.

(2) On the basis of statistical compilations of typical cross flow heat exchangers, the average values for U_o, the overall dry film coefficient is found and is arbitrarily set as constant throughout the exchanger. The construction of curves illustrating this is such that U_o is based on the log mean temperature difference for counterflow exchangers.

(3) Graphical integration of the temperature-enthalpy plot follows. This will give the required surface area for heat transfer equal to:

$$\int dA = \int \frac{w}{U} \frac{dH}{dt}$$

1/ Reference 7

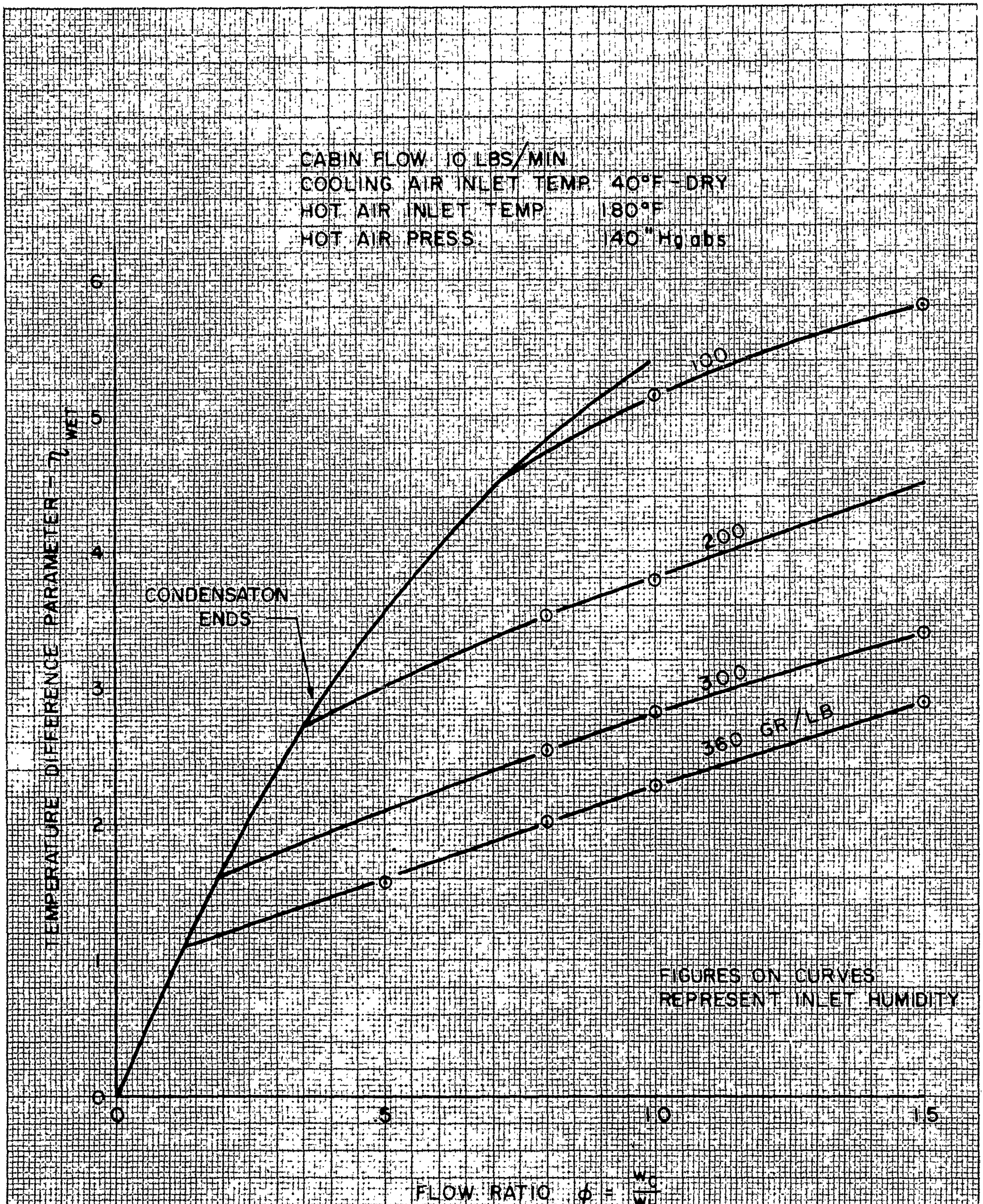


Figure 20. Analytical Results, Wet Heat Exchanger Performance as a Function of Flow Ratio

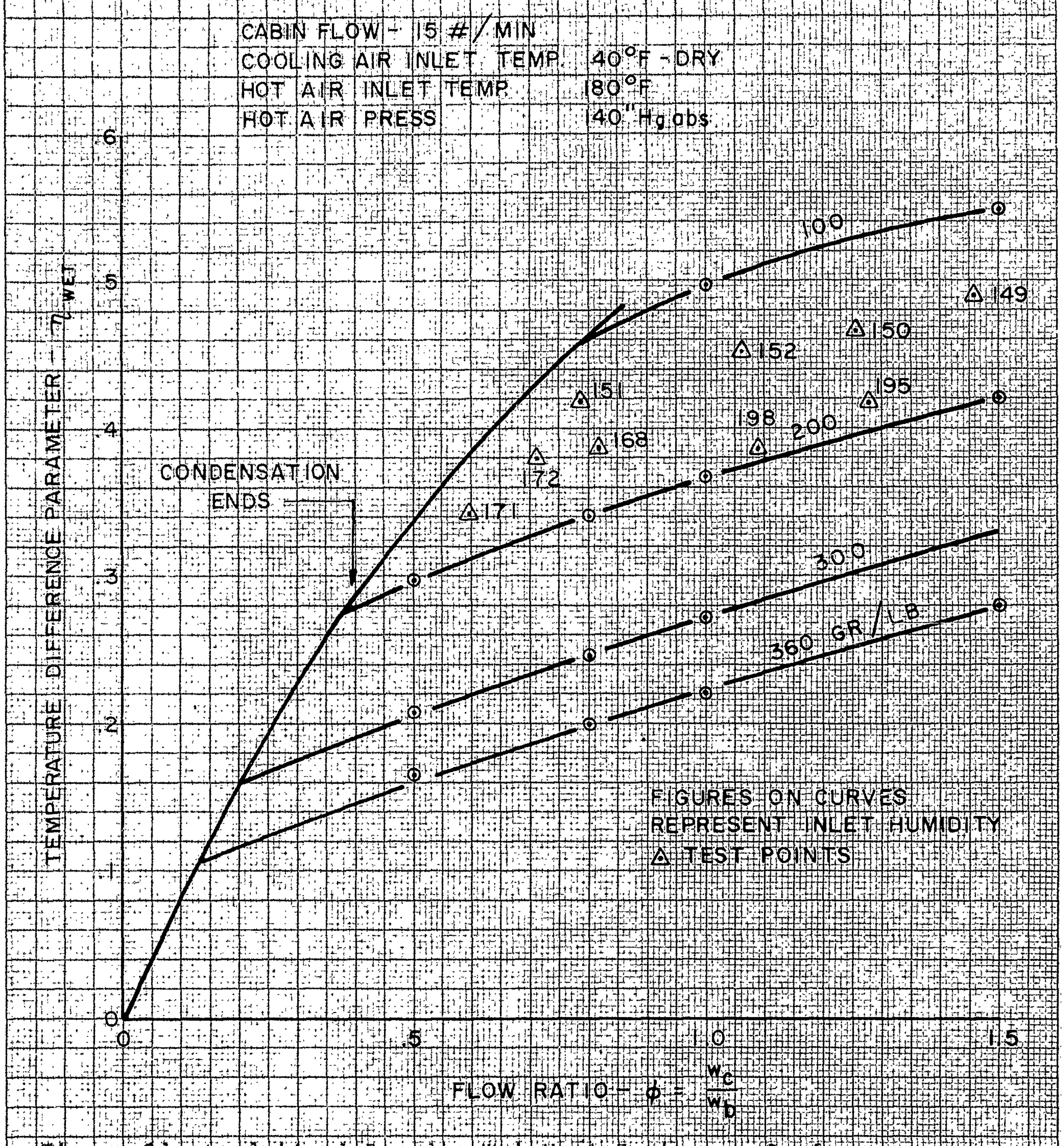


Figure 21. Analytical Results, Wet Heat Exchanger Performance as a Function of Flow Ratio

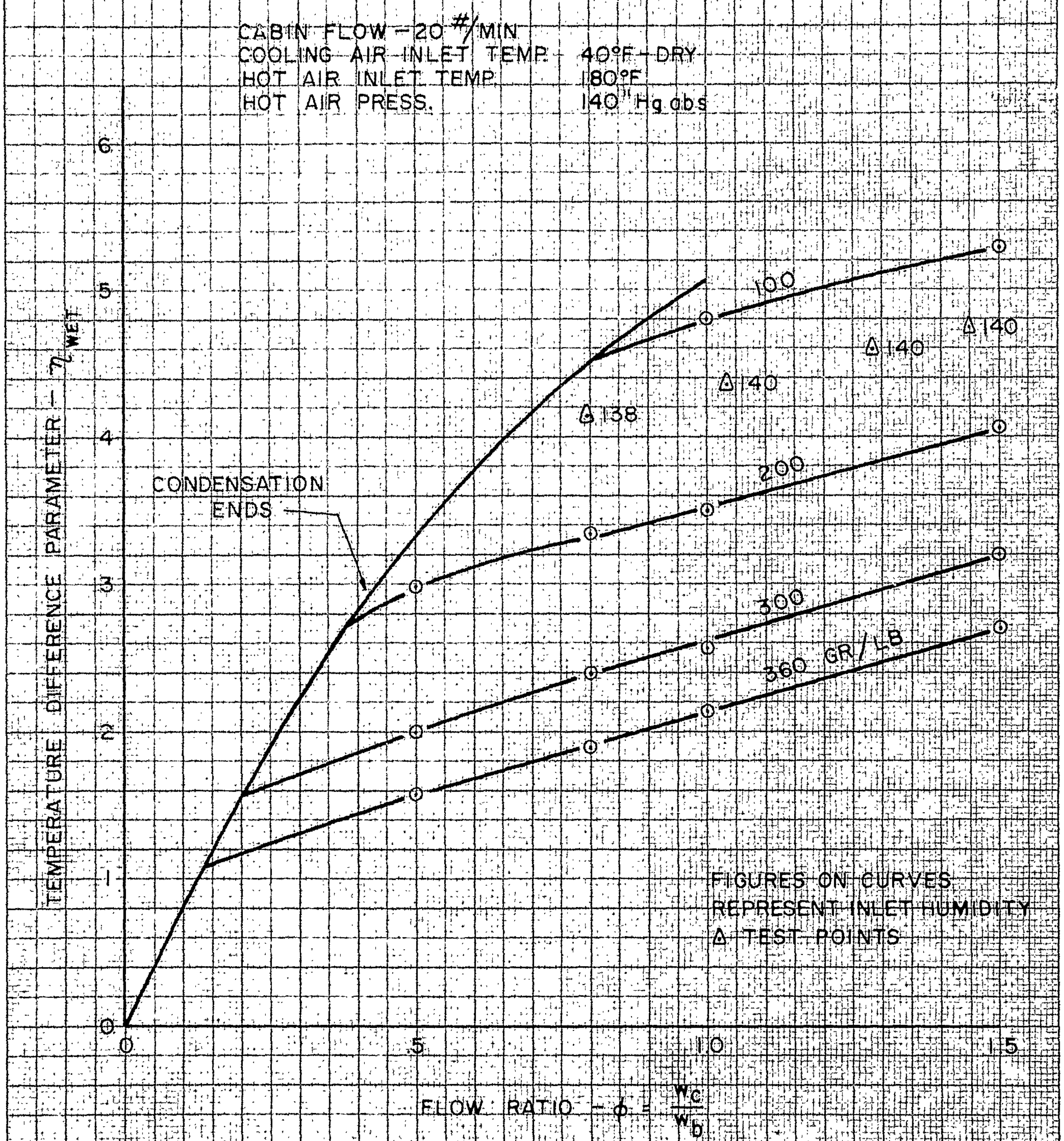


Figure 22. Analytical Results, Wet Heat Exchanger Performance of a Function of Flow Ratio

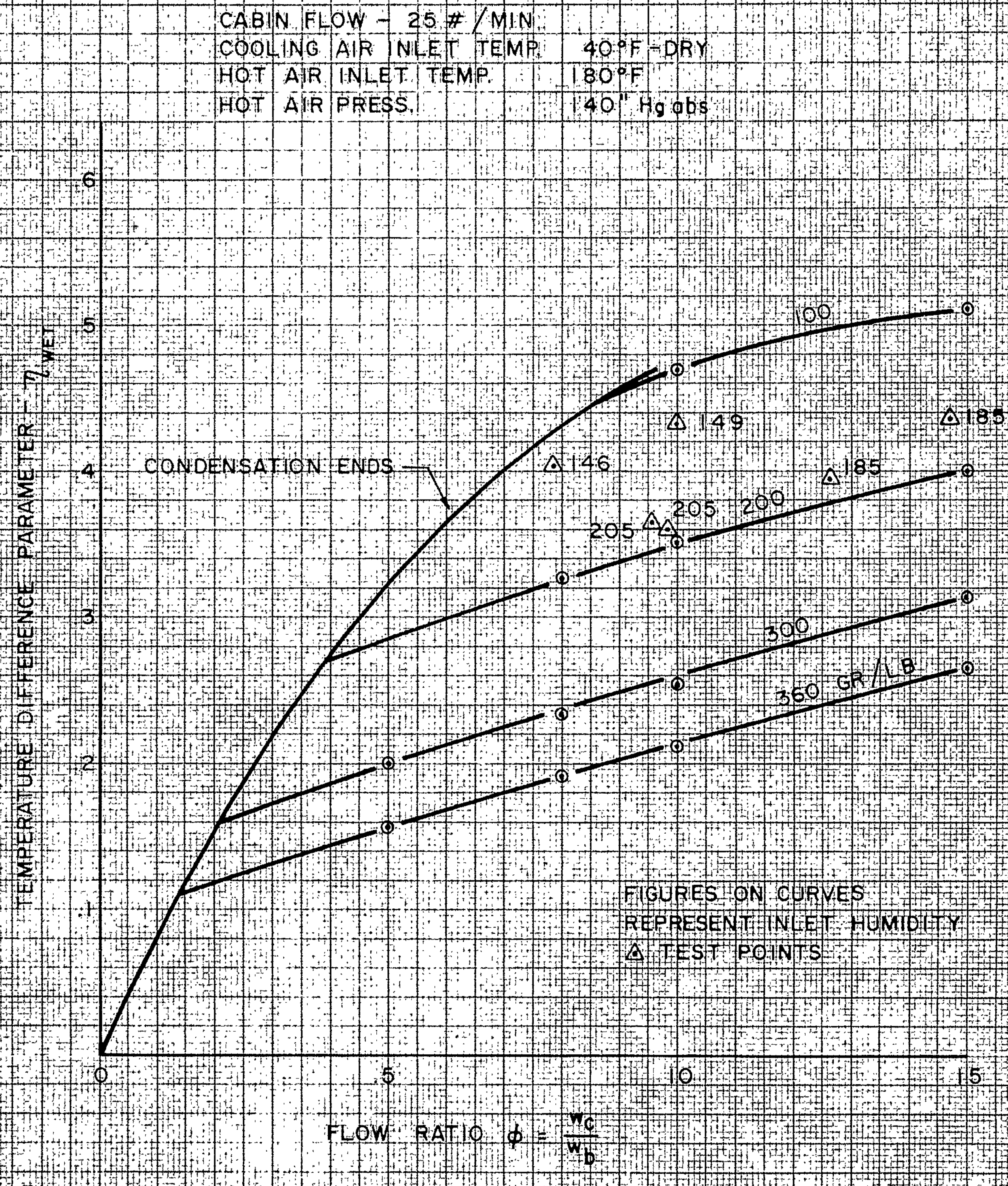


Figure 23. Analytical Results, Wet Heat Exchanger Performance as a function of Flow Ratio

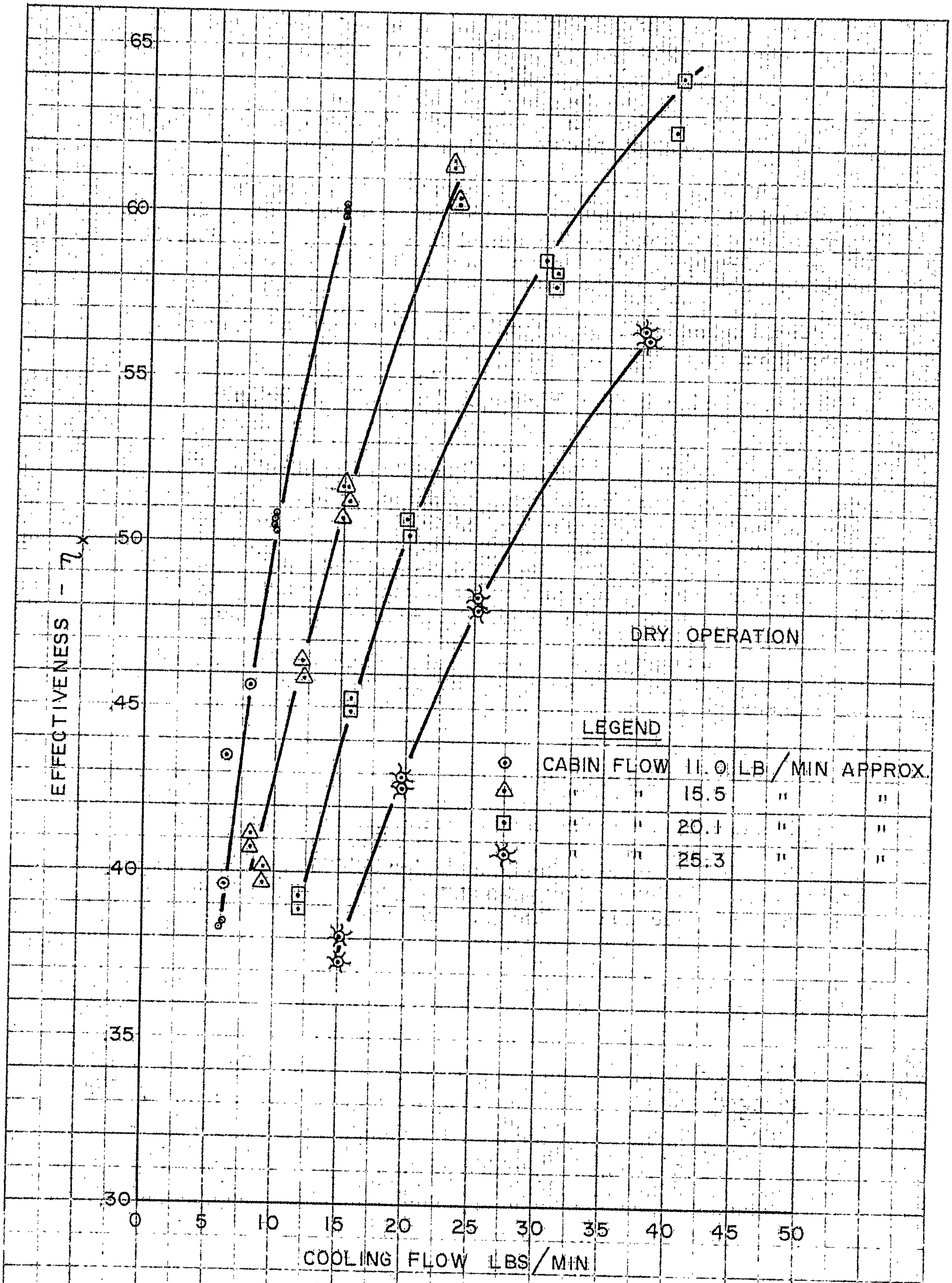


Figure 24. Heat Exchanger Calibration

For purposes of comparison, an outline of the method used in this report, illustrating the salient features, follows:

(1) The heat transferred, q, for cooling the gas and vapor, condensing the vapor and cooling the condensate is computed on a step by step basis throughout the exchanger.

(2) For each step, a heat balance applied across the condensate surface serves to define the heat flux, $U\Delta t$. This involves equating the heat transferred from the gaseous mixture to the condensate surface to the heat transferred from the condensate surface to the cooling medium.

When expressed in equation form, the heat balance is expressed as:

$$h_h(T_g-T_c)+k_gM_v\ (P_v-P_c)=h_c\left(\frac{A_c}{A_h}\right)(T_c-T_w)=U\Delta t$$

For each step, the condensate surface temperature, T_c, can be found to balance the above equation. When the entering gaseous body is unsaturated and the portion of the heat exchanger operates non-condensing, this equation is simplified by the use of the gas film coefficient alone. The determination of when condensation begins has been arbitrarily set as when T_g, the main gas body temperature, reaches the dew point.

(3) The surface area required can now be determined by either graphical or numerical integration of the basic equation for the heat flow.

$$\int dA = \int \frac{dQ}{d(U\Delta t)}$$

It is clear that the first method discussed simplified the design calculations by ignoring diffusional transfer and its influence on U. The use of the counterflow LMTD 1/ rather than the crossflow in determining the value for U, the dry film coefficient, was intended to offset any errors introduced by

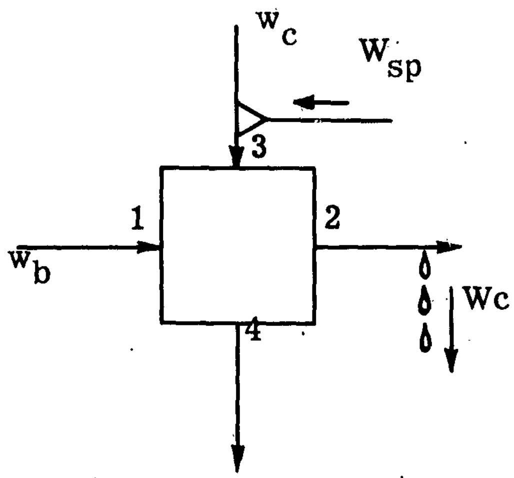
1/ Log Mean Temperature Difference

(1) Neglecting diffusional transfer (U is constant and equal to dry value).

