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FEASIBILITY STUDY OF A SMALL THERMAL CHAMBER FOR PROGRAMMED TRANSIENT INCREASES AND DECREASES IN TEMPERATURE

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[Prepared under Contract No. AF 33(616)-6763
by
Jeremy Crocker
John Lyman
University of California, Los Angeles, California]

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FOREWORD

This report was prepared under Contract No. AF 33(616)-6763, in support of Project No. 7222, "Biophysics of Flight," Task No. 722204, "Human Thermal Stress." The study was initiated by the Biomedical Laboratory, 6570th Aerospace Medical Research Laboratories, Aerospace Medical Division, Air Force Systems Command, Wright-Patterson Air Force Base, Ohio. William C. Kaufman, Major, USAF, of the Biothermal Branch, Physiology Division, Biomedical Laboratory, served as contract monitor. The study was made in the Biotechnology Laboratory of the Department of Engineering, University of California, Los Angeles, by Jeremy Crocker, Dr. John Lyman, and Robert Maddock. L.M.K. Boelter is Dean of the Department of Engineering and P.F. O'Brien acts as his representative for research activities. Dr. Lyman is the project leader for research under the above contract.

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ABSTRACT

After a brief review of experimental requirements and practical constraints, performance specifications of a rate of change of 200° F per minute and a maximum temperature of 600° F were arbitrarily set for a possible thermal chamber facility. Various methods of heating were examined and the decision was made to make a special study of a fluid wall thermal chamber as a promising approach to achieve the desired specifications. It was found that it should be possible to construct a circulating fluid wall chamber, using commercially available silicone derivative fluids for under $40 per square foot of area, plus a fixed cost for controlling equipment. Based on approximate figures, it was concluded that an 8-foot chamber, meeting the desired specifications and with a capability for extending its operation into a thermal range below room temperature, could be constructed for under $50,000.

PUBLICATION REVIEW

This technical documentary report has been reviewed and is approved.

Jos M Quashnock
JOS. M. QUASHNOCK
Colonel, USAF, MC
Chief, Biomedical Laboratory

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I. INTRODUCTION

Temperatures of the principal glandular and nervous organs of the human body are normally maintained constant within less than 2°F in spite of large variations in temperature of the external environment and of the energy released within the body. Knowledge of the heat regulatory mechanisms has progressed from simple identification of the principal pathways of heat loss to attempts to analyze transient measurements. These results have indicated the time dependent behavior of internal control mechanisms when the body is subjected to various environmental forcing functions. Although heat losses necessary to maintain the equilibrium have been correlated in a simple heat balance equation, transient physiological changes remain unsystematized.

Current interest in exploration of the more hostile regions of the earth, its atmosphere, and outer space has led to increased interest in the responses of the human organism to extremes of environmental temperature and a more precise definition of the limits of survival.

The following design study represents an attempt to create a thermal chamber as an experimental aid providing precisely programmed changes in the environment. Such a chamber would help to extend general understanding of the nature of the human temperature control mechanisms and to delineate the human tolerance limits to extreme environmental temperatures.

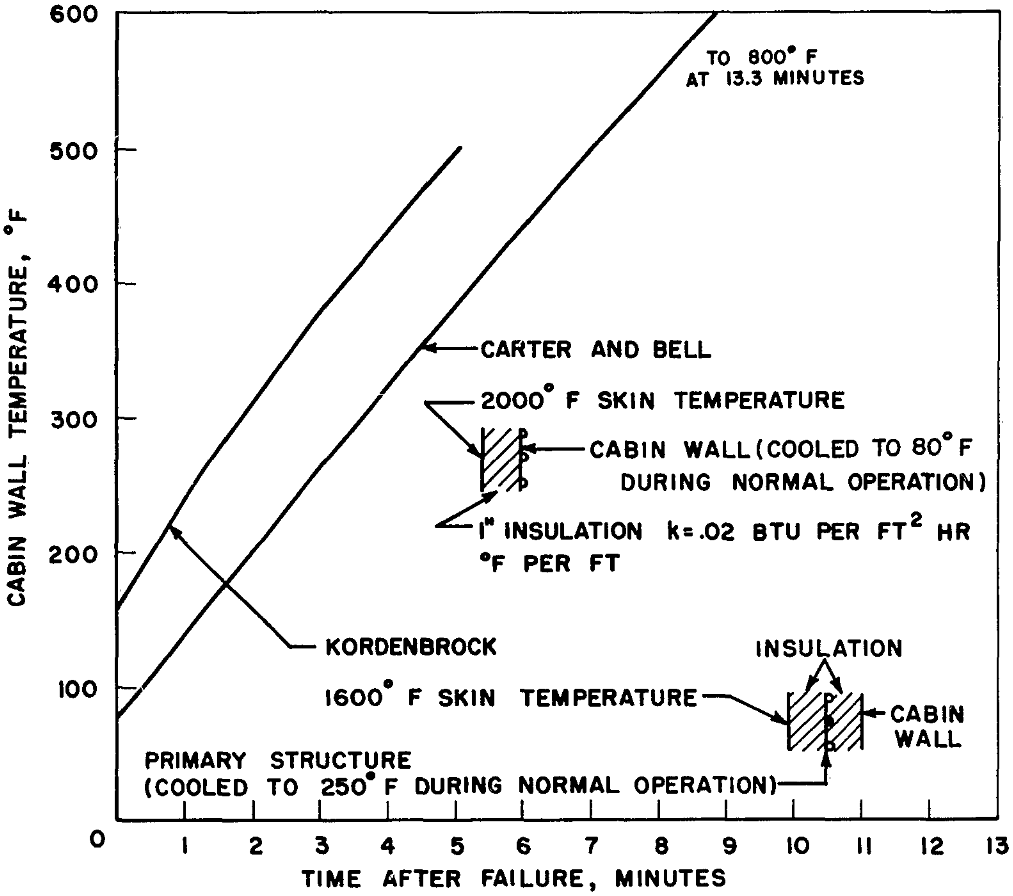
A truly efficient design, one in which the required performance is achieved at minimum expense, depends fundamentally upon the initial decisions specifying its performance. Inclusion of extraneous performance results in additional cost at every subsequent step. Failure to cover adequate capability would ultimately be uneconomic, also. For this reason, the first step in this study is devoted to a description of the experiments to be conducted in the proposed chamber in an attempt to establish a rationale for an optimum design.