(2) Temperature enthalpy plot based on counterflow.

MODIFIED WET HEAT EXCHANGER DESIGN

The following heat exchanger is to be considered, and a method is derived to permit calculation of its performance.



H_i = inlet humidity of cabin air, gr/lb

W_c = condensed moisture, gr/lb

W_{sp} = sprayed moisture, gr/lb

w_b = cabin flow rate, lbs/min

w_c = cooling flow rate. lbs/min

ϕ = flow ratio = $\dfrac{w_c}{w_b}$

λ = enthalpy of vaporization, BTU/lb

Total heat removed from cabin air is:

$$Q = w_b\, c_p\, (t_1 - t_2) + \frac{w_b W_c \lambda}{7000} + \frac{c_{p_v}(t_1 - t_2) H_i w_b}{7000} \qquad (1)$$

$$Q = w_b\, c_p \left[t_1 - t_2 + \frac{W_c\, \lambda}{7000\, c_p} \right] + \frac{c_{p_v}\, w_b\, H_i\, (t_1 - t_2)}{7000} \text{ BTU/min} \qquad (1a)$$

Under ideal spray conditions, where 100% of the spray is re-evaporated the heat absorbed by the cooling air is given as:

$$Q = w_c\, c_p\, (t_4 - t_3) + \frac{W_{sp}\, w_c \left[\lambda + t_3 - t_2 \right]}{7000} \text{ BTU/min} \qquad (2)$$

To simplify the above equation, the following assumptions can be made with negligible error:

In the range of temperature encountered

λ varies from 990 - 1030 BTU/lb. Use λ = 1010 BTU/lb

$$\frac{\lambda}{7000\, c_p} = \frac{1010}{7000\,(.241)} = .60$$

and

$$\frac{60\,(c_{p_v})}{7000} = \frac{60\,(.46)}{7000} = \frac{1}{253}$$

Rewriting equation (1a) in terms of BTU per hr, we obtain

$$Q = 14.52\, w_b\, (t_1 - t_2 + .60 W_c) + \frac{w_b\, H_i\, (t_1 - t_2)}{253} \qquad (3)$$

To account for incomplete re-evaporation of the sprayed moisture (representing actual conditions), a spray efficiency term, η_{sp}, has been adopted.

This term has been arbitrarily defined as that percentage of injected moisture that is re-evaporated in the cooling air stream. Under actual conditions then equation (2) is written as:

$$Q = w_c\, c_p\, (t_4 - t_3) + \frac{W_{sp}\, w_c}{7000}\, (t_3 - t_2) + \frac{W_{sp}\, \eta_{sp}\, w_c\, \lambda}{7000} \quad \text{BTU/min} \qquad (4)$$

Normally the liquid enthalpy term is negligible. For this condition equation (4) reduces to the following in terms of BTU/hr:

$$Q = 14.52 w_c\, (t_4 - t_3) + (W_{sp}\, \eta_{sp}\, w_c)\ (8.66) \qquad \text{BTU/hr.} \qquad (5)$$

where

$$\frac{60\,\lambda}{7000} \approx \frac{60\,(1010)}{7000} = 8.66$$

Spray efficiency is reduced due to ineffective spray action, poor positioning of nozzles, insufficient time for re-evaporation and abnormal duct wetting. There is also a definite relationship between spray efficiency and the vapor absorbing qualities of air which include the dry bulb temperature, relative humidity, and the amount of vapor injected. Since there was no additional investigation to discover the effects of the above variables upon spray efficiency, it is not clear how much each of these variables contribute toward the overall spray efficiency term. Some indication of its magnitude has been found from the test results and is listed in Table 7 (page 115).

Based on the limited data from this source an average spray efficiency of 55% and 33% was found for spray rates of 100 and 200 gr/lb dry cooling air respectively. These results are for 10 to 15°F dew points, and a dry bulb temperature of 40°F. It is obvious that increasing the dry bulb temperature will increase the spray efficiency.

This point was borne out when the problem of setting cabin initial humidity for the test program was encountered. It was discovered that increasing the dry bulb temperature permitted more moisture to be injected into the air stream before "moisture settling out" began. "Moisture settling out" is defined as the precipitation of sprayed moisture initially before it has time to evaporate in the air stream. As the dry bulb temperature increases more moisture may be injected before settling out begins for a given initial dew point.

A method of calculation has been developed to give a good approximation of heat exchanger performance under the following conditions:

(1) Dry and condensing on cabin side, dry on cooling side.

(2) Dry and condensing on cabin side, water spray on cooling side.

(3) Condensing on cabin side, dry on cooling side.

(4) Condensing on cabin side, water spray on cooling side.

This method is a simplifying modification of the basic "wet heat exchanger design" as described previously, considered applicable for heat exchangers of similar type and construction to the test heat exchanger. Needless to say this technique, confirmed by test result, relieves the designer of tedious cut and dry methods.

There are curves available which give the pure film coefficient, h, as a function of the mass flow rate, for the fin configuration of the test heat exchanger Figure 18. The equation for this curve comes from a modified Nusselt equation, which depicts the convective coefficient of heat transfer. For extended surfaces an "effective" film coefficient must be employed due to the temperature gradient existing along the fins. This thermal gradient serves to reduce the temperature effectiveness of the finned surface. A good indication of the fin efficiency (the proportion of finned surface area available for heat transfer) is given as 1/

$$\eta_f = \frac{\tanh 1 \sqrt{\frac{2h}{k\delta}}}{1 \sqrt{\frac{2h}{k\delta}}} \tag{6}$$

1/ Reference No. 12, page 17

When referring to the total surface available for heat transfer a weighted value for fin efficiency must be used. Since

$$A\eta_o - A_f\eta_f = A - A_f$$

Solving for η_o, the weighted fin efficiency term

$$\eta_o = 1 - A_f/A\ (1 - \eta_f) \qquad (7)$$

For the test heat exchanger, η_o is plotted as a function of the pure film coefficient in Figure 25.

The overall coefficient of heat transfer, as related to the individual coefficients, is given by:

$$U_h = \frac{1}{\dfrac{1}{\eta_{o_h} h_h} + \dfrac{t}{\left(\dfrac{A_c}{A_h}\right) k} + \dfrac{1}{\left(\dfrac{A_c}{A_h}\right) \eta_{o_c} h_c}} \text{ BTU/hr sq ft}^{o}\text{F} \qquad (8)$$

where U_h is based on the hot side total heat transfer area. The subscripts "h", "c" refer to hot and cold side respectively.

For materials of high thermal conductivity, the component of tube wall resistance in equation (8) may be neglected relative to the fluid side resistances, and equation (8), therefore, reduces to the following:

$$U_h = \frac{1}{\dfrac{1}{h_{e_h}} + \dfrac{1}{\left(\dfrac{A_c}{A_h}\right) h_{e_c}}} \qquad (9)$$

where $h_e = \eta_o h$ (the effective film coefficient)

The effective film coefficient, h_{e_h} for the condensing side of the heat exchanger, was determined from the analytical solutions given in this report and the use of the equation (9) in the following manner:

From the analytical solution, the overall heat transfer coefficient was first calculated. Then the value for the effective dry film coefficient on the cooling air side is found from Figure 19 (page 60). Substitution into equation (9) gives h_{e_h}, the effective condensing film coefficient. The results of these calculations are given in Figure 27 (page 105).

These curves give some idea of the increase of effective film coefficient due to the action of vapor condensation. The overall effect on heat transfer is small, since the overall coefficient of heat transmission is composed of the sum of the reciprocals of the film coefficients. Should one of the film coefficients (the cooling air side) be small in comparison with the other, the reciprocal of a small number is little affected when added to the reciprocal of a large number. While the effect is minor, it still exists and should be considered for greater accuracy. Based on the foregoing considerations, the design of a heat exchanger requires the calculation of $UA_{req'd}$ (BTU/hr ^{o}F required to transmit the heating load) and UA_{avail} (the actual BTU/hr ^{o}F that can be transferred through the heat exchanger surface). When UA_{avail} equals $UA_{req'd}$ the heat exchanger performance has been correctly appraised.

The equation used in determining the $UA_{req'd}$ is:

$$UA_{req'd} = \frac{\text{Heating Load}}{\text{Mean Temperature Difference}} \qquad (10)$$

When cross flow exchangers are employed, cross flow factors are available which relate terminal temperatures to mean temperature differences (see Figure 26 page 102). These factors however have been derived and based on the following postulates, which include in part:

(1) Constant heat capacities of the fluids.

(2) No change in phase during any portion of the fluid's path through the heat exchanger.

Although the regenerative type heat exchanger operates under conditions where the foregoing assumptions are not valid, a comparison with the weighted Δt (calculated) indicates that the use of the cross flow temperature difference introduces negligible error in the design of a "wet" cross flow heat exchanger.

The comparison is presented in the following table:

Cabin Flow lb/min	Flow Rates	Initial Humidity gr/lb	Weighted Δt ^oF	Crossflow Δt ^oF	Log Mean Temp Difference ^oF
15	1.0	360	60.45	60.9	66.5
15	1.0	300	59.10	60.9	66.5
15	1.0	200	59.20	60.9	66.5
15	1.0	100	60.70	56.1	65.7
15	0.8	360	56.19	56.0	62.0
15	0.8	300	57.48	57.6	63.5
15	0.8	200	57.85	55.4	61.8
15	0.8	100	58.86	52.9	60.2
15	0.5	360	51.98	51.2	56.8
15	0.5	300	54.30	54.0	55.9
15	0.5	200	55.40	52.0	58.6

Under conditions where the heat exchanger operates with initially non-saturated air and with condensed moisture at outlet, the heat exchanger should be divided into two portions. The first portion is essentially a "dry" operation, beginning from the inlet dry bulb temperature and ending at the dew point temperature. The second portion is a completely "wet" operation, commencing with the dew point and ending at the discharge temperature. For the first portion the dry film coefficient is used in conjunction with the cross flow temperature difference. For the latter portion, the condensing film coefficient is employed together with the cross flow temperature difference.

Before determining the UA_{avail} for a given heat exchanger operating partially condensing, there is a question as to how much of the total hot air surface is operating dry and wet. A similar question exists on the correct division of the cold air surface in absorbing the heat.

A good approximation of this division, which was substantiated by test results, is found from the following assumptions:

For the cooling air surface

$$\zeta_c = \frac{A_{dry_c}}{A_{wet_c}} \approx \frac{Q_{dry}\left(\Delta t_{m_{wet}}\right)}{Q_{wet}\left(\Delta t_{m_{dry}}\right)} \qquad (11)$$

For the hot air surface

$$\xi_h = \frac{A_{dry_h}}{A_{wet_h}} \approx \frac{Q_{dry}}{Q_{wet}} \frac{\Delta t_{m_{wet}}}{\Delta t_{m_{dry}}} \frac{U_{wet_h}}{U_{dry_h}} = \xi_c \frac{U_{wet_h}}{U_{dry_h}} \tag{12}$$

where

$$U_{wet_h} = \frac{1}{\frac{1}{h_{h\ cond}} + \frac{1}{h_c \left(\frac{A_c}{A_h}\right)}} \qquad \text{BTU/hr sq ft } {}^{o}\text{F}$$

and

$$U_{dry_h} = \frac{1}{\frac{1}{h_h} + \frac{1}{h_c \left(\frac{A_c}{A_h}\right)}} \qquad \text{BTU/hr sq ft } {}^{o}\text{F}$$

Therefore, if A_h and A_c represent the total hot air and cooling air core surface, then the UA_{avail} is:

$$(UA)_{wet} = \frac{1}{\frac{1}{h_{h\ cond} \frac{A_h}{(1+\ h)}} + \frac{1}{h_c \frac{A_c}{(1+\xi_c)}}} \tag{13}$$

$$(UA)_{dry} = \frac{1}{\frac{1}{\frac{h_h A_h \xi_h}{(1+\xi_h)}} + \frac{1}{\frac{h_c A_c \xi_c}{(1+\xi_c)}}} \tag{14}$$

and the sum of equation (13) and (14) represent the total UA_{avail}

$$\Sigma\, UA = (UA)_{wet} + (UA)_{dry} \tag{15}$$

Equations (11) through (15) can be used to approximate UA_{avail} when operating both dry and condensing on the hot air side.

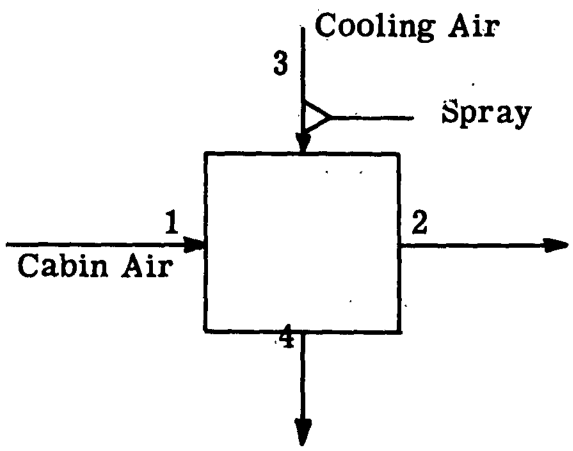
Under conditions where the hot air is initially saturated at inlet, equations (11) through (15) can be used with the following simplifications:

$$\xi_c = \xi_h = 0$$

$$(UA)_{dry} = 0$$

$$\Sigma\, UA = (UA)_{wet}$$

It was felt that the effective film coefficient can be increased by water spray into the cooling air stream. Results from the tests, while not conclusive, indicate that unlike the case of condensation, the influence of such water spray action upon film coefficient is relatively small. The reason for this is not perfectly clear. It may be due to ineffective spray action and that most of the vapor evaporation occurs well into the heart of the heat exchanger. Therefore, for the cooling air side, the effective film coefficient is practically independent of water spray. Results of the test are shown in Figure 28, (page 106) where the effect of water spray on the film coefficient is plotted as a function of the unit weight flow. The error involved in using these curves for spray rates slightly greater than the range specified is considered small.



A heat exchanger with moisture being sprayed into the cooling air inlet stream, is illustrated above. The moisture spray serves to reduce the size of a heat exchanger for a given heat capacity. The reason for this is as follows:

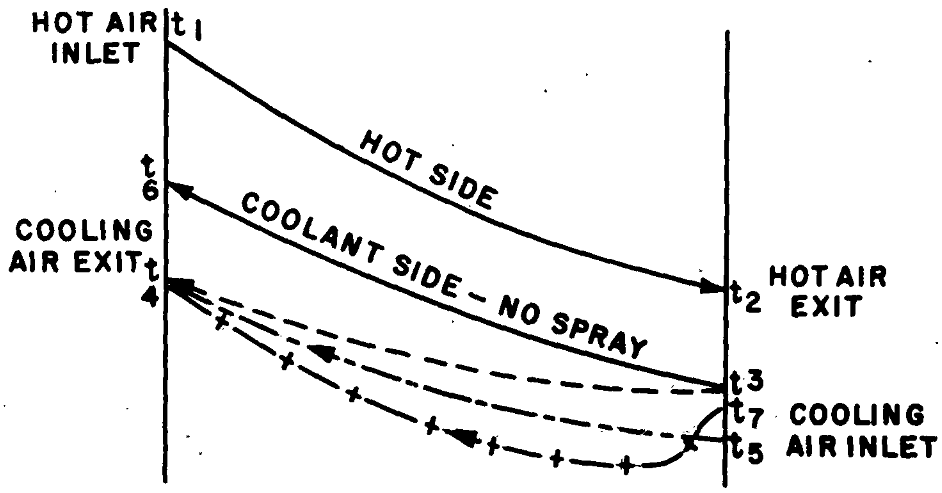
(1) The spraying action increases the film coefficient on the cooling air side, thus reducing the heat transfer surface required for a given heat capacity.

(2) The effective mean temperature difference, Δt_m, across the heat exchanger is increased. Recalling that the $UA_{req'd}$ is equal to $Q/\Delta t_m$, an increase in Δt_m reduces the $UA_{req'd}$, and the heat exchanger size. The increase of Δt_m is due primarily to evaporative cooling by the water spray and the large quantity of latent heat absorbed in this way. The sensible heat the cooling air must absorb is therefore substantially reduced, lowering the cooling air temperatures and increasing Δt_m.

It must be noted that unless adequate design precautions are undertaken to insure sufficient area and time for evaporation, a small portion, if any, of the injected moisture will have been evaporated before the wet cooling air is actually in the heat exchanger. Subsequent evaporation, however, occurs along the cooling air path in the heat exchanger.

Proper design technique requires due consideration to the foregoing factors in the estimating of heat exchanger performance.

Referring to the figure below, the probable temperature gradient of a cross flow heat exchanger with moisture spray into the cooling air stream is depicted. Counterflow operation is presented for simplicity in representation only.



The use of spray efficiency, as previously defined, accounts for evaporative cooling by reducing the cold side discharge temperature from t_6 to t_4. The coolant air path is then presumed to go from t_3 to t_4. In all probability the coolant air path varies considerably from this pattern since evaporative cooling must begin at least the instant the moisture spray strikes the sandwich core of the heat exchanger, for now time and area conducive for evaporation are available. Taking this fact into consideration the actual path should follow the line t_7 to t_4, or a path similar to it. Since no mathematical means are at hand to compute the cross flow mean temperature difference of this path, a path yielding the same effective Δt_m must be created, that can be easily determined by the standard cross flow equations. Such a path is t_5 to t_4, where t_5 is a reduced cooling air inlet temperature used merely to calculate the effective cross flow mean temperature difference. For heat balances, etc. the actual inlet temperature t_3 is used.

One method for estimating t_5 is to assume that t_5 is the reduced temperature based on the initial evaporation of some fraction of the total moisture evaporated. The fraction, ξ, is easily applied to basic thermodynamic equations to calculate t_5.

Let W_{sp} = injected water - gr/lb.

η_{sp} = spray efficiency defined as the fraction of W_{sp} evaporated throughout the heat exchanger.

ξ = fraction of moisture initially evaporated.

An energy balance applied to the system across the presumed mixing zone is:

$$c_p \; (t_3 - t_5) \approx \frac{W_{sp} \; \eta_{sp}}{7000} \quad (\lambda + t_5 - t_2) \; (\xi) \tag{16}$$

From equation (16), solving for t_5.

$$t_5 \approx \frac{c_p \; t_3 - \dfrac{\xi \eta_{sp} \; W_{sp}}{7000} \quad (\lambda - t_2)}{c_p + \dfrac{\xi \eta_{sp} \; W_{sp}}{7000}} \tag{17}$$

To simplify equation (17) the following assumptions are made with negligible error:

$$c_p + \frac{\mathcal{J}\eta_{sp} W_{sp}}{7000} \approx c_p \quad \text{(within 3\%)}$$

In actual practice the enthalpy of evaporation corresponding to t_5 is of the order of magnitude of 1040 BTU/lb. Similarly, t_2 is approximately $50^{o}F$.

For this case then equation (17) reduces to

$$t_5 \approx t_3 - 0.588 \, (\mathcal{J}) \quad (\eta_{sp}) \quad (W_{sp})$$

The value for $\mathcal{J}$ varies for every exchanger depending in part on the following factors:

(1) Cooling air inlet temperature.
(2) Cooling air inlet duct and spray system design.
(3) The moisture injection rate.
(4) The preinjection cooling air humidty and wet bulb temperature.

Since the magnitude of $\mathcal{J}$ is questionable, it should be based upon results from actual practice. For the bootst rap regenerative system, as proposed in this report, the value of $\mathcal{J} = 0.30$ has been assumed. It is felt that suitable design technique applied to the cooling air inlet ducting design will justify this assumption.

APPENDIX II

THEORETICAL INVESTIGATION OF "WET" HEAT EXCHANGER

This section describes the methods used to calculate "wet" heat exchanger performance. The first method shown refers to a basic "wet" heat exchanger design, used for the analytical study described on page 61; the second method, described as a modified "wet" heat exchanger design, follows. The performance calculated by the use of both methods, applied to a given test heat exchanger 1/, are substantiated by actual tests.

BASIC "WET" HEAT EXCHANGER DESIGN

The heat exchanger performance is to be calculated for the following conditions:

Cabin air flow	15 lbs/min
Cooling air flow	15 lbs/min
Cabin air inlet humidity	300 gr/lb dry air
Cabin air inlet pressure	140 in. Hg abs
Cabin air inlet temp.	180°F
Coolant air inlet temp.	40°F

Mixture dew point:

Substituting into the Carrier equation 2/, we obtain

$$\text{Sat. pressure} = \frac{W_s \; P_t}{4354+W_s} = \frac{(300)\,(140)}{4354+300} = 9.024 \text{ in. Hg abs}$$

Dew point = 157.5°F 3/

$$\text{Cabin air flow} = \frac{(15)\,(60)}{29} = 31 \text{ mol/hr}$$

$$\text{Cabin vapor flow} = \frac{(15)\,(60)\,(300)}{(7000)\,(18)} = 2.14 \text{ mol/hr}$$

At this point cabin air exit temperature must be assumed and a heat balance determined for a number of intervals in the heat exchanger.

1/ See Table 3, page 89, for surface construction of test heat exchanger.
2/ Reference No. 8,
3/ Reference No. 26

As a first approximation assume $T_{g_{exit}} = 142^oF$.

Consider points at 180, 170, 157.5, 150 and 142^oF.
The heat load, q, is computed for each interval.

Interval 180-170^oF

Sensible heat loss:

Vapor = (Mol vap) (Mol wt) (c_p) (Δt) =
(2.14) (18) (0.46) 180-170) = 177.2 BTU/hr

Air = $w_b\ c_p\ t$ = (15) (0.24) (60) (10) = 2178 BTU/hr

Total for interval, q = 2355.2 BTU/hr

Interval 170-157.5^oF

$$q = \frac{2355.2\ (170-157.5)}{(180-170)} = 2945 \text{ BTU/hr}$$

Interval 157.5 - 150^oF (Condensation begins)

at $T_g = 150^oF$ $\quad P_v = 0.2528$ atm
$\quad P_g = 4.6-0.2528 = 4.427$ atm

$$\text{mols vapor sat.} = 31\left(\frac{.2528}{4.427}\right) = 1.773 \text{ mols}$$

mols condensed = 2.14 - 1.773 = 0.367 mol/hr

Sensible heat loss: Vapor = (2.14) (18) (0.46) (7.5) = 132.9BTU/hr

Air = (15) (60) (0.24) (7.5) = 1634 BTU/hr

Latent heat loss = (Mol wt)(Condensed mols)(h_{fg})

= (18)(0.367)(1008.2) = 6660 BTU/hr

Total for interval, q = 8427 BTU/hr

Interval 150-142^oF

at $T_g = 142^oF$ $\quad P_v = 0.2067$ atm
$\quad P_g = 4.473$ atm

$$\text{Mols condensed} = \underline{1.773-(31)}\ \left(\frac{0.2067}{4.473}\right) = 0.338 \text{ mol/hr}$$

Sensible heat loss: Vapor = (2.14)(18)(0.46)(8) = 141.8 BTU/hr

Air = (15)(60)(0.24)(8) = 1742 BTU/hr

Latent heat loss = (18)(0.338)(1012.9) = 6162 BTU/hr

q = 8046 BTU/hr

The total heat transferred for all intervals is:

$$Q = \sum_{0}^{n} q = 21,773 \text{ BTU/hr}$$

$$\text{Coolant air temperature rise} = \frac{Q}{w_c c_p} = \frac{21,773}{(15)(60)(0.24)} = 100^{o}F$$

Coolant air outlet temperature = 40 + 100 = 140°F

We now proceed to determine $U \Delta t$ from point to point in the unit. Until the dew point is attained, cooling surface temperature is calculated by considering only sensible heat transfer from gas to surface. When the mixture temperature falls below the dew point, the surface temperature of the condensate is evaluated by considering the diffusing rate as well as the sensible heat transfer rate.

Actual cabin unit weight flow =

$$\frac{15\left(1 + \frac{W_s}{7000}\right)}{0.05725} = \frac{15\left(1 + \frac{300}{7000}\right)}{0.05725} = 273.2 \text{ lb/min sq ft}$$

where cabin side free flow = 0.05725 sq ft
(see Table 3, Page 90)

From Figure 18 (Page 59) pure film coefficient is

$$h_h = 42 \text{ BTU/hr sq ft } ^{o}F$$

When using pure film coefficients, a fin efficiency factor η_o must be used to account for temperature gradients existing along extended surfaces 1/. With the aid of the following equations, a curve of η_o as a function of h_h was drawn, and is included as Figure 25 (Page 92).

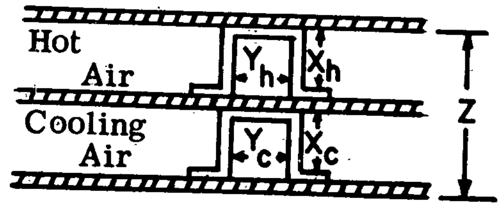
1/ Reference No. 12, Page 17

TABLE 3

SURFACE CONFIGURATION OF TEST HEAT EXCHANGER

NURH 15   Blower Heat Exchanger - Stratos Part No. 16804 - Surface Data

		Hot Air	Coolant Air
1.	Fin type	Herringbone	Herringbone
2.	Fin height (in)	0.375	0.375
3.	Fin spacing(fins per in)	18	15
4.	Fin thickness (in)	0.006	0.006
5.	Part. sheet thickness (in)	0.01	0.01
6.	Effective passage length (in)	7-1/8	4-3/16
7.	Passage width (in)	4-3/16	7-1/8
8.	Number of passages	6	7
8a.	Number of passes	1	1



X_h = 0.369
X_c = 0.369
Y_h = 0.0495
Y_c = 0.0607
Z = 0.77

No Flow Length = 5-1/8 in.

9. Coolant air free Area = $\frac{(X_c)(Y_c) \times 3 \times 7 \times 8}{144} = \frac{(0.369)(0.0607\,(15)(7\text{-}1/8)(7)}{144} = 0.1163$ sq ft

10. Hot air free area = $\frac{(X_h)(Y_h) \times 3 \times 7 \times 8}{144} = \frac{(0.369)(0.0495)(18)(4\text{-}3/16)(6)}{144} = 0.05725$ sq ft

11. Coolant air core surface = $\frac{2(X_c + Y_c) \times 3 \times 6 \times 7 \times 8}{144} = \frac{(2)(0.4297\,(15)(4\text{-}3/16)(7\text{-}1/8)(7)}{144} = 18.68$ sq ft

12. Hot air core surface = $\frac{2(X_h + Y_h) \times 3 \times 6 \times 7 \times 8}{144} = \frac{2(0.4185)(18)(7\text{-}1/8)(4\text{-}3/16)(6)}{144} = 18.68$ sq ft

$$\eta_o = 1 - \frac{A_f}{A}(1-\eta_f)$$

$$\eta_f \approx \frac{\tanh ml}{ml}$$

$$m = \sqrt{\frac{2h_h}{k\delta}}$$

A_s/A = 0.88 (ratio of secondary surface to total surface) (see Table 3).

From Figure 25, η_{o_h} = 0.9114

Similarly, actual coolant unit weight flow = $\frac{15 \times 60}{0.1163}$ = 129 lb/min sq ft

where coolant side free flow area = 0.1163 sq ft (see Table 3).

h_c = 27 BTU/hr sq ft °F

η_{o_c} = 0.9395

If we neglect the thermal resistance of the aluminum wall, then the heat transferred can be expressed as,

$$q = h_h \eta_{o_h} A_h (T_g - t_s) = h_c \eta_{o_c} A_c (t_s - t_w) = U_h A_h (T_g - t_w) \text{ BTU/ hr}$$

and

$$\frac{1}{U_h} = \frac{1}{\eta_{o_h} h_h} + \frac{1}{\eta_{o_c} h_c (A_c/A_h)}$$

Solving for t_s the wall temperature,

$$t_s = \frac{h_c \eta_{o_c} \frac{A_c}{A_h} t_w + h_h \eta_{o_h} T_g}{h_c \eta_{o_c} \frac{A_c}{A_h} + h_h \eta_{o_h}} \quad °F$$

Substituting values for the first point, and noting that

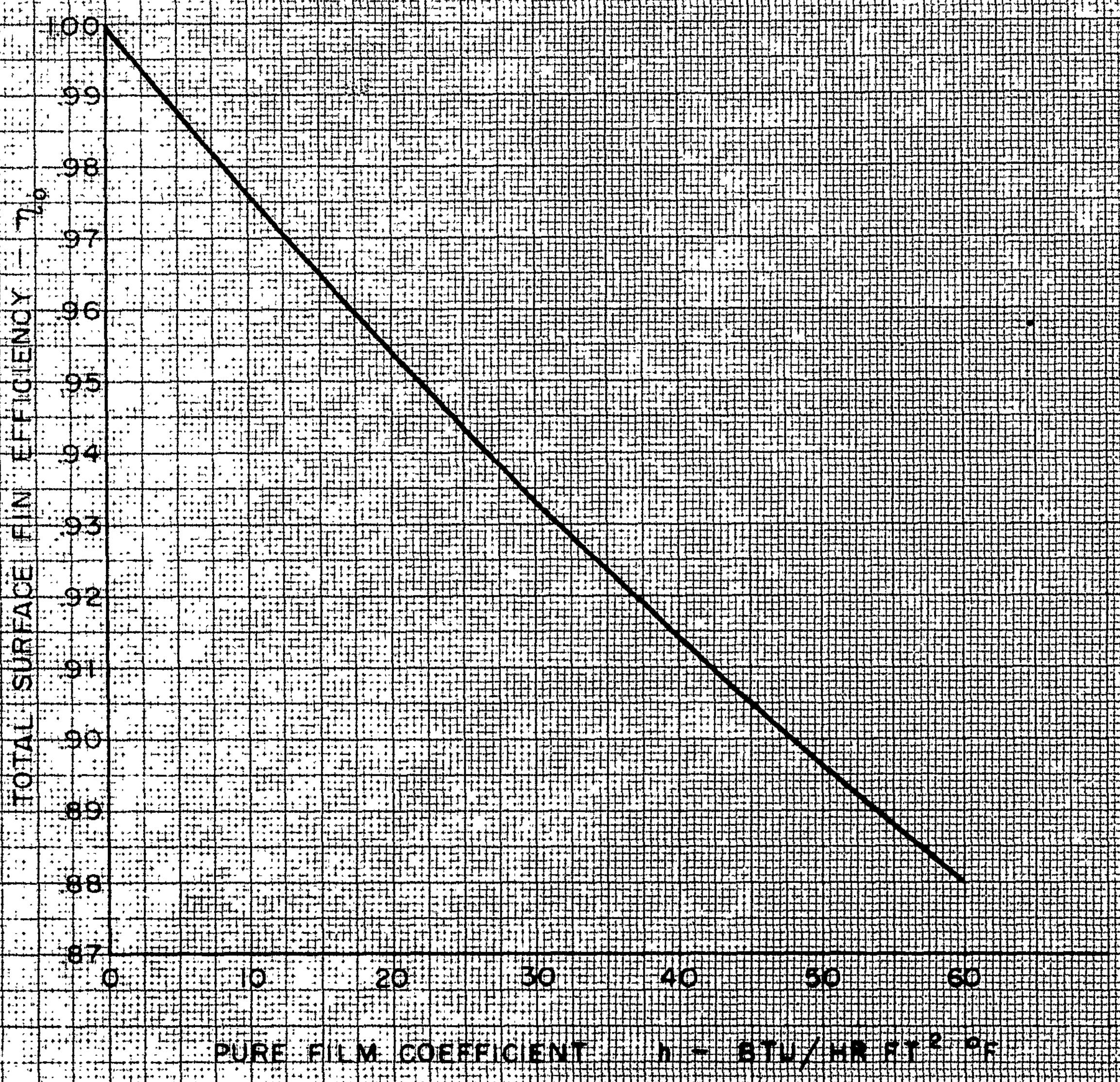


Figure 25. Fin Efficiency of Extended Surfaces

$$\frac{A_c}{A_h} = 1 \text{ (see Table 3).}$$

$$t_s = \frac{(27)(0.9395)(140) + (42)(0.9114)(180)}{(0.9395)(27) + (42)(0.9114)} = 164.1\ ^{o}F$$

U_h = 15.28 BTU/hr sq ft ^{o}F

$U\Delta t$ = (15.28)(180-140) = 611 BTU/hr sq ft

For the second point,

T_g = 170 ^{o}F

$$t_w = 140 - \frac{q}{w_c\ c_p\ 60} = 140 - \frac{2355}{15(0.24)(60)} = 129.2\ ^{o}F$$

$$t_s = \frac{(0.9114)(42)(170) + 0.9395\ (27)(129.2)}{(0.9114)(42) + (0.9395)(27)} = 153.8\ ^{o}F$$

U_h = 15.28 BTU/hr sq ft ^{o}F

$U\Delta t$ = 15.28 (170-129.2) = 622 BTU/hr sq ft

For the third point,

T_g = 157.5^{o}F (dew point)

$$t_w = 129.2 - \frac{2945}{(15)(0.24)(60)} = 115.7\ ^{o}F$$

Mass transfer must now be considered, since the dew point is reached.

P_v = 0.303 atm (see psych. charts corresponding to 157.5^{o}F) 1/

P_g = 4.377 atm

$$\text{Vapor weight flow} = \frac{(31)(0.303)(18)}{4.377} = 38.5 \text{ lb/hr}$$

1/ Reference No. 8

Properties of fluids 1/ :

$$\rho_{air} = \frac{(39.7)(P_g)}{(Tg+460)} = \frac{(39.7)(4.377)}{460+157.5} = 0.281 \text{ lb/ft}^3$$

$\rho_{vapor} = 0.01223 \text{ lb/ft}^3$ 2/

$$\rho_{mix} = \frac{(60)(15)(0.281) + 38.5\,(0.01223)}{(60)(15) + 38.5} = 0.27 \text{ lb/ft}^3$$

$\mu_{air} = 0.0496$ lb/ft hr

$\mu_{vapor} = 0.0299$ lb/ft hr

$$\mu_{mix} = \frac{(900)(.0496) + (38.5)(0.0299)}{938.5} = 0.0488 \text{ lb/ft hr}$$

$k_{air} = 0.01711$ BTU/hr sq ft ^{o}F/ft

$k_{vapor} = 0.01259$ BTU/hr sq ft ^{o}F ft

$$k_{mix} = \frac{(900)(0.01711) + (38.5)(0.01259)}{938.5}$$

$= 0.01692$ BTU/hr sq ft ^{o}F/ft

$c_{p_{air}} = 0.2408$ BTU/lb ^{o}F

$c_{p_{vapor}} = 0.461$ BTU/lb ^{o}F

$$c_{p_{mix}} = \frac{(900)(0.2408) + (38.5)(0.461)}{938.5} = 0.2498 \text{ BTU/lb } ^{o}\text{F}$$

* The diffusivity can be found from the following equation 3/:

$$k_d = \frac{0.0166 \quad T^{3/2}}{P_t\,(v_1^{1/3} + v_2^{1/3})^2} \left[\frac{1}{M1} + \frac{1}{M_2}\right]^{1/2} \qquad \text{sq ft/hr}$$

1/ Reference No. 23
2/ Reference No. 26
3/ Reference No. 5

where

T = temperature ^{o}K $\quad P_t$ = pressure, atm

v = molecular volume $\quad v_{vap} = 14.8 \quad v_{air} = 29.9$

M = molecular weight $\quad M_{vap} = 18 \quad M_{air} = 28.97$

$$k_d = \frac{(0.0166)(1/18 + 1/28.97)^{1/2}}{\left[(14.8)^{1/3} + (29.9)^{1/3}\right]^2} \frac{T^{3/2}}{4.68} = 0.0000344\, T^{3/2} \text{ sq ft/hr}$$

for $T = 157.5^{o}F = 342.7^{o}K$

$$k_d = 0.218 \text{ sq ft/hr}$$

$$Sc = \mu/\rho k_d = \frac{0.0488}{(0.2703)(0.218)} = 0.829$$

$$Pr = \frac{c_p \mu}{k} = \frac{(0.2498)(0.0488)}{0.01692} = 0.7207$$

$$M_m = \frac{900 + 39.5}{31 + 2.14} = 28.28$$

The diffusion coefficient is found as follows:

$$k_g = \frac{h \; (Pr)^{2/3}}{c_p P_{g_f} M_m (Sc)^{2/3}} = \frac{h \; (.7207)^{2/3}}{P_{g_f}(0.24987)(28.28)(0.829)^{2/3}}$$

$$= \frac{0.123\, h}{P_{g_f}}$$

Assume temperature of the condensate film $T_c = 151.7^{o}F$

From psychrometric charts,

$$P_c = 0.2637 \text{ atm}$$

$$P'_g = 4.4163 \text{ atm}$$

$$P_g = 4.377 \text{ atm}$$

Since $P_g \approx P'_g$

$$P_{g_f} = \frac{P_g + P'_g}{2} = 4.396 \text{ atm}$$

From Reference 26 read

$$h_{f_g} = 1003.8 \text{ BTU/lb}$$

The heat balance across the condensate film is given as follows:

$$\eta_{o_h} h_h (T_g - T_c) + k_g M_v \lambda \eta_{o_h} (P_v - P_c) = h_{o_c} \eta_{o_c} (T_c - t_w) = U_h (T_g - t_w)$$

$$= (42)(0.9114)(157.5-151.7) + \frac{0.123(42)}{4.396}(18(1003.8)(0.9114)(0.303-0.2637)$$

$$= 222 + 755 = 27\,(0.9395)(151.7-115.7)$$
$$977 \approx 914$$

Both sides of the equation are in sufficient agreement to avoid recalculation with a new assumption for T_c.

$$U_h \Delta t = \frac{977 + 914}{2} = 945 \text{ BTU/hr sq ft}$$

$$U_h = \frac{U\Delta t}{\Delta t} = \frac{945}{157.5 - 115.7} = 22.6 \text{ BTU/hr sq ft } ^oF$$

The coolant temperature for the next point is found as follows:

$$\Delta t \text{ rise} = \frac{q}{w_c c_p} = \frac{8427}{(15)(60)(0.24)} = 38.6^oF$$

$$t_w = 115.7 - 38.6 = 77.1\ ^oF$$

In a similar manner calculations for intervals, 150 and 142°F, are carried out. The results are tabulated below.

T_g	t_w	T_c	P_{g_f}	$U \Delta t$	U
157.5	115.7	151.7	4.396	945	22.6
150	77.1	138.5	4.459	1603	22.0
142	40	124.0	4.512	2131	20.9

We can now proceed to compute the required heat transfer surface area by the solution of equation (19) (see page 65). An accurate method for solving equation (19) involves plotting $\sum Q$ versus $1/U\Delta t$; the area under the curve corresponds to the required hot surface area, A_h.

An alternative method, a numerical averaging technique, is illustrated in Table 4. Results obtained in this way are considered acceptable for the degree of accuracy desired.

The results from Table 4 indicate a required hot surface area of 18.50 sq ft. Since this value reasonably agrees with the available surface area of 18.68 sq ft (see Table 3), the initial assumption of the cabin air flow exit temperature was correct. Usually three runs are required to bracket the available surface area, after which graphical interpolation yields the correct exit temperature.

Therefore, the temperature difference parameter is

$$\eta_{wet} = \frac{180 - 142}{180 - 40} = 0.2714$$

MODIFIED WET HEAT EXCHANGER DESIGN

The accuracy of the proposed modified design technique is demonstrated by application to actual test runs. (See Appendix III for description of tests).

RUN NO. 147

w_{cabin}	$= 15.18$ lb/min
$w_{cooling}$	$= 19.64$ lb/min
$T_{cabin\ in}$	$= T_1 = 179.75^oF$
$T_{cabin\ out}$	$= T_2 = 110.75^oF$
$t_{cool\ in}$	$= t_1 = 40^oF$
$t_{cool\ out}$	$= t_2 = 103.3^oF$
H_1	$= 196.46$ gr/lb cabin air
W_{sp}	$= 192.5$ gr/lb cool air
P_{cab}	$= 140$ in Hg abs

TABLE 4

SUMMARY OF RESULTS

• BASIC METHOD OF "WET" HEAT EXCHANGER DESIGN

$wb = 15$ lb/min
$\phi = 1:1$
$H_i = 300$ gr/lb

1	2	3	4	5	6	7	8	9	10	11	12	13
P_t	T_g	T_c	$(U\Delta t)$	$(U\Delta t)_{avg}$	U	Q	ΣQ	$A_h = \frac{Q}{(U\Delta t)_{avg}}$	ΣA	Δt	$(\Delta t)_{avg}$	$\frac{Q}{(\Delta t)_{avg}}$
1	180	--	611	--	15.28	--	0	--	0	40	--	--
2	170	--	622	617	15.25	2,355	2,355	3.817	3.817	40.8	40.4	58.3
3	157.5	151.7	945	783	22.61	2,945	5,300	3.761	7.578	41.8	41.3	71.3
4	150	138.5	1,603	1274	22.00	8,427	13,727	6.615	14.193	72.9	57.4	146.8
5	142	124	2,131	1867	20.9	8,046	21,773	4.31	18.50	102	87.5	92.0

$$U = \frac{\Sigma Q/\Delta t_{avg}}{\Sigma A} = 19.91 \qquad \Delta t_{mtd} = \frac{\Sigma Q}{\Sigma Q/\Delta t_{avg}} = 59.1 \qquad \Sigma \frac{Q}{\Delta t_{avg}} = 368.4$$

$$\eta_{wet} = \frac{180-142}{180-40} = 0.2714$$

From psychrometric chart at 140 in Hg abs, 196.46 gr/lb

Dew point = 141.^{o}F

From psychrometric chart at 140 in Hg abs, 110.75^{o}F

Sat. vapor = 84.66 gr/lb

W_c = 196.46 - 84.66 = 111.8 gr/lb condensed

Total heat load:

$$Q_{tot} = 14.52\, w_{cab} (T_1 - T_2 + 0.60\, W_c) + \frac{w_{cab}\, H_i(T_1 - T_2)}{253} \text{ BTU/hr}$$

$$= 14.52(15.18) \left[179.75 - 110.75 + 0.6(111.8)\right] + \frac{15.18(196.46)}{253}\left[179.75 - 110.75\right]$$

= 28,998 BTU/hr

Heat load to dew point:

$$Q_{dry} = w_{cab} \left[14.52 + \frac{H_i}{253} (T_1 - T_{dp})\right]$$

$$= 15.18 \left[14.52 + \frac{196.46}{253} (179.75 - 141.1)\right]$$

= 8975 BTU/hr

Cooling air temperature corresponding to cabin air dew point:

$$t_3 = t_2 - \frac{Q_{dry}}{Q_{total}} \left[t_2 - t_1\right]$$

$$= 103.3 - \frac{8975}{28,998} (103.3 - 40)$$

= 83.71^{o}F

Calculation of mean temperature differences:

Dry section:

$$\eta\, x_{cab} = \frac{T_1 - T_{dp}}{T_1 - t_3} = \frac{179.75 - 141.1}{179.75 - 83.71} = 0.4024$$

$$\eta_{x_{cool}} = \frac{t_2 - t_3}{T_1 - t_3} = \frac{103.3-83.71}{179.75-83.71} = 0.2040$$

From Figure 26:

$$\frac{\Delta t_m}{T_1 - t_3} = 0.674$$

$$\Delta t_m = 0.674(179.75-83.71) = 64.73\ ^{o}F$$

Wet section:

$$\eta_{x_{cab}} = \frac{T_{dp} - T_2}{T_{dp} - t_1} = \frac{141.1-110.75}{141.1 \quad -40} = 0.300$$

$$\eta_{x_{cool}}{}^{1/} = \frac{t_3 - t_1}{T_{dp} - t_1} = \frac{83.71-40}{141.1-40} = 0.4323$$

From Figure 26:

$$\frac{\Delta t_m}{T_{dp} - t_1} = 0.607$$

$$\Delta t_m = 0.607 \; [141.1-40] = 61.37^{o}F$$

$UA_{req'd}$:

$$(UA)_{reqd} = \frac{Q_{dry}}{\Delta t_{mdry}} + \frac{Q_{total} - Q_{dry}}{\Delta t_{m_{wet}}}$$

$$= \frac{8975}{64.73} + \frac{28,998-8975}{61.37} = 465\ BTU/hr\ ^{o}F$$

1/ No reduced cooling air inlet temperature using "$\mathcal{T}$" is considered for this case due to the very low cooling air inlet temperature. It cannot be expected that the sprayed moisture will have a chance to evaporate at this low temperature until well into the heart of the heat exchanger. For this condition then the temperature gradient need not be modified.

Determination of UA_{avail}:

Surface area available (see Table 3)

Cold side = A_c = 18.68 sq ft

Cabin side = A_h = 18.68 sq ft

Film coefficients

Cabin side: $\frac{w}{A_h} = \frac{15.18}{0.05725} = 265.2$ lb/min sq ft

From Figure 27: h_h = 38 BTU/hr sq ft ^oF (Dry)

h_{hcond} = 160 BTU/hr sq ft ^oF (111.8 gr/lb)

Cold side: $\frac{w}{A_c} = \frac{19.64}{.1163} = 168.9$ lb/min sq ft

From Figure 28, for $\frac{w}{A_c}$ = 168.9, W_{sp} = 192.5 gr/lb

h_c = 34 BTU/hr ft^oF

Overall film coefficients:

$$U_{dry} = \frac{1}{\frac{1}{h_h} + \frac{1}{h_c}} = \frac{1}{\frac{1}{38} + \frac{1}{34}} = 17.95 \text{ BTU/hr sq ft } ^oF$$

$$U_{dry} = \frac{1}{\frac{1}{h_{h\,cond}} + \frac{1}{h_c}} = \frac{1}{\frac{1}{160} + \frac{1}{34}} = 28.04 \text{ BTU/hr sq ft} ^oF$$

Area Ratio:

$$\xi \text{ cool side} = \frac{A_{dry}}{A_{wet}} = \frac{Q_{dry}}{\Delta t_{m dry}} \times \frac{\Delta t_{m\ wet}}{(Q_{total} - Q_{dry})}$$

$$= \frac{8975}{64.73} \times \frac{61,37}{(28,998-8,975)}$$

$$= 0.4249$$

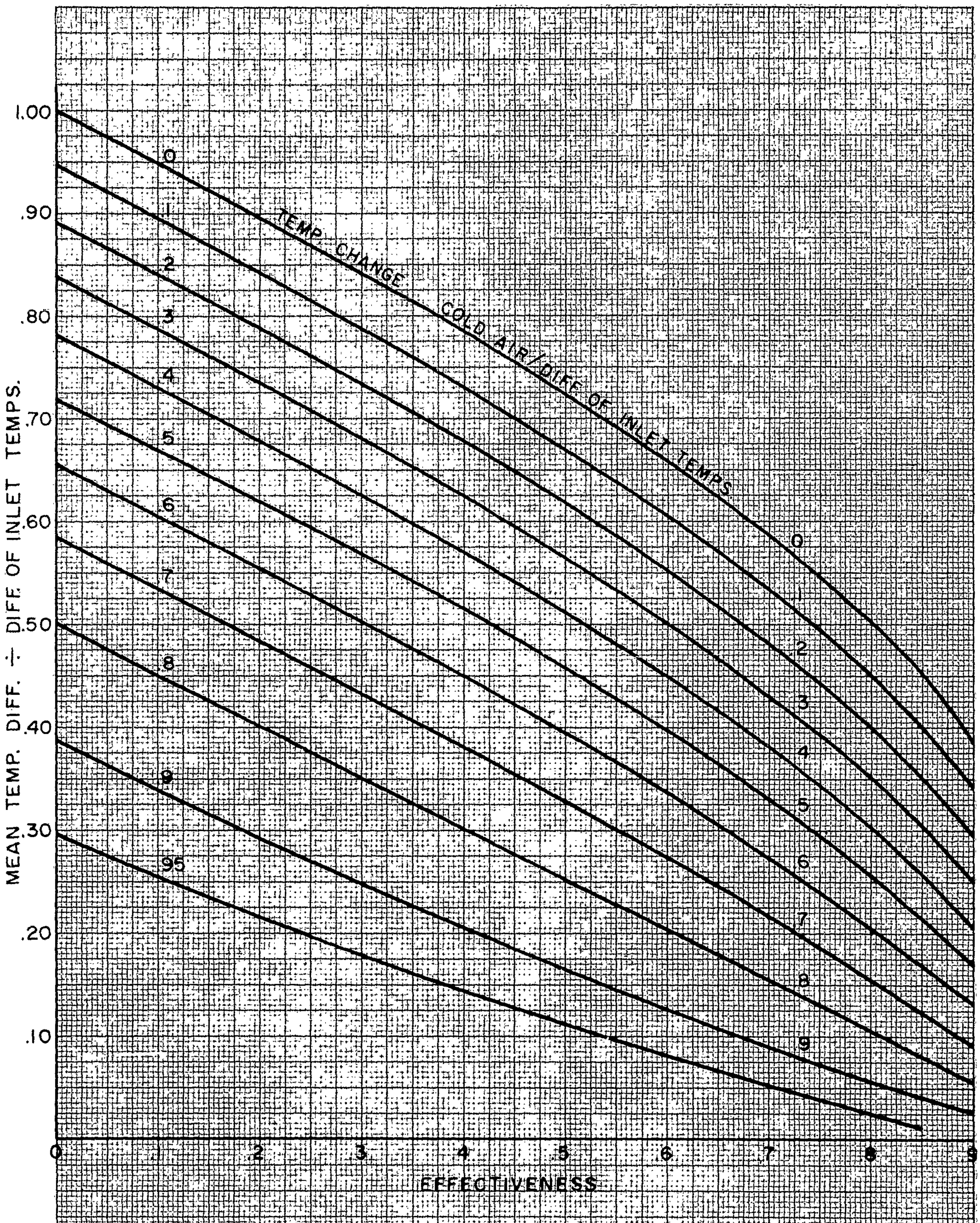


Figure 26. Cross Flow Mean Temperature Difference

$$\xi_{\text{hot side}} = \xi_c \left(\frac{U_{wet}}{U_{dry}}\right) = \frac{0.4249 \times 28.04}{17.95} = 0.6637$$

$$(UA)_{wet} = \cfrac{1}{\cfrac{\cfrac{1}{(A_{tot_h})(h_{h_{wet}})}}{1+\xi_h} + \cfrac{\cfrac{1}{(A_{tot_c})\, h_c}}{1+\xi_c}} = \cfrac{1}{\cfrac{\cfrac{1}{160 \times 18.68}}{1+0.6637} + \cfrac{\cfrac{1}{34 \times 18.68}}{1+0.4249}}$$

$$= 357.1 \text{ BTU/hr }^{\circ}\text{F}$$

$$(UA)_{dry} = \cfrac{1}{\cfrac{\cfrac{1}{h_{h_{dry}}\, \xi_h\, A_{tot_h}}}{1+\xi_h} + \cfrac{\cfrac{1}{h_c\, A_{tot_c}\, \xi_c}}{1+\xi_c}} = \cfrac{1}{\cfrac{\cfrac{1}{38 \times 0.6637 \times 18.68}}{1+0.6637} + \cfrac{\cfrac{1}{34 \times 0.4249 \times 18.68}}{1+0.4249}}$$

$$= 119.1 \text{ BTU/hr }^{\circ}\text{F}$$

$$(UA)_{avail} = 357.1 + 119 = 476.2 \text{ BTU/hr }^{\circ}\text{F}$$

$(UA)_{avail}$ reasonably agrees with $(UA)_{reqd}$ (465 BTU/hr $^{\circ}$F). The modified wet heat exchanger design method is therefore substantiated by the use of test measurements from an actual test run. The same substantiation exists when applied to all runs as is evidenced by Table 5.

TABLE 5

SUBSTANTIATION OF MODIFIED METHOD OF WET HEAT EXCHANGER DESIGN

Run No.	$(UA)_{reqd}$	$(UA)_{avail}$
141	415.2	532.3 1/
142	420.1	532.6 1/
143	373.6	373.5
144	362.4	358.5
145	438.6	417.7
146	444.4	432.4
147	465.0	476.2
148	487.0	470.3

1/ Cooling air discharge temp. measurements were in error.

TABLE 5 CONT.

Run No.	$(UA)_{req'd}$	$(UA)_{avail}$
149	441.0	471.7
150	435.8	453.4
151	343.7	344.8
152	344.9	347.4
153	414.2	416.4
154	415.8	415.7
155	476.0	502.4
156	470.0	503.7

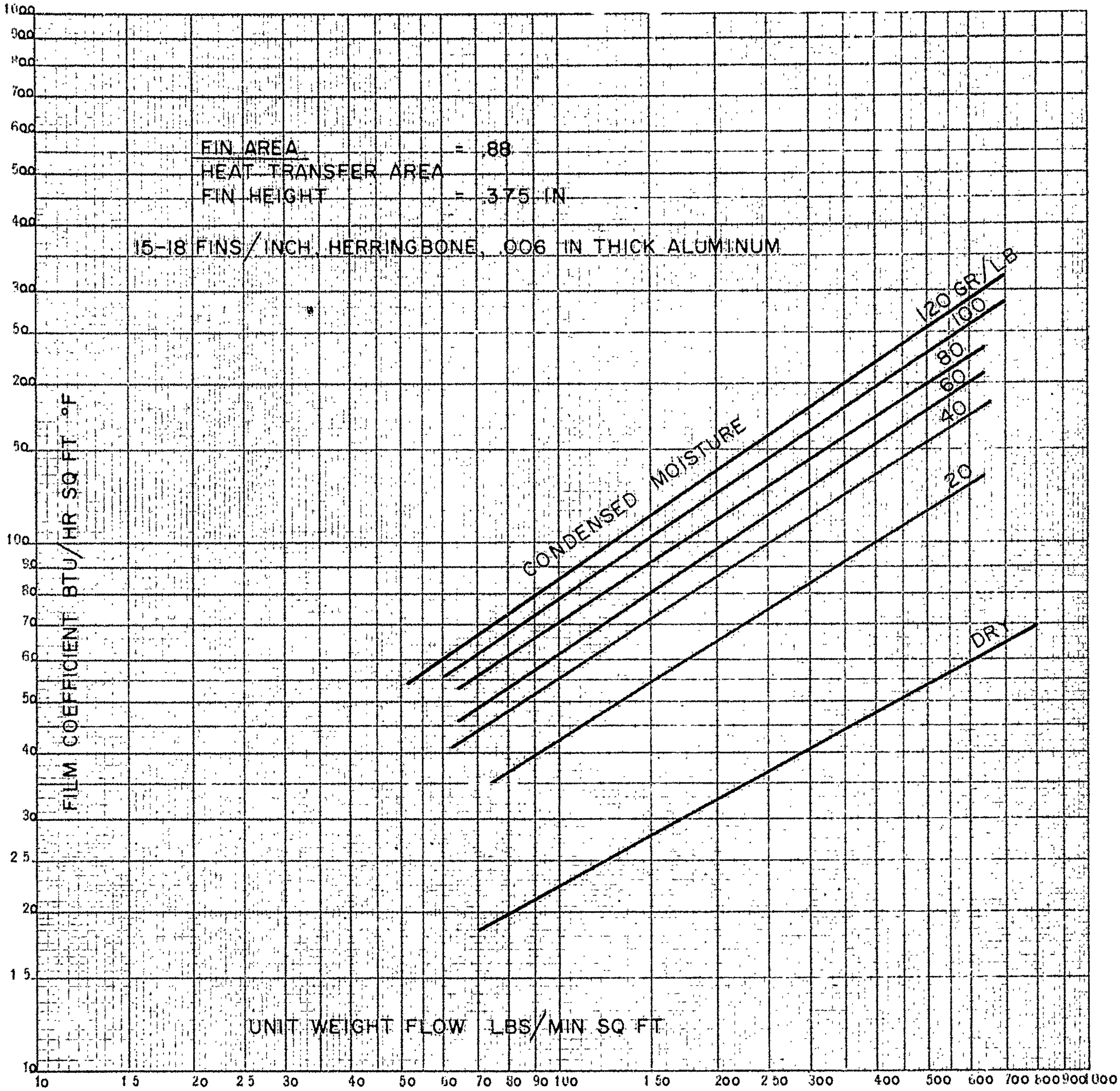


Figure 27. Condensing Film Coefficient for Extended Surface Dehumidification Equipment

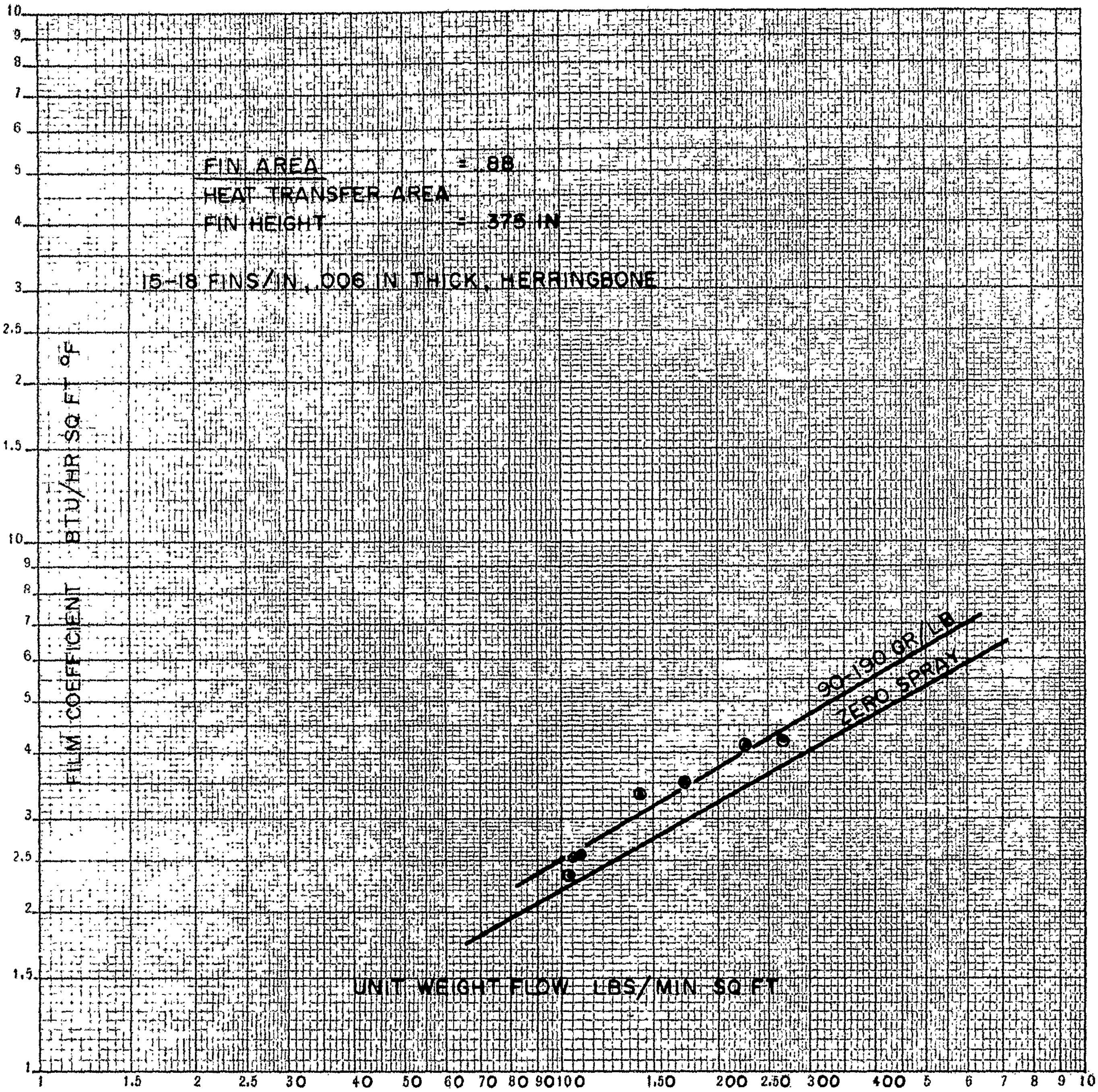


Figure 28. Air Film Coefficient of Heat Transfer for Extended Brazed Aluminum Surfaces with Water Spray

APPENDIX III

DESCRIPTION OF TESTS

The purpose of these tests is to define the operational characteristics of a typical heat exchanger and to substantiate the analytical solution presented in this report under the following conditions:

- Phase 1. Dry cabin air with dry coolant air.
- Phase 2. Condensation in cabin air with dry coolant air.
- Phase 3. Condensation in cabin air, moisture sprayed into coolant air.
- Phase 4. Condensation in cabin air, freezing coolant air (qualitative data only).
- Phase 5. Simulate airplane condition at minimum cabin inlet temperature and moderate humidity and freezing coolant air (qualitative data only).

Phase 1 was conducted to calibrate the test heat exchanger under dry conditions. Phase 2 was intended to give quantitative as well as qualitative information on the performance of a "wet" heat exchanger with dry coolant air. The analytical derivation, as shown in this report, was applied for conditions of Phase 2 only, and it has been assumed that agreement between the analytical and test results of Phase 2 is sufficient to justify the application of the analytical technique for Phase 3. The purpose of Phases 4 and 5 was to qualitatively indicate the possible icing limitations of regenerative operation. Phase 4 duplicates conditions under which the usual regenerative heat exchanger operates.

In order to determine whether 50 gr/lb constituted an icing limitation at the turbine discharge, Phase 5 was instituted. For this condition minimum cabin air temperatures at 50 gr/lb humidity were heat-exchanged against freezing cooling air. It was expected that discharge cabin air under freezing conditions would result. While the heat exchanger discharge configuration does not resemble the turbine discharge physical construction, the information gained on icing might prove valuable.

Testing Procedure

The moisture content of the inlet cabin air requires precise control in measurement of amount and constancy. Stability of injection was maintained by direct measurement of water flow rate. This water was then evaporated into the cabin air. The rate of evaporation was indicated by a constant dew point of the cabin air after reheating with a constant separation rate at the separator and no moisture accumulation in the separator. To accomplish this, two dew point indicators were employed. An Alnor dew pointer was used for quantitative measurement of dew points about 10°F. A G. E. dew point indicator was used for qualitative measurement (for indication of stability) and for visual examination of the cabin inlet air for entrained water. Condensation in the instrumentation lines and in the dew point indicator was considered. Insultation of the rubber dew point gas lines, as well as internal heating of the jacketed dew point indicator to 100° F, served to minimize heat losses. At the various instrumentation stations, dew points were recorded but were found to be in error when compared to actual measurements of water at high dew points. The dew points by G. E. and Alnor measurements were in the order of 80-90°F at the inlet and outlet conditions. The inlet and outlet dew points indicated the same in some cases, even though water was physically separated between the inlet and outlet measuring stations. Measured water flow rate was lower than indicated from dew points by the G. E. dew point indicator. These two conditions illustrated the need for other means for accounting for the water in its circuit. The test equipment design incorporated a reheater after the separator as a check on dew point measurement. This, therefore, became the primary means of accounting for the water and, together with physical separation, was the means of

(1) determining the condensation in the heat exchanger, and
(2) obtaining a balance of injected water + initial water equal to the condensed water + remaining vapor.

Figures 29 and 29a show the test set-up employed. The numbers shown refer to the measuring stations indicated in the test description. Figure 30 shows the schematic of the test set-up.

The water, as vapor in the supply air, was measured by the Alnor dew pointer and is of the order of 0-15°F dew point. The difference between the desired water content and the initial water content was sprayed into the air at station (1) at as high as 600°F at the spray stations. Three spray stations were used to allow time for evaporation. Evaporative cooling dropped the air temperature to (2) (425° F maximum safe temperature of the cooling coil), as the water injected vaporized. This air temperature was then lowered by a water coil to 180° F, the control temperature condition (3). A sample (4) of this air was then put into the G. E. Indicator (dew point indicator where the air passes over a mirror in a chamber with a light and a magnifying glass) to examine the air for entrained moisture. No moisture was allowed to be visible at this point or separated at a baffle type separator (6) at the cabin air heat exchanger inlet (5).

Cooling air was maintained at 40°F (7) at various flow ratios, condensation (8) being required in the cabin air side of such amount to obtain a measurable amount over a 20-minute period. A baffle-type separator (9) then separated some of the condensed water, but not all of it. Moisture taps (10) at the heat exchanger cabin air outlet and downstream of the separator (9) also trapped water. These taps permitted visual examination for condensation. Therefore, when measuring the total amount of water separately, moisture collected from these taps must be included with the moisture collected by the separator. However, throttling of these moisture taps minimized the amount bled in this way. Carry-over at the collection flasks was not tolerated. A maximum bleed of the separator without carry-over was maintained as separator efficiency was thought to be increased with increased bleed. To assure no collection of water in the separator, the separator was checked after each collection. The remaining entrained moisture was re-evaporated with an electric reheater (11). No indication of entrained water was tolerated at the reheater outlet. The heat input to the reheater, considering heater efficiency, was used to calculate the water not collected at the separator which was condensed and subsequently re-evaporated. The dew point at the reheater outlet (12) was used as an indicator of stability only. A constant rate of evaporation, condensation, collection, and re-evaporation was indicated by a constant dew point for at least one-half hour after a minor change of condition was made, such as flow ratio. A longer stabilizing time was required when an injection rate was changed.

Test Results

The first part of the test program, before the effects of condensation were investigated, consisted of a dry calibration of the test heat exchanger, for cabin flow rates of 10, 15, 20, 25 lbs per minute, flow ratios of 1.5, 1, 0.8, 0.5. This was intended to obtain some basis of comparison with wet operation. The results of this calibration are presented in Figure 24, where effectiveness is plotted as a function of cooling air flow for various cabin flow rates.

In the test procedure, the need for continuous checking to ascertain the accuracy of the run was stressed. Before a given run was considered acceptable, both a moisture and heat balance were required. The moisture balance dictates that the sum of the moisture injected into the air stream upstream of the heat exchanger and the ambient humidity present in the compressor bleed air must equal the moisture accountable by downstream measurements at the heat exchanger exit. These measurements include moisture present as saturated vapor, the moisture condensed and separated, and the remaining entrained moisture re-evaporated by the reheater. Satisfaction of both moisture and heat balances constituted an acceptable run, and only results from these runs were used in subsequent calculations.

Comparison of test and analytically predicted heat exchanger performance for those runs in which both heat and moisture balances are possible is given in Table 6. These test results are plotted on Figures 20-23. It is quite clear that close agreement exists between these two values, indicating that the proposed analytical method for "wet" heat exchanger design is fairly accurate for extended surface heat transfer equipment similar to the test heat exchanger.

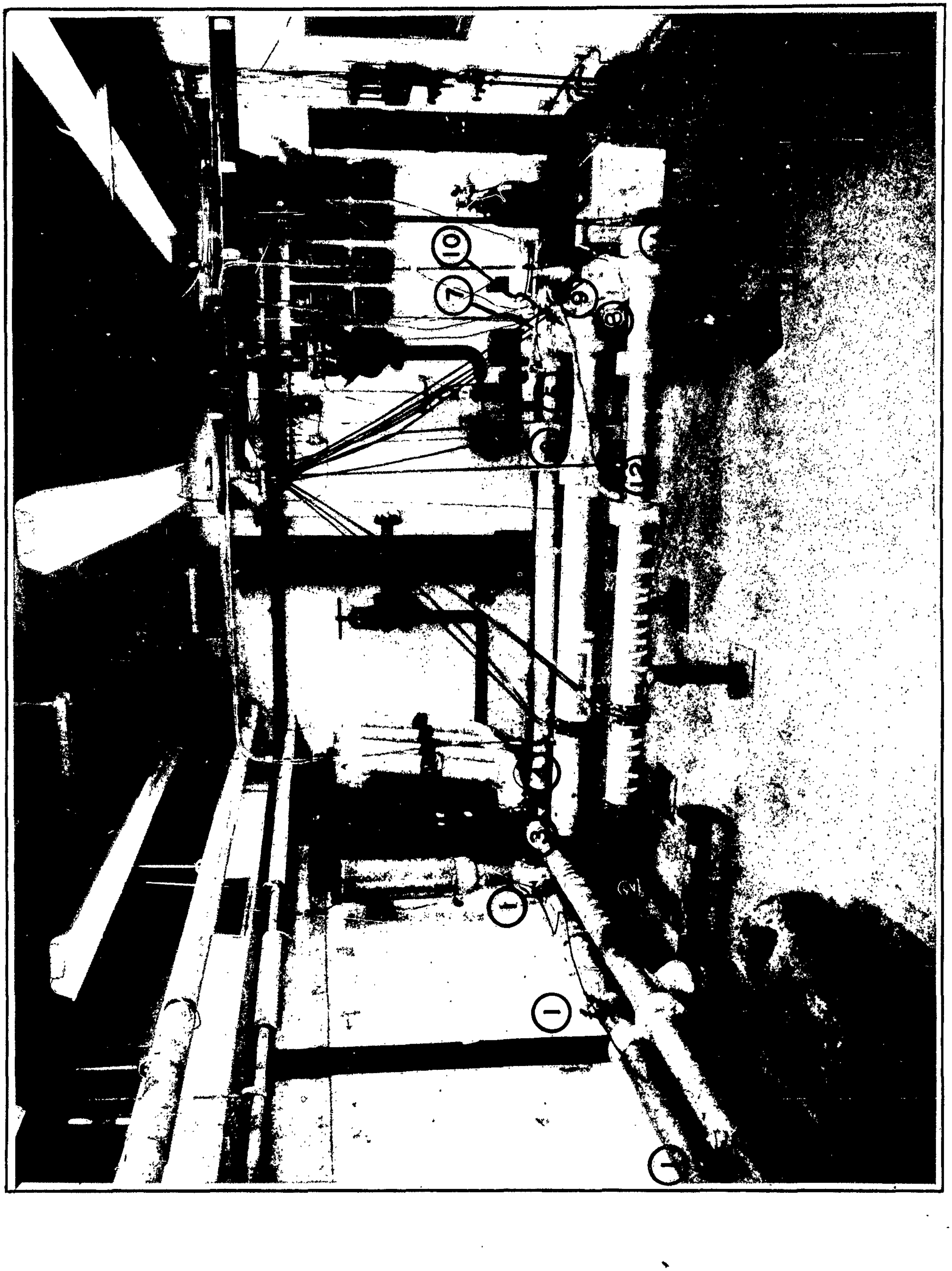


FIGURE 29. TEST INSTALLATION; OVERALL VIEW

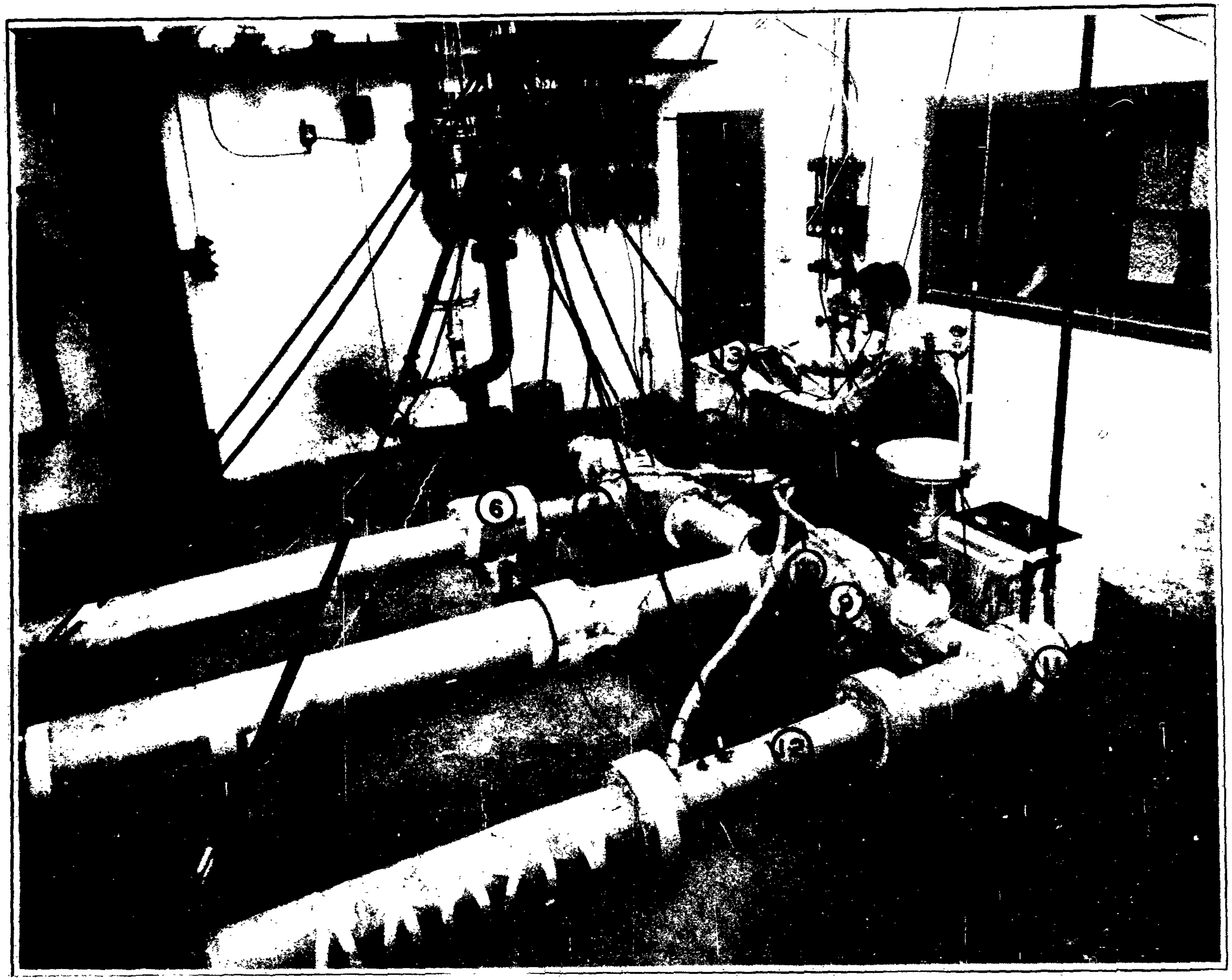


FIGURE 29-A. TEST INSTALLATION; THE MEASUREMENT OF DEWPOINT

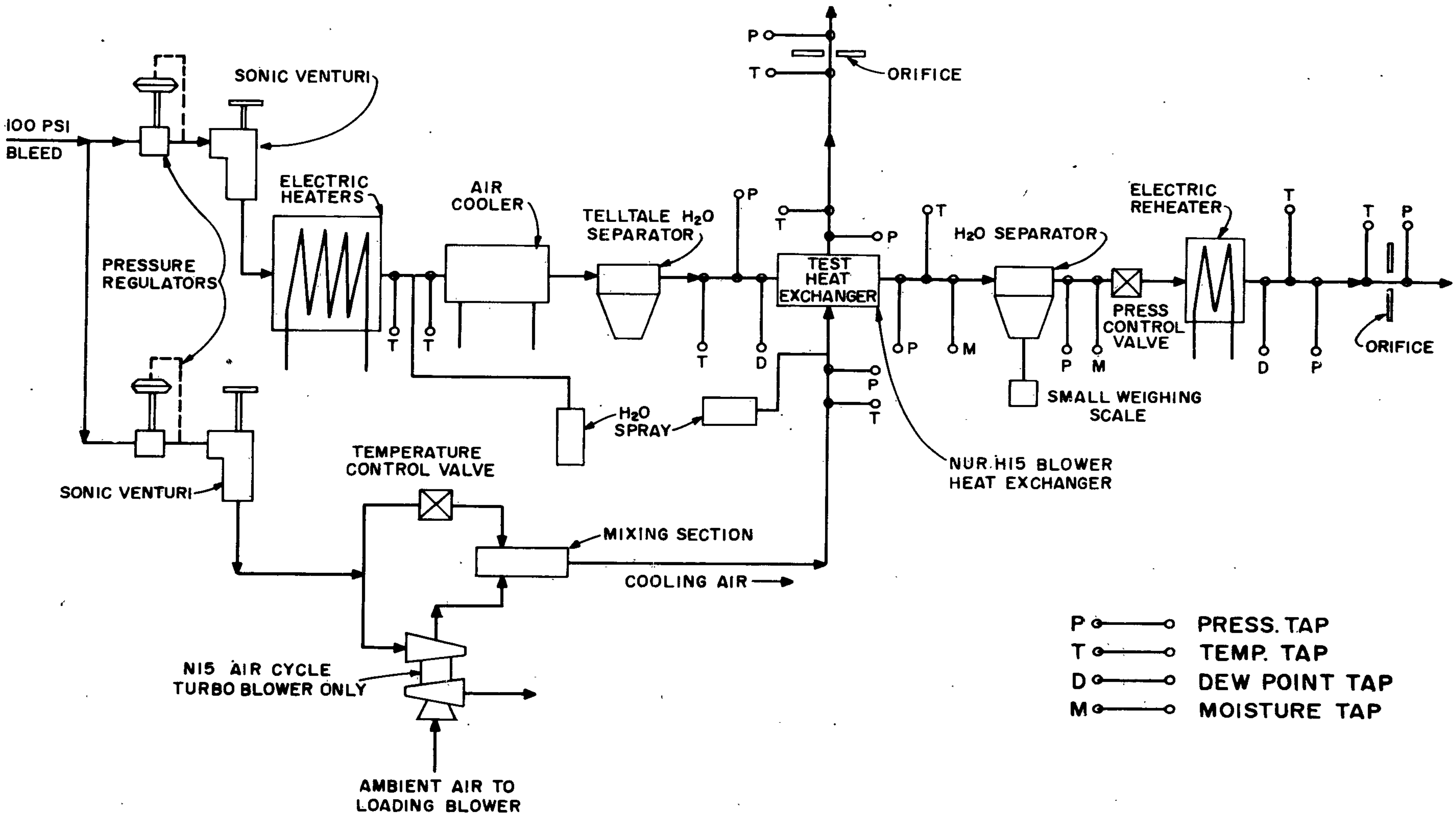


Figure 30. Schematic Diagram of Regenerative Heat Exchanger Test Set-Up

TABLE 6

COMPARISON OF TEST RESULTS WITH PREDICTED PERFORMANCE

	Temperature Ratio Parameter			Temperature Ratio Parameter	
Run No.	Test	Analytical	Run No.	Test	Analytical
25	0.3661	0.340	71	0.4143	0.404
26	0.3615	0.342	72	0.4173	0.404
27	0.3661	0.336	77	0.4894	0.488
45	0.4275	0.415	78	0.4875	0.484
46	0.4320	0.412	79	0.4661	0.460
47	0.4053	0.386	80	0.4699	0.458
48	0.4060	0.385	81	0.4525	0.436
49	0.4371	0.416	82	0.4478	0.436
50	0.4359	0.416	83	0.4161	0.396
51	0.4406	0.417	87	0.4161	0.406
52	0.3933	0.394	88	0.4161	0.406
53	0.3933	0.394	89	0.3857	0.382
54	0.4184	0.393	90	0.3875	0.383
65	0.4748	0.476	91	0.3903	0.379
66	0.4643	0.467	92	0.3848	0.381
67	0.4604	0.460	93	0.3901	0.381
68	0.4357	0.434	94	0.3820	0.362
69	0.4388	0.434	95	0.3786	0.360
70	0.4357	0.432	96	0.3411	0.341

An important measure of the effectiveness of water injection is given by spray efficiency, defined as the fraction of sprayed moisture that must re-evaporate to complete the heat balance across the heat exchanger. The results of the third phase of the test are given in Table 7 where the spray efficiencies are tabulated.

The fourth phase of the test program attempted to discover possible icing limitations of regenerative operation (at the turbine discharge and in the regenerator). First, duplication of the regenerative heat exchanger action was made. For this, high temperature air with a relatively high amount of humidity was directed into the hot air side. The cold air entered at below freezing temperatures. The results of this test indicated that no ice-blocking on the hot air side occurred when the condensate came in contact with below freezing cooling air.

The last phase of the test program consisted of determining whether 50 gr/lb moisture content in the hot air is sufficient to ice-block the heat exchanger when subjected to below freezing cooling air temperature, the assumption being that, if ice-blocking does not occur in the heat exchanger, it will not occur in the turbine discharge. The results of this test are not considered conclusive. It appears that some ice-blocking may have occurred on the cold air side, since the pressure drop across the cooling air side increased. The hot air side was not ice-blocked, and it was the hot air side which contained 50 gr/lb dry air. In the absence of contradictory data, it is felt that moisture contents of 50 gr/lb are not large enough to present icing problems of any consequence.

TABLE 7

SPRAY EFFICIENCIES FROM TEST RESULTS
Based on 40°F Dry Coolant Air, 31 in. Hg abs

Run No.	Cooling Flow lb/min	Moisture Spray gr/lb	Spray Eff.	Run No.	Cooling Flow lb/min	Moisture Spray gr/lb	Spray Eff.
133	12.11	101	0.1990	145	16.03	201	0.3225
134	12.47	99.5	0.1315	146	17.02	190	0.3073
135	22.53	102	0.6131	147	19.64	192.5	0 3371
136	22.49	104	0.6127	148	19.58	193	0.3216
137	12.34	195	0.1913	149	20.03	88.5	0.4802
138	12.40	191	0.1843	150	18.39	95.5	0.5961
139	22.72	201	0.3506	151	12.19	88	0.5667
140	22.95	199	0.3485	152	12.31	85	0.5818
141	23.77	195	0.4950	153	16.7	94.5	0.49
142	23.67	196	0.5101	154	16 55	89.6	0.4747
143	12.94	170	0.3499	155	22.44	89.5	0.4937
144	12.72	173	0.3211	156	22.44	87	0.5123

APPENDIX IV

SAMPLE CALCULATION: PERFORMANCE OF THE HPCC SEA LEVEL-FLIGHT CONDITION A

The following values were selected for design. They were found wherever possible from the referenced illustrations and test data. All subscripts refer to the schematic drawing of the HPCC shown in Figure 10.

Cabin air:
- Package inlet pressure, P_b - 182.6 in. Hga (Figure 31).
- Package inlet temperature, T_b = 1048°R (Figure 32).

Cooling air:
- Package inlet temperature, T_1 = 612°R (Figure 33).
- Ram inlet duct efficiency η_r = 85% (assumed).
- Ram air inlet duct friction factor, K = 0.00239 (assumed)
- Ram air discharge duct friction factor, K = 0.0001 (assumed)

Plane conditions:
- Speed - 540 mph (Figure 34)
- Pressure - P_{cab} = 29.92 in. Hga (Figure 35)

Ambient conditions:
- Temperature; Ta = 560°R (Figure 3
- Pressure, P_a 29.92 in. Hga (Figure 36)
- Humidity, H_i = 184 gr/lb dry air (Figure 36)

(1) Ram temperature rise:

$$\Delta t_{ram} = \frac{V^2}{2gJc_p} = \frac{(540)^2(5280)^2}{(3600)^2(2)(32.2)(778)(.24)} = 52°F$$

(2) Cooling air inlet pressure to package:

$$\pi_r = 1 + \eta R\left[\left(\frac{\Delta t_{ram}}{Ta} + 1\right)^{3.5} - 1\right] \quad \text{(ram pressure ratio)}$$

$$= 1.3094$$

$$P_1 = \pi ram\,(P_a) = 39.177 \text{ in. Hga}$$

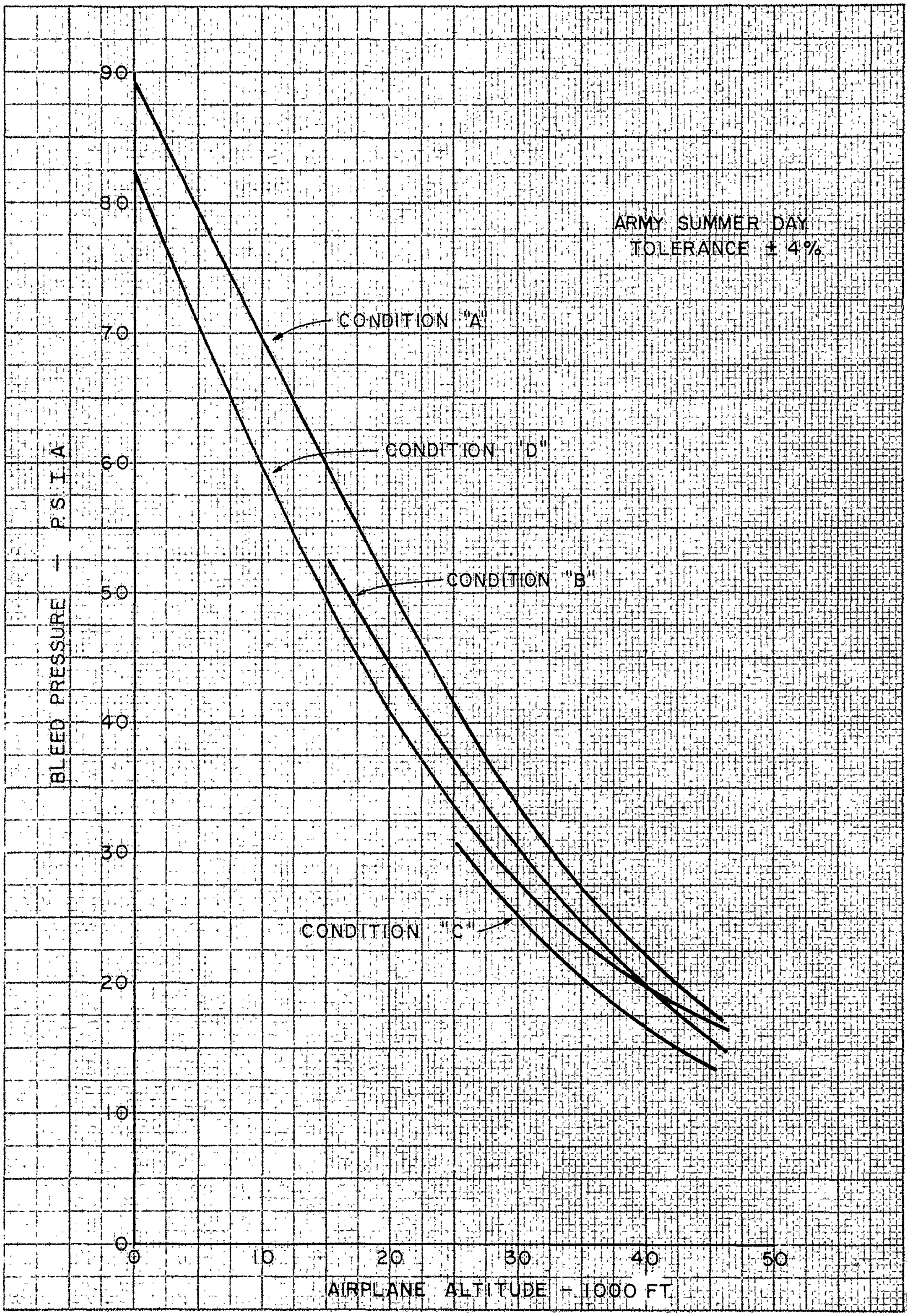


Figure 31. Engine Bleed Pressure VS Altitude.

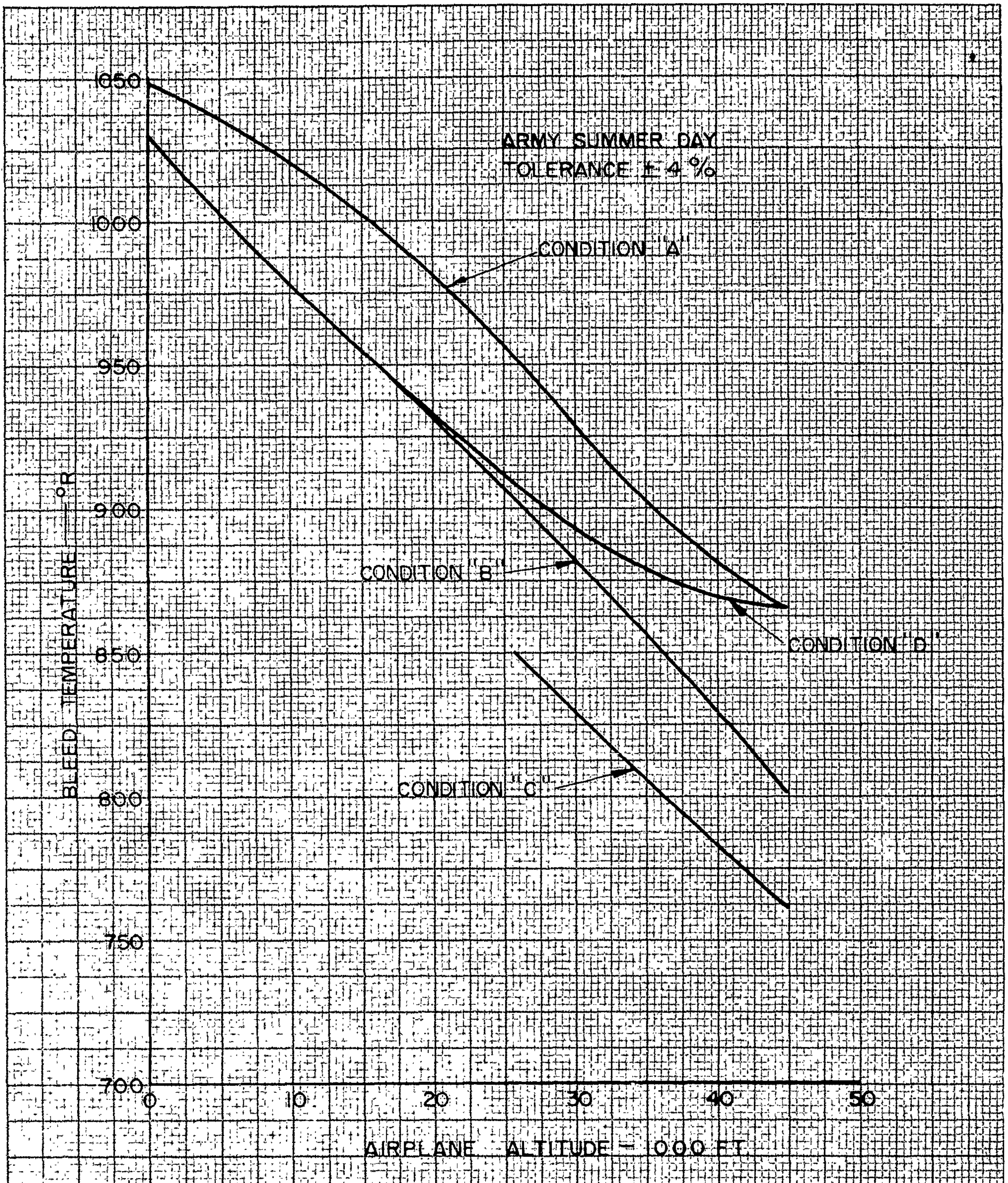


Figure 32. Engine Bleed Temperature VS Altitude

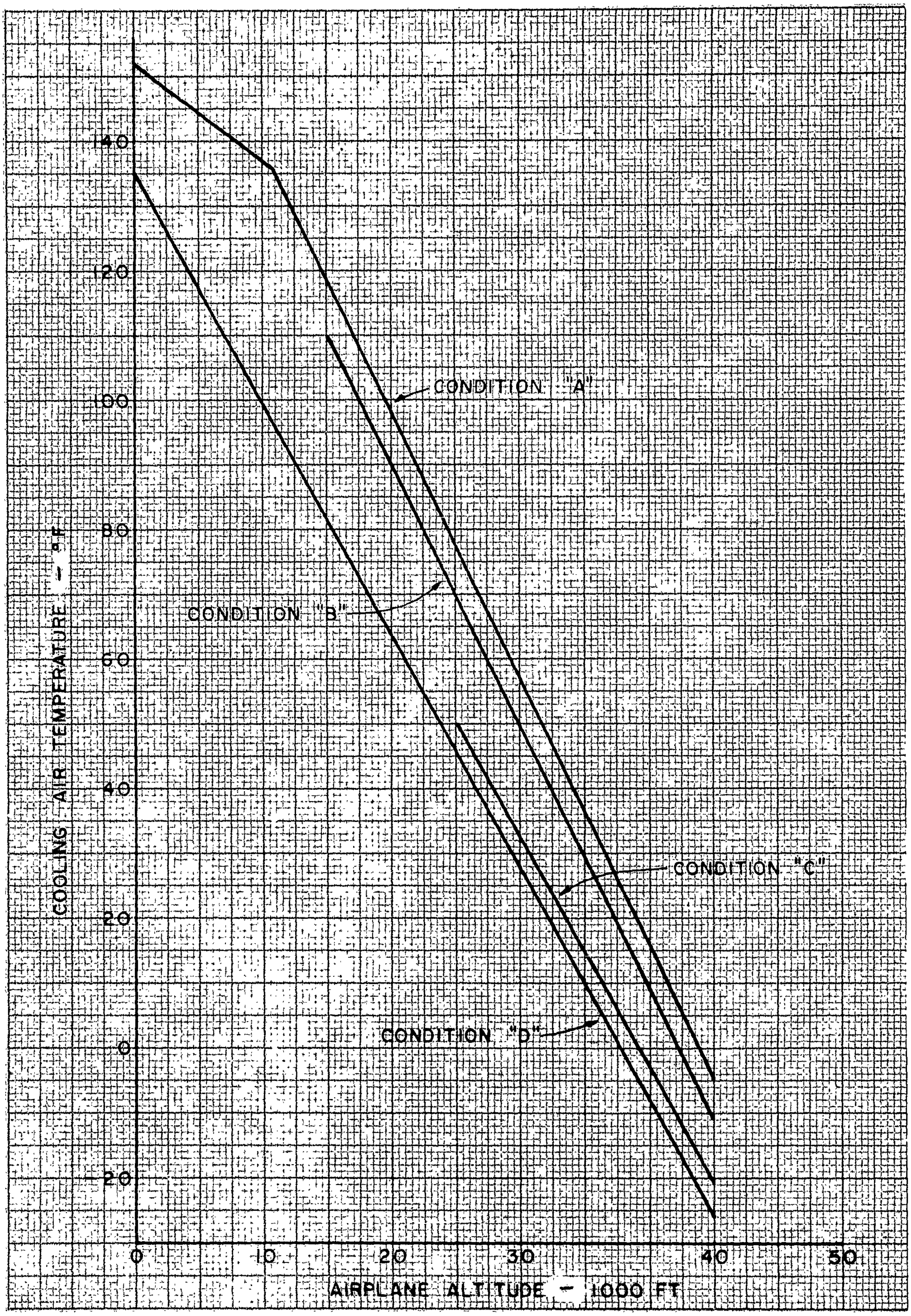


Figure 33. Cooling Air Temperature as a Function of Altitude.

(3) Cooling air inlet pressure to ram heat exchanger:

Assume cooling air flow $w_{r1} = 170$ lb/min

$$\Delta p = \frac{(w_{r1})^2}{13.56} \frac{K}{\delta}$$

$$\delta = \frac{\rho}{0.0765} = \frac{1.325}{0.0765} \frac{P_1}{T_1} = 1.1084 \text{ (Density ratio)}$$

$$\Delta P_{1-2} = \frac{(170)^2 \times 0.00239}{13.56 \times 1.1084} = 4.596 \text{ in. Hg}$$

$$P_2 = P_1 - \Delta P_{1-2} = 39.177 - 4.596 = 34.581 \text{ in. Hga}$$

(4) Cooling air discharge pressure from ram heat exchanger:

Assume hot air flow = 53 lb/min

From Figure 37 (page 124)

$$\eta_x = 0.932$$

$$T_3 = T_2 + \frac{\eta_x \, w_b}{w_{r1}} \left[T_9 - T_2\right]$$

$$T_3 = 612 + \frac{.932\,(53)}{170} \left[1048 - 612\right] = 738.7^{\circ}R$$

Assume $P_{2-3} = 32.844$ in. Hga

$$T_{2-3} = \frac{738.7 + 612}{2} = 675.4^{\circ}R$$

$$\rho_{2-3} = 1.325 \times \frac{32.844}{675.4} = 0.0644 \text{ lb/ft}^3$$

From Figure 39: (page 135)

$$\sigma \Delta P_{2-3} = 2.924 \text{ in. Hg}$$

$$\Delta P_{2-3} = 2.924 \times \frac{0.0765}{0.0644} = 3.474 \text{ in Hg}$$

$$P_{2-3} = 34.581 - \frac{3.474}{2} = 32.844 \text{ in. Hga}$$

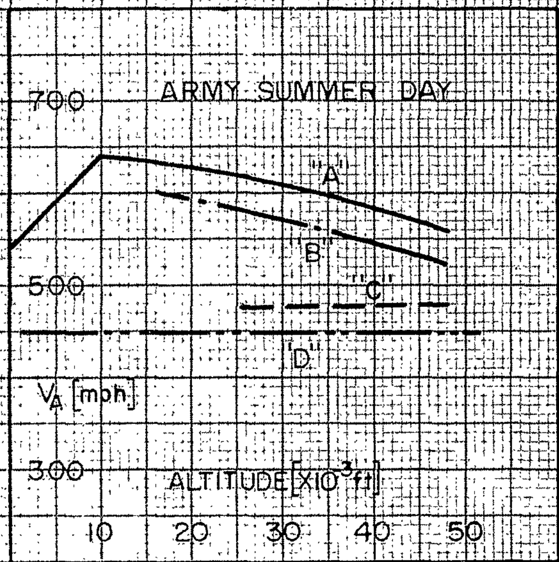


Figure 34. Airplane Speed as a Function of Altitude

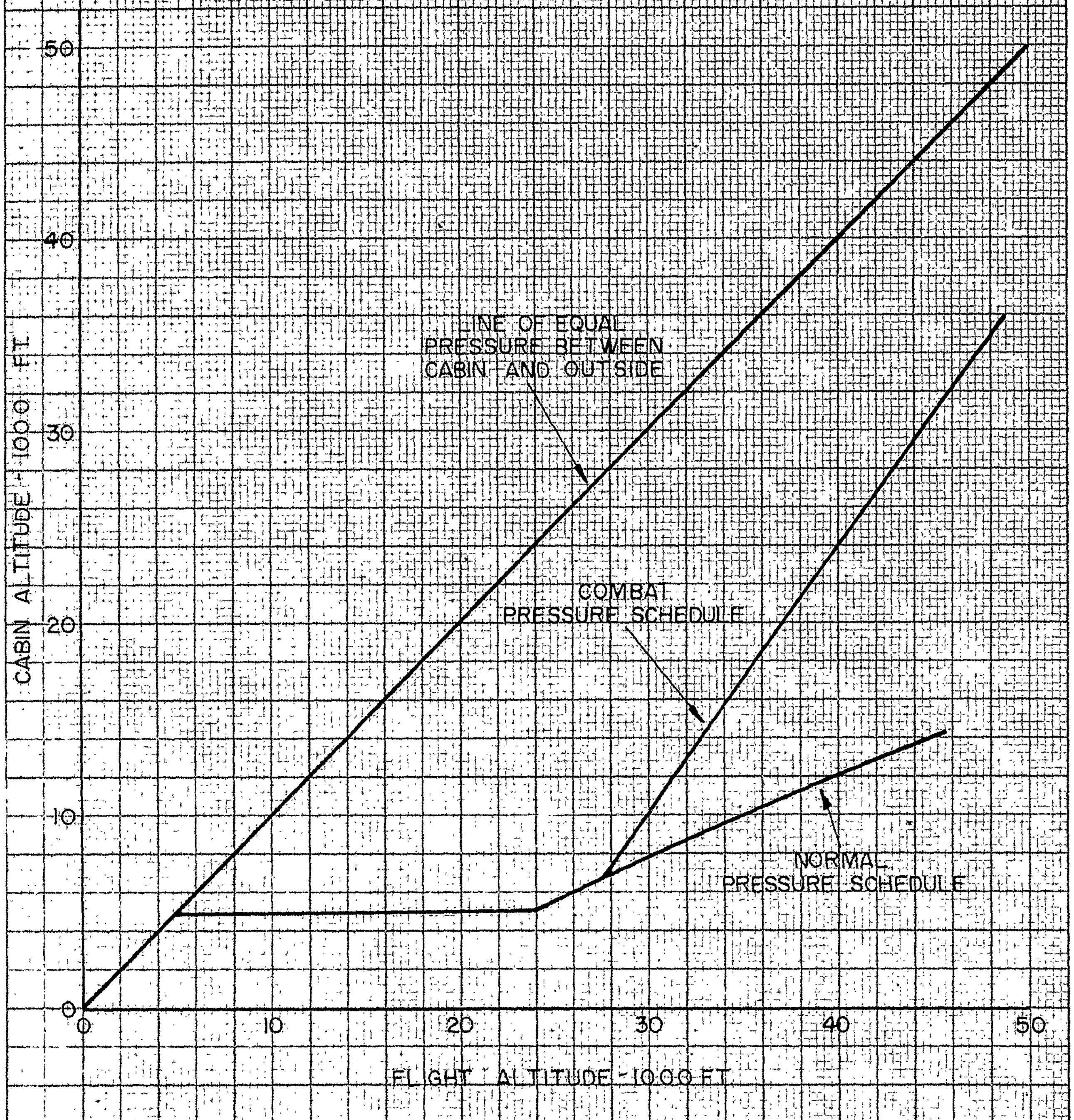


Figure 35. Cabin Pressure Schedule

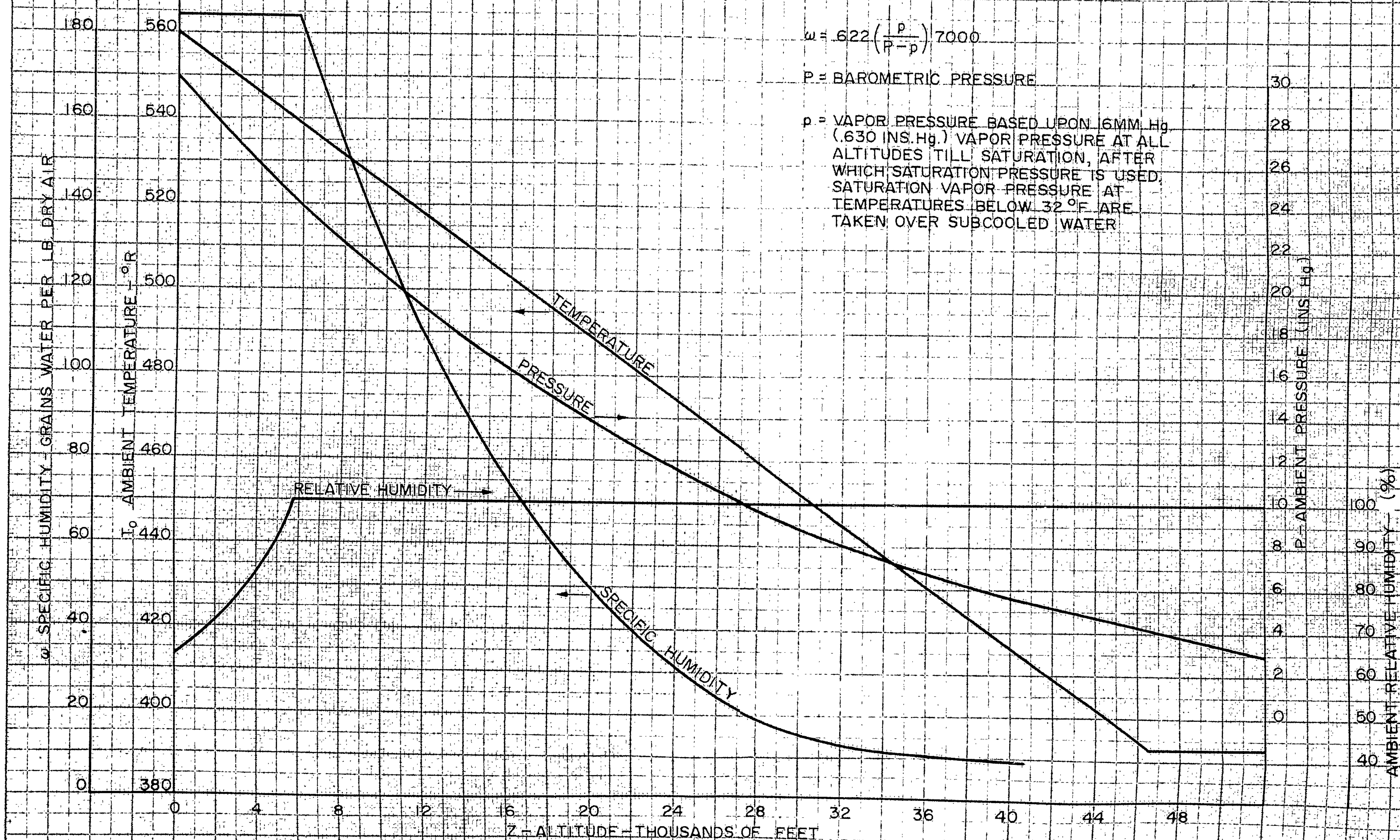


Figure 36. Ambient Temperature and Pressure Conditions Based on Army Summer Day and Relative and Specific Humidities as Functions of Altitude Humidity Functions Modified in Accordance with Governing Specifications

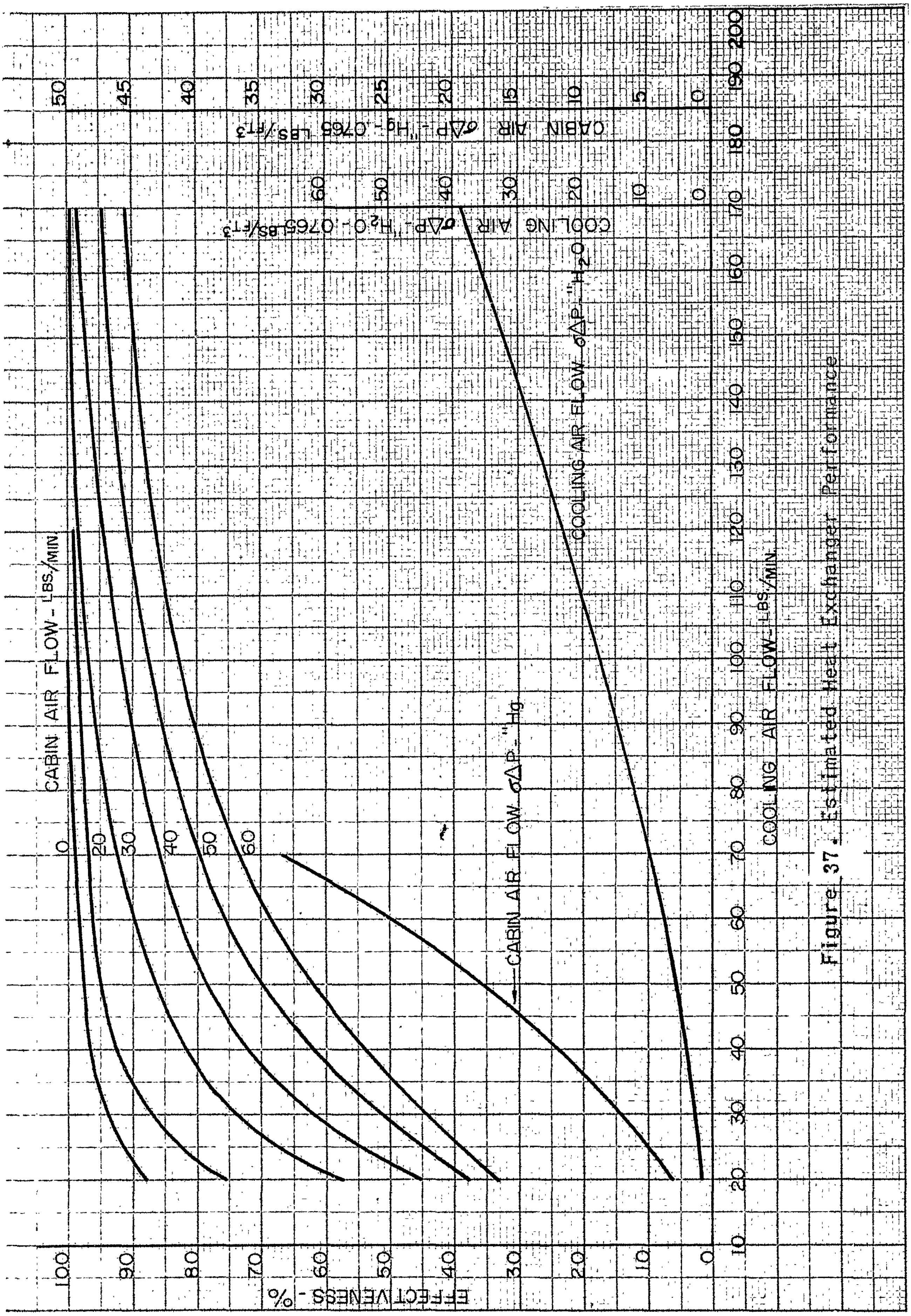


Figure 37. Estimated Heat Exchanger Performance

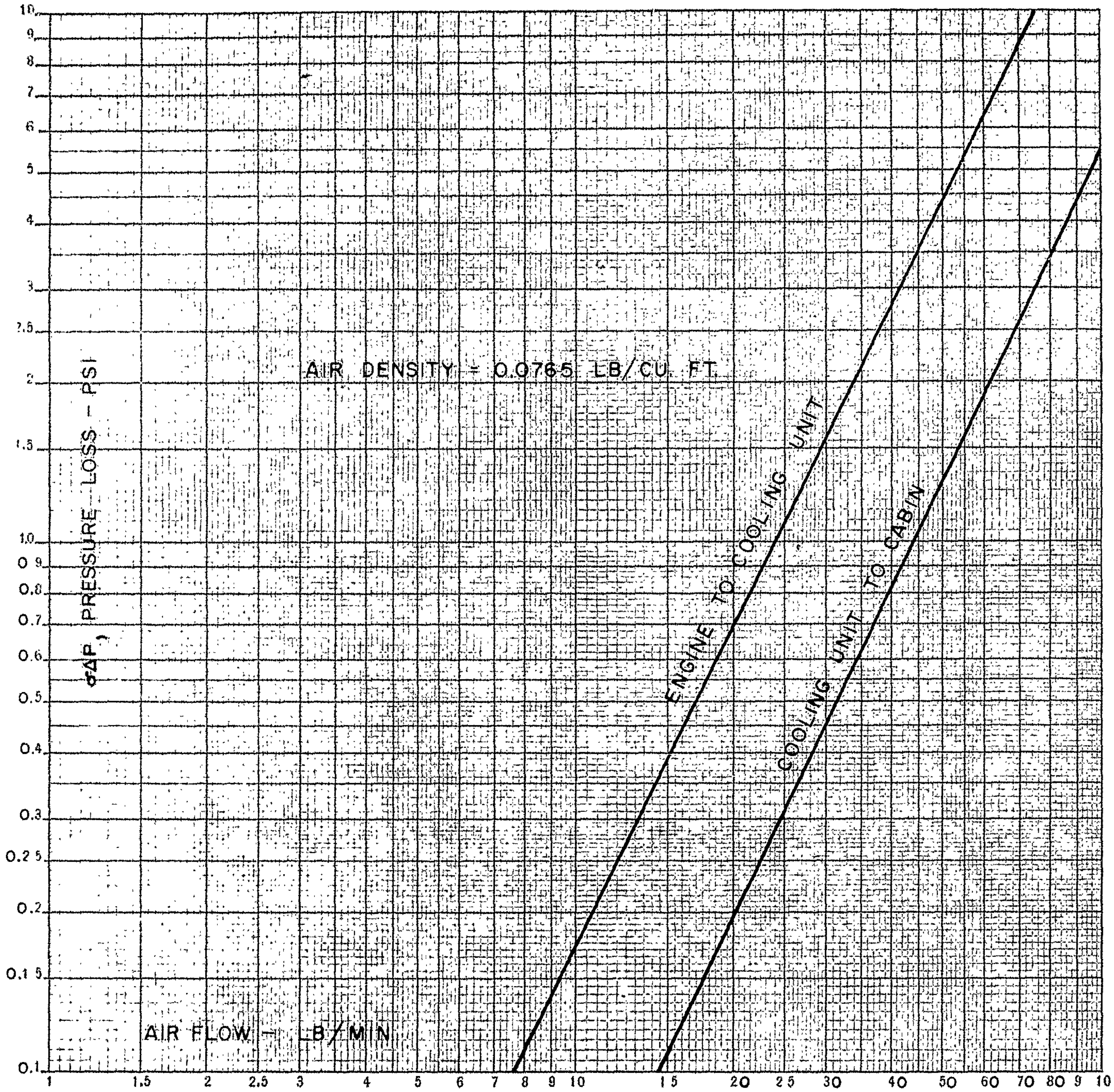


Figure 33. Duct System $\sigma\Delta P$ VS Cabin Airflow

The assumed value of P_{9-10} is correct and

$$P_{10} = 179.26 - 5.53 = 173.73 \text{ in. Hga}$$

(7) Compressor operating characteristics:

$$F = \frac{w_b \sqrt{T_{10}}}{P_{10} D_c^2} = \frac{53 \sqrt{641.6}}{173.73 \times (3.85)^2} = 0.5213$$

Assume $\pi_c = 1.710$

From Figure 12 (Page 38) read

Adiabatic efficiency, $\eta_c = 0.629$

Inlet Mach No. = Nm = 0.829

Solving for the speed of the unit

$$N = \frac{11,230 \text{ (Nm)} \sqrt{T_{10}}}{D_c} = \frac{11,230 \ (0.829) \sqrt{641.6}}{3.85} = 61,250 \text{ rpm}$$

(8) Hot air pressure, temperature at discharge from compressor:

Let $Y_c = \pi_c^{\frac{k-1}{k}} - 1$

$$Y_c = (1.71)^{0.283} - 1 = 0.16396$$

$$\Delta t_{10-11} = \frac{Y_c \quad T_{10}}{\eta_c} = \frac{0.16396 \ (641.6)}{0.629} = 167.2^{\circ}F$$

$$T_{11} = 641.6 + 167.2 = 808.8^{\circ}R$$

$$P_{11} = 1.71 \times 173.73 = 297.1 \text{ in. Hga}$$

(9) Cooling air inlet pressure to ram heat exchanger: Assume cooling air flow $w_{r_2} = 171$ lb/min

$\sigma = 1.1084$ (see Page 120)

$$\Delta P_{1-5} = \frac{0.00239 \ (171)^2}{13.56 \times 1.1084} = 4.65 \text{ in. Hg}$$

$P_5 = 39.177 - 4.65 = 34.527$ in. Hga

(10) Cooling air discharge pressure from ram heat exchanger:

From Figure 37 read

$\eta_x = 0.933$

$$T_7 = 612 + \frac{0.933 \times 53}{171} (808.8 - 612) = 668.9^{\circ}R$$

$$T_{5-7} = \frac{612 + 668.9}{2} = 640.5^{\circ}R$$

Assume $P_{5-7} = 32.868$ in. Hga

$$\rho_{5-7} = \frac{1.325 \times 32.868}{640.5} = 0.068 \text{ lb/ft}^3$$

From Figure 37:

$\sigma \Delta P_{5-7} = 2.95$ in. Hg

$$\Delta P_{5-7} = \frac{2.95 \times 0.0765}{0.068} = 3.319 \text{ in. Hg}$$

$$P_{5-7} = 34.527 - \frac{3.319}{2} = 32.868 \text{ in. Hga}$$

The assumed value of P_{5-7} is correct and

$P_7 = 34.527 - 3.319 = 31.208$ in. Hga

(11) Cooling air pressure at mixing point:

Assumed density of cooling flows after mixing
$\rho_{4-8} = 0.0574$ lb/ft^3

$w_4 = 170 + 171 = 341$ lb/min

$$\sigma_4 = \frac{0.0574}{0.0765} = 0.7502$$

$$\Delta P_{4-8} = \frac{0.0001 \times (341)^2}{13.56 \times 0.7502} = 1.143 \text{ in. Hg}$$

$$P_{4-8} = 29.92 + \frac{1.143}{2} = 30.492 \text{ in. Hga}$$

$$T_8 = \frac{(^{w}r_1)\ (T_3) + (^{w}r_2)\ (T_7)}{^{w}r_1 + {}^{w}r_2} = 703.7^{\circ}R$$

$$\rho_{4-8} = 1.325 \times \frac{30.492}{703.7} = 0.0574 \text{ lb/ft}^3$$

The assumed value of ρ 4-8 is correct

$P_4 = 29.92 + 1.143 = 31.063$ in. Hga 1/

(12) Hot air discharge pressure from ram heat exchanger:

$$T_{12} = 808.8 - (0.933)\ (808.8 - 612) = 625.2^{\circ}R$$

Assume $P_{11-12} = 295.7$ in.Hga

$$T_{11-12} = \frac{808.8 + 625.2}{2} = 717^{\circ}R$$

$$\rho_{11-12} = 1.325 \times \frac{295.7}{717} = 0.5464 \text{ lb/ft}^3$$

From Figure 37:

$$\sigma\, \Delta P_{11-12} = 20.0 \text{ in. Hg}$$

$$\Delta P_{11-12} = 20 \times \frac{0.0765}{0.5464} = 2.80 \text{ in. Hg}$$

$$P_{11-12} = 295.7 \text{ in. Hga}$$

1/ This compares very well with $P_3 = 31.107$ in. Hga and $P_7 = 31.208$ in. Hga. (See pages 125, 128). The cooling flow rates as previously assumed on pages 125, 128 are therefore reasonably correct.

The assumed value of P_{11-12} is correct and

$$P_{12} = 297.1 - 2.8 = 294.3 \text{ in. Hga}$$

The saturation vapor content of the hot air is now determined:

$$W_{s_{12}} = \frac{4354 \quad P_{s_{12}}}{P_{12} - P_{s_{12}}}$$

$$P_{s_{12}} = 10.88 \text{ in. Hga}$$

$$W_{s_{12}} = \frac{4354 \times 10.88}{294.3 - 10.88} = 168 \text{ gr/lb}$$

This indicates that 184 - 168 = 16 gr/lb dry air had condensed at this point.

In the design of the split regenerator, conservative practice dictates that the regenerator be sized for the maximum inlet humidity possible. For this case, then, the initial condensation of 16 gr/lb dry air is neglected. This in no way alters the performance of the system but rather inparts an additional safety factor.

(13) The physical configuration of the split regenerator is as follows:

Second Cooling Pass:

	Hot Side	Cooling Side
Fin type:	18 FPI Herr.0.375 in. x 0.01 in.;	18 FPI Herr0.375in.x0.01 in.
Number passages	12	13
Number passes	2	1
Free flow area, -sq ft	0.1644	0.1913
Surface area - sq ft	97.2	105.2
Effective length/pass-in	6.446	12

First Cooling Pass:

	Hot Side	Cooling Side
Fin type:	18 FPIHerr. 0.375 in. x 0.01 in.	18 FPIHerr. 0.375 in. x 0.01 in.
Number passages	6	7
Number passes	1	1
Free flow area - sq ft	0.1644	0.187
Surface area - sq ft	88.2	102.8
Effective length/pass-in	11.702	12

(14) Hot air discharge pressure and temperature from second cooling pass:

Assume $w_{reg} = 31.5$ lb/min

$P_{16a} = 30.816$ in. Hga

$T_{16a} = 124.2^{o}F$

$W_{c_{tot}} = 179.08$ gr/lb dry "hot air"

The performance of the regenerative heat exchanger is found by assuming a number of values for η_{wet} until the basic equation of $(UA)_{reqd} = (UA)_{avail}$ is satisfied.

After a number of trials, assume $\eta_{wet} = 0.584$

$$T_{12a} = 165.2 - 0.584\ (165.2-124.2) = 141.3^{o}F$$

$$P_{s_{12a}} = 6.0840 \text{ in. Hga}$$

$$W_{s_{12a}} = \frac{4354 \times 6.0840}{294.3 - 6.084} = 92 \text{ gr/lb dry air}$$

$$W_{c_1} = 184 - 92 = 92 \text{ gr/lb dry air (condensed)}$$

Reasonable values for moisture separator and spray efficiency as assumed for these conditions are:

$$\eta_{sep} = 0.81$$

$$\eta_{sp} = 0.727$$

$$W_{sp} = W_{c_{tot}} \quad (\eta_{sep}) \ (\frac{w_b}{w_{reg}})$$

$$W_{sp} = \frac{(179.08)(0.81)(0.53)}{(31.5)} = 244 \text{ gr/lb cooling air}$$

Corrected cooling air inlet temperature is:

$$T_{16a_c} = T_{16_a} - 0.588 \mathcal{J} \eta_{sp} \ W_{sp}$$

For $\mathcal{J} = 0.3$ (see Page 86).

$$T_{16a_c} = 124.2 - \left[0.588 \times 0.3 \times 0.727 \times 244\right] = 92.9^{o}F$$

Heat transferred to cooling air is:

$$Q = 14.5 \ w_b \left[T_{12} - T_{12a} + 0.6W_{c_1}\right] + \frac{w_b H_i}{253} \left[T_{12} - T_{12a}\right]$$

$$Q = 14.5 \times 53 \quad \left[165.2 - 141.3 + 0.6 \ (92.09)\right] + \frac{53 \times 184}{253} \left[165.2 - 141.3\right]$$

$Q = 61,835$ BTU/hr

Cooling air discharge temperature is:

$$T_{17} = T_{16a} + \frac{Q - 8.66 \left[W_{sp} \quad w_{reg} \quad \eta_{sp}\right]}{14.5 \quad w_{reg}}$$

$$= 124.2 + \frac{61,835 - 8.66 \left[244 \times 31.5 \times 0.727\right]}{14.5 \times 31.5} = 153.6^{o}F$$

The effective mean temperature difference for cross flow exchangers is now calculated.

$$\frac{T_{17} - T_{16a_c}}{T_{12} - T_{16a_c}} = \frac{153.6 - 92.9}{165.2 - 92.9} = 0.8394$$

$$\frac{T_{12} - T_{12a}}{T_{12} - T_{16a_c}} = \frac{165.2 - 141.3}{165.2 - 92.9} = 0.3306$$

Referring to Figure 26, (page 102),

$$\frac{\Delta t_m}{T_{12} - T_{16a_c}} = 0.291$$

$$\Delta t_m = 0.291\ (165.2 - 92.9) = 21.04^{\circ}F$$

For exchangers with more than one pass, the effective temperature difference more nearly approaches the LMTD 1/ value. For this case, LMTD = $25.6^{\circ}F$.

To avoid any speculation the cross flow temperature difference will be employed.

$$(UA)reqd = \frac{Q}{\Delta t_m} = \frac{61,835}{21.04} = 2940\ BTU/hr\ {}^{\circ}F$$

Determination of $(UA)_{avail}$:

$$\text{Hot side unit weight flow} = \frac{53}{0.1644} = 322.4\ lb/min\ sq\ ft$$

$$\text{Cooling side unit weight flow} = \frac{31.5}{0.1913} = 164.8$$

Referring to Figures 27 and 28 (pages 105 and 106),

Hot side film coefficient = 160 BTU/hr sq ft $^{\circ}F$

Cold side film coefficient = 33.2 BTU/hr sq ft $^{\circ}F$

$$\frac{1}{(UA)} = \frac{1}{(hA)_{\text{hot side}}} + \frac{1}{(hA)_{\text{cold side}}}$$

1/ Log Mean Temperature Difference

$$\frac{1}{(UA)} = \frac{1}{(160)(97.2)} + \frac{1}{(33.2)(105.2)}$$

$$(UA)_{avail} = 2840 \text{ BTU/hr}^{o}\text{F}$$

Since $(UA)_{avail}$ reasonably agrees with (UA) reqd, η wet as assumed on page 131 is correct. The pressure drop through the exchanger is now determined.

Hot side:

Referring to Figure 39,

$$\frac{\sigma \Delta P_{12-12a}}{l} = 2.36 \text{ in. } H_2O/\text{in. length, pass}$$

$$\sigma \Delta P_{12-12a} = 2.36 \times 6.446 \times 2 = 30.425 \text{ in. } H_2O$$

$$T_{12-12a} = \frac{165.2 + 141.3}{2} = 153.25^{o}F = 613.25^{o}R$$

$$\Delta P_{12-12a} = \frac{\sigma \Delta P_{12-12a} \ (T_{12-12a})}{236 \ (P_{12})}$$

$$\Delta P_{12-12a} = \frac{30.425 \times 613.25}{236 \times 294.3} = 0.2686 \text{ in. Hg (core drop)}$$

Multiplying this figure by a factor of two, so as to include the additional pressure drop from bends, elbows and transitions,

$$\Delta P_{12-12a} = 2 \times 0.2686 = 0.5373 \text{ in. Hg}$$

$$P_{12a} = 294.3 - 0.5373 = 293.8 \text{ in.Hga}$$

Cold side:

Referring to Figure 39.

$$\frac{\sigma \Delta P_{16a-17}}{l} = 0.74 \text{ in. } H_2O/\text{in. length, pass}$$

$$\sigma \Delta P_{16a-17} = 0.74 \times 12 = 8.88 \text{ in. } H_2O$$

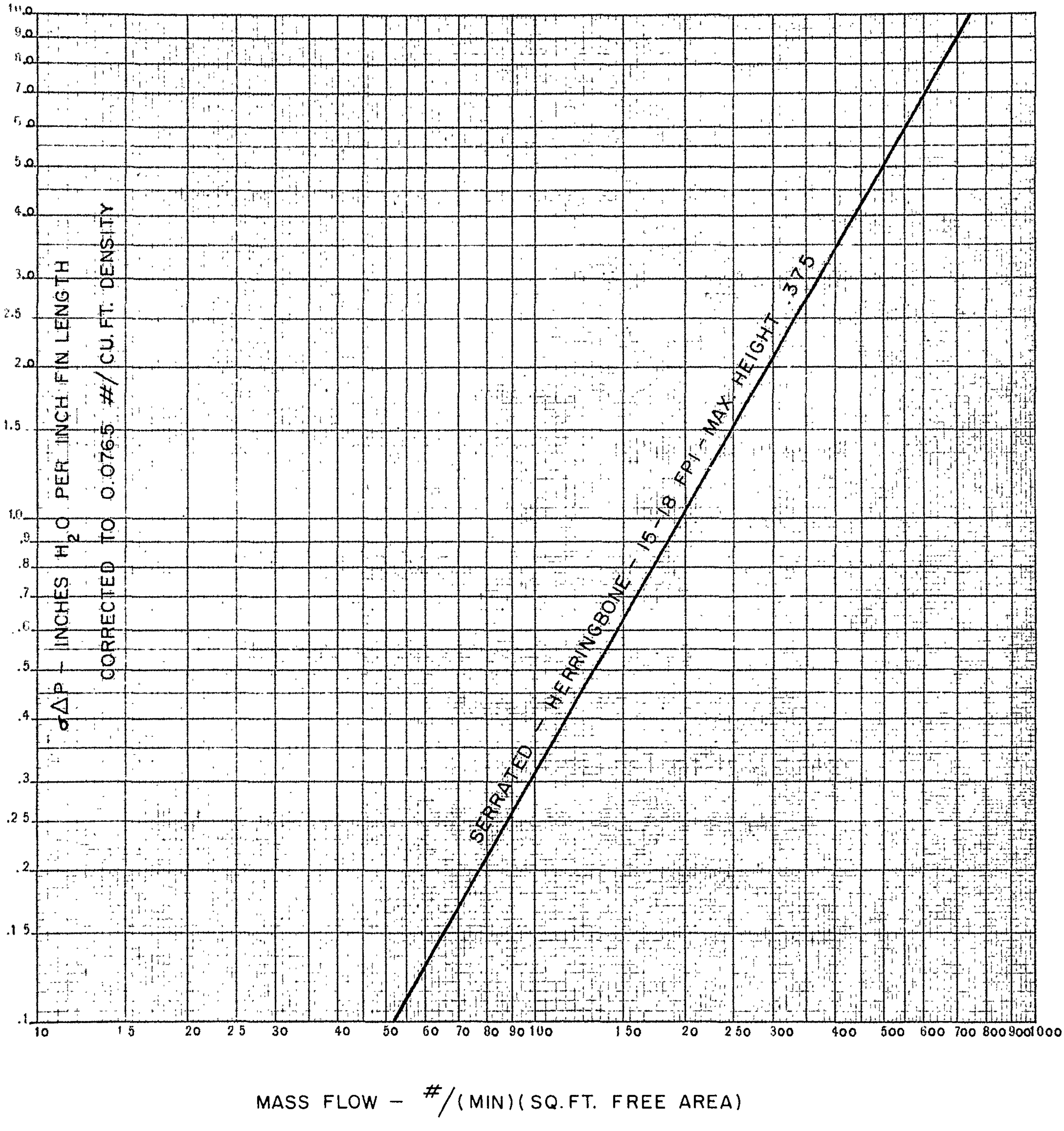


Figure 39. Intercooler Pressure Drop Data

$$T_{16a-17} = \frac{124.2 + 153.596}{2} = 138.9^{o}F = 598.9^{o}R$$

$$\Delta P_{16a-17} = \frac{8.88 \times 598.9}{236 \times 30.816} = 0.7313 \text{ in. Hg (core \& total drop)}$$

$$P_{17} = 30.816 - 0.7313 = 30.0847 \text{ in. Hga } \underline{1/}$$

(15) Hot air discharge pressure, temperature from first cooling pass:

Assume $T_{13} = 48.1^{o}F$

$T_{16} = -128.3^{o}F$

$P_{16} = 31.406$ in. Hga

From reference (8)

$$P_{s_{13}} = 0.33757 \text{ in. Hga}$$

$$W_{s_{13}} = \frac{4354(0.33757)}{293.763 - 0.33757} = 5 \text{ gr/lb dry air}$$

$$W_{c_2} = 92 - 5 = 87 \text{ gr/lb dry air (condensed)}$$

$$W_{c_{tot}} = 87 + 92 = 179 \text{ gr/lb } \underline{2/}$$

Heat transferred to cooling air:

$$Q = 14.5 \times 53 \left[141.3 - 48.1 + 0.6(87)\right] + \frac{184(53)(141.3 - 48.1)}{253}$$

$$Q = 115{,}481 \text{ BTU/hr}$$

Cooling air discharge temperature:

$$T_{16a} = T_{16} + \frac{Q}{14.5 \times w_{reg}}$$

1/ This value is slightly greater than atmospheric to include the pressure drop across the cabin pressure regulating control valve. Therefore, the assumed value for w_{reg} is correct (page 131).

2/ The values for $W_{c_{tot}}$, T_{16a} check the assumptions made on page 131.

$$T_{16a} = -128.3 + \frac{115,481}{(14.5)(31.5)} = 124.2^{o}F \ \underline{1}/$$

The effective mean temperature difference for cross flow exchangers is now calculated

$$\frac{T_{12a} - T_{13}}{T_{12a} - T_{16}} = \frac{141.3 - 48.1}{(141.3 - (-128.3))} = 0.3457$$

$$\frac{T_{16a} - T_{16}}{T_{12a} - T_{16}} = \frac{[124.2 - (-128.3)]}{[141.3 - (-128.3)]} = 0.937$$

Referring to Figure 26 (Page 102),

$$\frac{\Delta t_m}{T_{12a} - T_{16}} = 0.1793$$

$$\Delta t_m = (0.1793) \quad [(141.3) - (-128.3)] \quad = 48.34^{o}F$$

$$(UA)\ \text{reqd} = \frac{115,481}{48.34} = 2389\ \text{BTU/hr} \cdot {}^{o}F$$

Determination of $(UA)_{avail}$:

Hot side unit weight flow: $= \frac{53}{0.1644} = 322.4$ lb/min sq ft

Cooling side unit weight flow: $= \frac{31.5}{0.187} = 169$ lb/min sq ft

Referring to Figure 19, 27 (Pages 60 and 105),

Hot side film coefficient = 158 BTU/hr sq ft ^{o}F

Cold side film coefficient = 29.8 BTU/hr sq ft ^{o}F

$$1/UA = \frac{1}{(158)(88.2)} + \frac{1}{(29.8)(102.8)}$$

$$(UA)_{avail} = 2511\ \text{BTU/hr}\ {}^{o}F$$

$\underline{1}/$ The values for $W_{c_{tot}}$, T_{16a} check the assumptions made on Page 131.

Since $(UA)_{avail}$ reasonably agrees with $(UA)_{reqd}$, the value of T_{13} as assumed on Page 135 is correct.

The pressure drop through the exchanger is now determined;

Hot side:

Referring to Figure 39,

$$\frac{\sigma \Delta P_{12a-13}}{l} = 2.36 \text{ in. } H_2O/\text{in. length, pass}$$

$$\sigma \Delta P_{12a-13} = 2.36 \times 11.702 \times 1 = 27.62 \text{ in. } H_2O$$

$$T_{12a-13} = \frac{141.3 + 48.1}{2} = 94.7^{\circ}F = 554.7^{\circ}R$$

$$\Delta P_{12a-13} = \frac{27.62\ (554.7)}{236\ (293.8)} = 0.2210 \text{ in. Hg (core)}$$

Multiplying this figure by a factor of two, so as to include the additional drop from bends, elbows, and transitions,

$$\Delta P_{12a-13} = 2 \times 0.221 = 0.442 \text{ in. Hg}$$

$$P_{13} = 293.763 - 0.442 = 293.321 \text{ in. Hga}$$

Cold side:

Referring to Figure 39,

$$\frac{\sigma \Delta P_{16-16a}}{l} = 0.795 \text{ in. } H_2O/\text{in. length, pass}$$

$$\sigma \Delta P_{16-16a} = 12 \times 0.795 \times 1 = 9.55 \text{ in. } H_2O$$

$$T_{16-16a} = \frac{-128.3 + 124.2}{2} = -2.05^{\circ}F = 458^{\circ}R$$

$$\Delta P_{16-16a} = \frac{9.55 \times 458}{236 \times 31.406} = 0.59 \text{ in. Hg}$$

$$P_{16a} = 31.406 - 0.59 = 30.82 \text{ in. Hga } \underline{1/}$$

1/ The value of P_{16a} checks the assumption made on Page 131.

The moisture content of the hot air at inlet to the moisture separator consists of 5 gr/lb of saturated vapor and 179 gr/lb of condensed entrained moisture.

For a moisture separator efficiency of 81%, the total amount of moisture entering the turbine is:

Saturated vapor - $W_{s_{14}}$ = 5 gr/lb dry air

Entrained moisture $W_{w_{14}} = (1-.81)(179) = 34$ gr/lb dry air

Total moisture content = 39 gr/lb dry air

This value is far less than the prescribed maximum moisture content of 50 gr/lb.

(16) Hot air pressure at inlet to turbine:
Referring to Figure 40,

$$P_{13-14} = 4.62 \text{ in. Hga}$$

$$\rho_{13-14} = \frac{1.325\ (293.321)}{508.1} = 0.764 \text{ lb/ft}^3$$

Actual pressure drop across separator is

$$\Delta P_{13-14} = 4.62 \times \frac{0.0765}{0.764} = 0.4621 \text{ in. Hg}$$

$$P_{14} = 293.321 - 0.4621 = 292.86 \text{ in. Hga}$$

(17) Hot air discharge pressure from turbine:

Overall friction factor for turbine to cabin discharge ducting is

$$K = 0.0699\ \underline{1/} \qquad \text{(assumed)}$$

$$\sigma \Delta P_{15\text{ - cab}} = \frac{K\ (w_{cab})^2}{13.56}$$

$$T_{15} = 331.7^{\circ}R$$

1/ This factor includes the additional resistance of an added flow resistor.

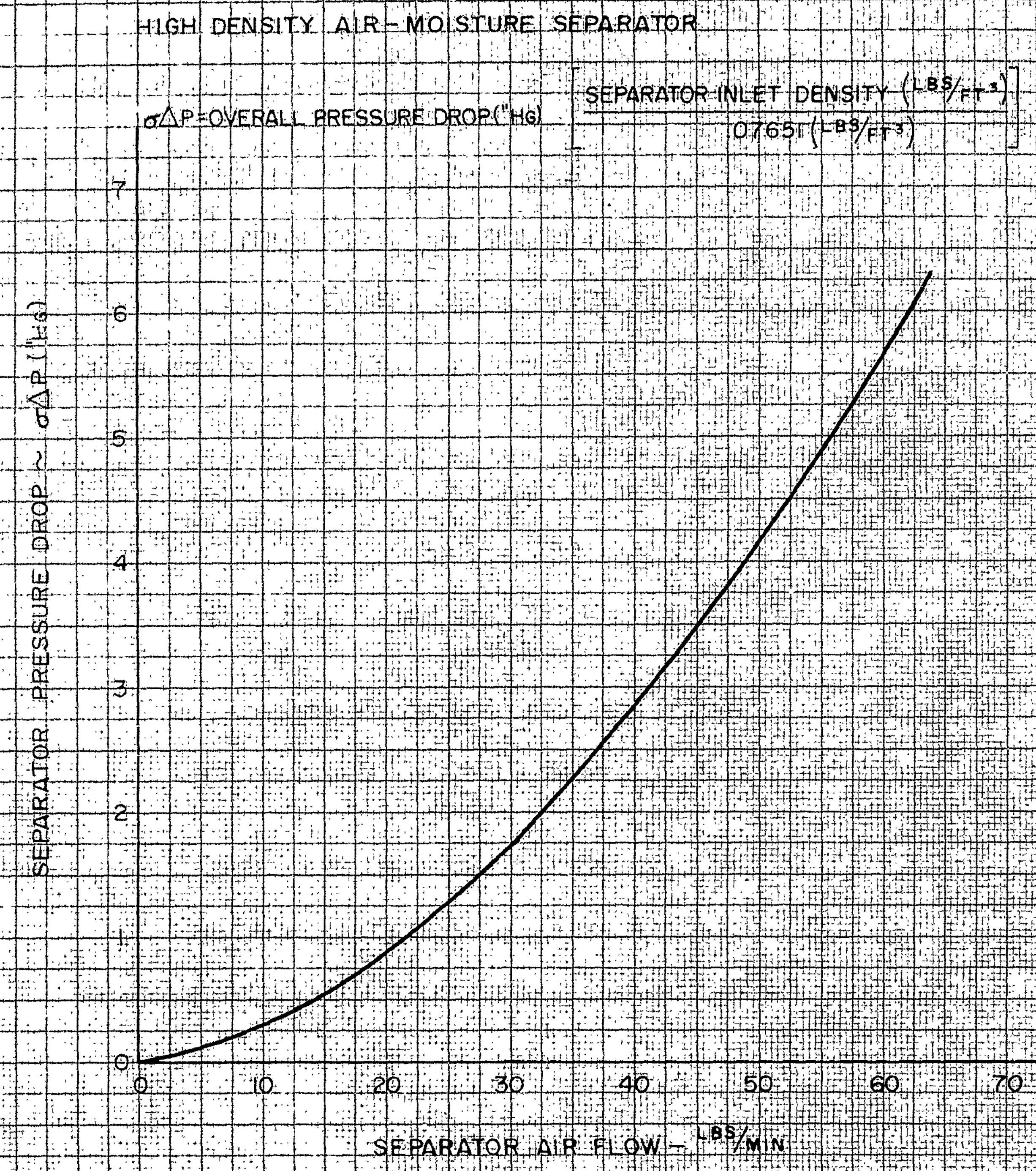


Figure 40. Estimated Maximum Separator Pressure Drop

Assume $\Delta P_{15\text{-cab}} = 1.486$ in. Hg

$$P_{15\text{-cab}\ avg} = 29.92 + \frac{1.486}{2} = 30.663 \text{ in. Hga}$$

$$\sigma = \frac{1.325 \times 30.663}{0.0765 \times 331.7} = 1.602$$

$$\Delta P_{15\text{-cab}} = \frac{0.0699\ [53 - 31.5]^2}{13.56 \times 1.602} = 1.486 \text{ in. Hg}$$

The assumed value of $\Delta P_{15\text{-cab}}$ is correct and $P_{15} = 29.92 + 1.486 = 31.406$ in. Hga (check) 1/

(18) Turbine and compressor power balance check:

$$\pi_t = \frac{P_{14}}{P_{15}} = \frac{292.86}{31.41} = 9.325$$

Adiabatic temperature drop:

$$\delta T_{14\text{-}15} = T_{14} \frac{Y}{1 + Y} = (508.1)(0.4684) = 238^{\circ}F$$

Turbine pitch line velocity:

$$C_t = \frac{\pi\ D_t\ N}{720} = \frac{\pi\ (3.8)(61,250)}{720} = 1015 \text{ fps}$$

Theoretical spouting velocity:

$$V = \sqrt{12,160\ \delta T_{14\text{-}15}} = \sqrt{12,160 \times 238} = 1701 \text{ fps}$$

Velocity ratio:

$$C_t/V = 1015/1701 = 0.597$$

Referring to Figure 11 (Page 37),

Turbine adiabatic efficiency is

$$\eta_t = 0.7425$$

1/ The value for P_{15} must equal the value for P_{16}, which was previously assumed on Page 135.

Actual turbine temperature drop is:

$$\Delta t_{14-15} = \eta_t \; (\delta T_{14-15}) = (0.7425)(238) = 176.7^{o}F$$

Turbine power output:

$$hp_t = 0.00566 \; w_b \quad \Delta T_{14-15} \; = 53.15 \text{ hp}$$

Compressor power input:

$$hp_c = 0.00566 \; w_b \; \Delta T_{10-11} \; = 50.33 \text{ hp}$$

Horsepower available for bearing, windage losses, and for oil pumping is

$$hp = 53.19 - 50.33 = 2.86 \text{ hp}$$

Mechanical efficiency is:

$$\eta_{mech} = \frac{50.33}{53.19} = 0.9462$$

The value for η_{mech} is a function of many indeterminate variables and, therefore, cannot be arbitrarily established. From this writer's experience, air cycle machines operate with a mechanical efficiency of 95% (approximately). Therefore the operating point of the compressor as assumed on Page 127 is correct.

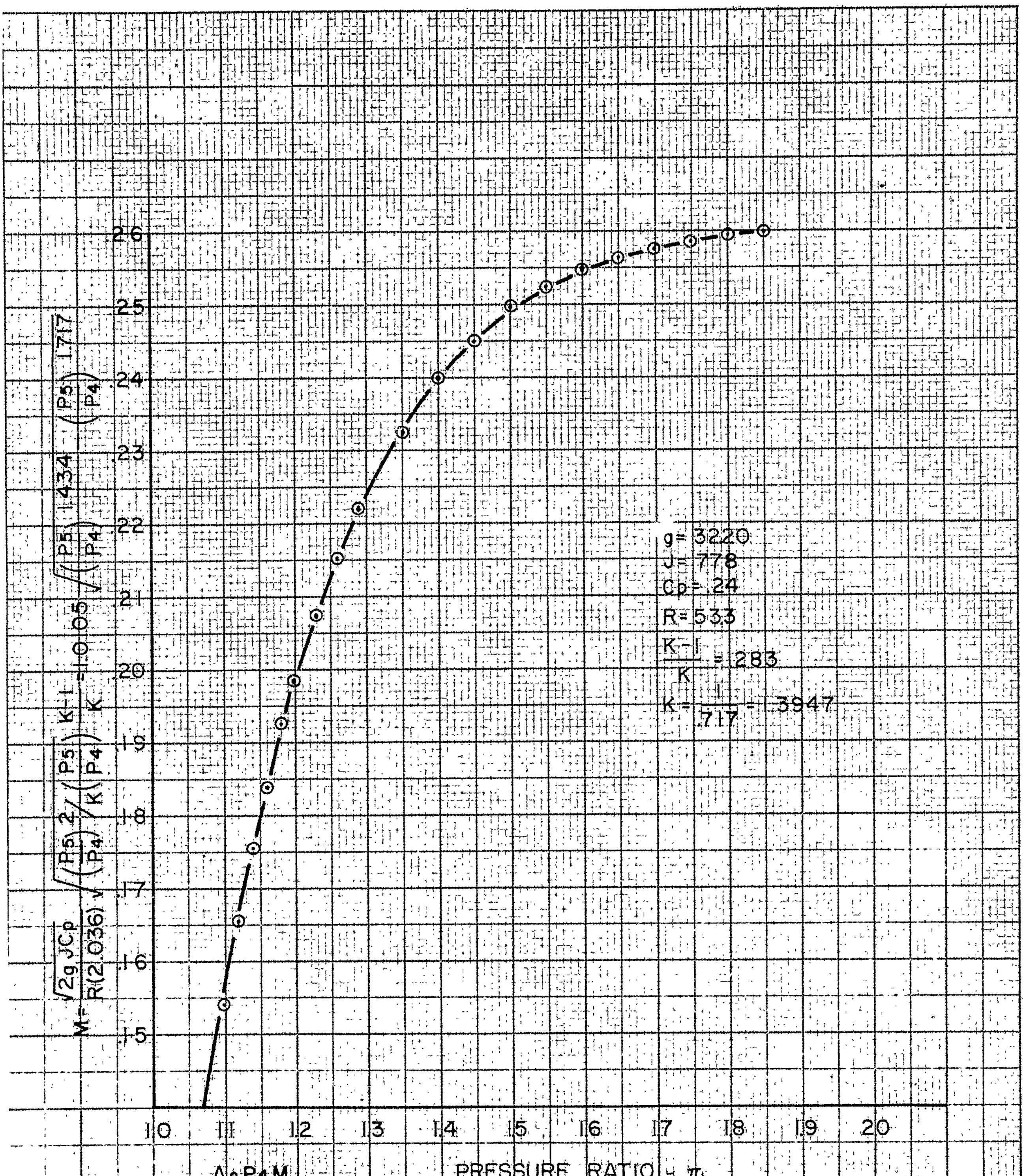
The flow through the turbine is now determined by the following equation:

$$w_b = \frac{60 \; P_{14} \; C_d \; A_n \; M}{\sqrt{T_{14}}}$$

Where M = restriction factor (see Figure 41).
A_n = nozzle area - sq in.
C_d = coefficient of discharge.

For C_d = 0.97 (assumed)
A_n = 0.2677 sq in.
M = 0.26

$$w_b = \frac{(292.859)(0.97)(0.2677)(0.26)(60)}{\sqrt{508.1}} = 53 \text{ lb/min}$$



$W = \dfrac{A_e P_4 M}{\sqrt{T_4}}$

W = WEIGHT FLOW − #/SEC.

A_e = EFFECTIVE NOZZLE AREA − IN^2

P_4 = PRESSURE (IN Hg) AT TURBINE ENTRANCE − T_4 CORRESPONDING TEMP. °R

M IS READ FROM GRAPH

Figure 41. Weight Flow VS Pressure Drop Pure Impulse Turbine

This value agrees with the value for engine bleed flow assumed initially on page 135 .

(19) Turbine exit temperature (dry air basis):

$$T_{15} = T_{14} - \Delta t_{14-15} = 48.1 - 176.7 = -128.6^{o}F$$

Turbine exit temperature must agree with the assumed regenerative cooling air inlet temperature at point 16 (page 135).

$$T_{16} \text{ (assumed) } = -128.3^{o}F$$

Corrected turbine exit temperature 1/

Referring to Figure 2, 2/ (page 4),

$$T_{15_c} = T_{15} + 0.726\,(W_{c_{15}}) = -128.6 + 0.726(5) = -125^{o}F$$

(20) Cooling capacity:

Design cabin temperature, $T_{cab} = 80^{o}F$ (see Figure 42).

Since all entrained moisture at the cabin inlet re-evaporates in the cabin itself, the cooling capacity is therefore,

$$Q = 14.5\,(w_{cab}) \left[T_{cab} - T_{15_c} + 0.726\,(W_w)\right]$$

$$Q = 14.5 \times 21.5 \left[80-(-125) + 0.726(39)\right] = 72{,}730 \text{ BTU/hr}$$

Required cooling capacity:

Q = 65,000 BTU/hr (see Figure 1).

Excess cooling capacity:

$$\frac{72{,}730 - 65{,}000}{65{,}000} \times 100 = 11.89\%$$

1/ The corrected turbine discharge temperature includes the heating effect of the moisture condensed during expansion. Actually, subsequent reevaporation cooling of this condensate in the cabin nullifies this effect.

2/ The curves in Fig. 2 do not encompass the range of temperatures presently encountered. It is recommended that the appropriate equation listed therein be used.

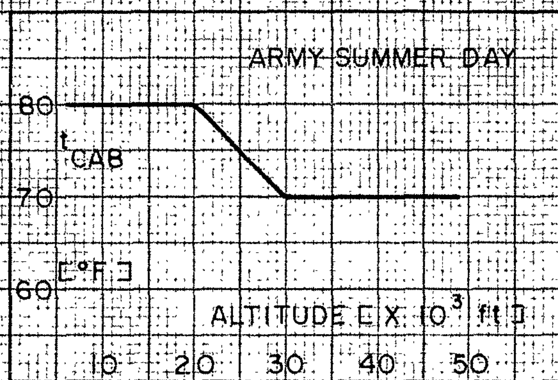


Figure 42 Cabin Temperature as a Function of Altitude