II. DERIVATION OF PRELIMINARY PERFORMANCE SPECIFICATIONS FROM THE RANGE OF EXPERIMENTS TO BE CONDUCTED IN THE CHAMBER

A. Performance Characteristics of Existing Vehicles

The high and low body surface temperatures at which the control mechanisms of the human body are fully exercised are small compared to the extremes of environmental temperature which can reasonably be expected to occur during flight at high speeds or when stationary in extra-terrestrial locations, such as the moon. For this reason, the first factor to be considered is the range of temperatures required to simulate realistically extreme conditions.

Transient high temperatures have long been predicted to occur within a vehicle entering the earth's atmosphere from orbit. Kordenbrock,[12] for example, assumed a vehicle configuration in which heat penetrating through a layer of insulation from the outer skin is blocked by vaporization of water in channels in the primary, load-bearing structure. Only a fraction of the heat would be driven through a second layer of insulation by the gradient between the primary structure at 250°F and the cabin wall at 160°F. This remaining heat is removed by air circulated through the cabin heat exchanger. The failure shown in Figure 1 would follow a loss of water from the channels of the primary structure, the inner wall temperature rising to 500°F at an average rate of about 70°F per minute. Carter and Bell(6) assume a configuration in which the vehicle skin serves also as the load bearing structure. The cabin wall is cooled to 80°F by vaporization of water in channels, providing a comfortable environment for the crew without additional cabin cooling.



RISE IN CABIN TEMPERATURE DURING COOLING SYSTEM FAILURE

FIGURE 1

Failure in the Carter and Bell case again is assumed for a loss of fluid in the cooled structure. An emergency re-entry along a trajectory of maximum permissible vehicle skin temperature is begun immediately, requiring 13.3 minutes to reach an altitude and speed at which cool outside air can be inducted for cooling. Wall temperature reaches 500°F in 7 minutes at an average rate of 60°F per minute, continuing to 800°F at the end of re-entry.

Bishop, Schumacher and Bloetscher[1] assume an escape capsule with a structure similar to Kordenbrock's, re-entering on a ballistic trajectory. However, since this is the normal case for escape, no failure is assumed and the cooled wall remains intact. This condition is illustrated in Figure 2.

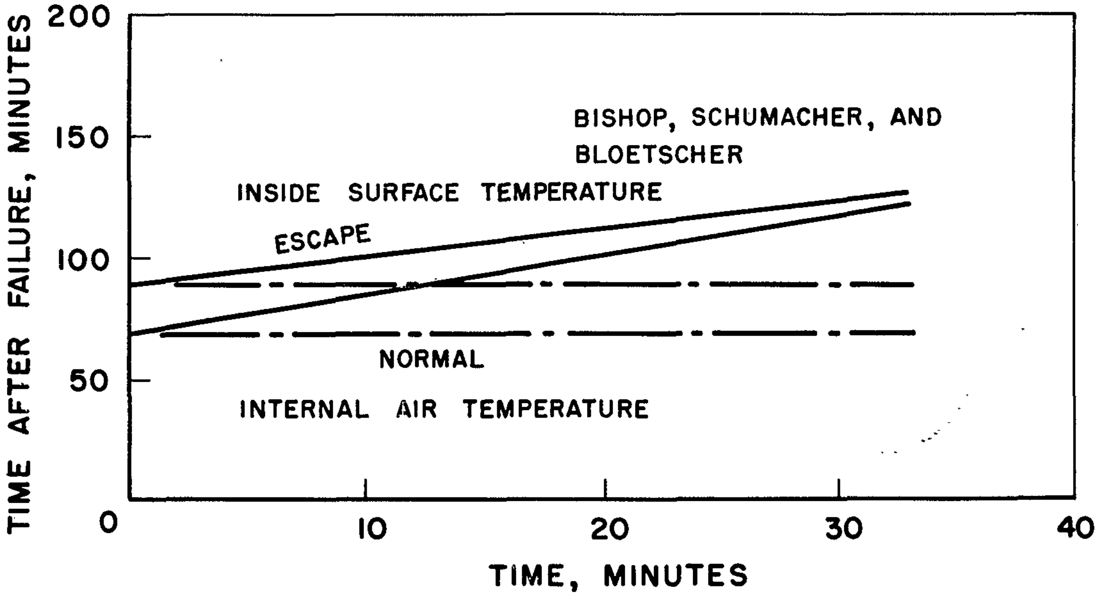


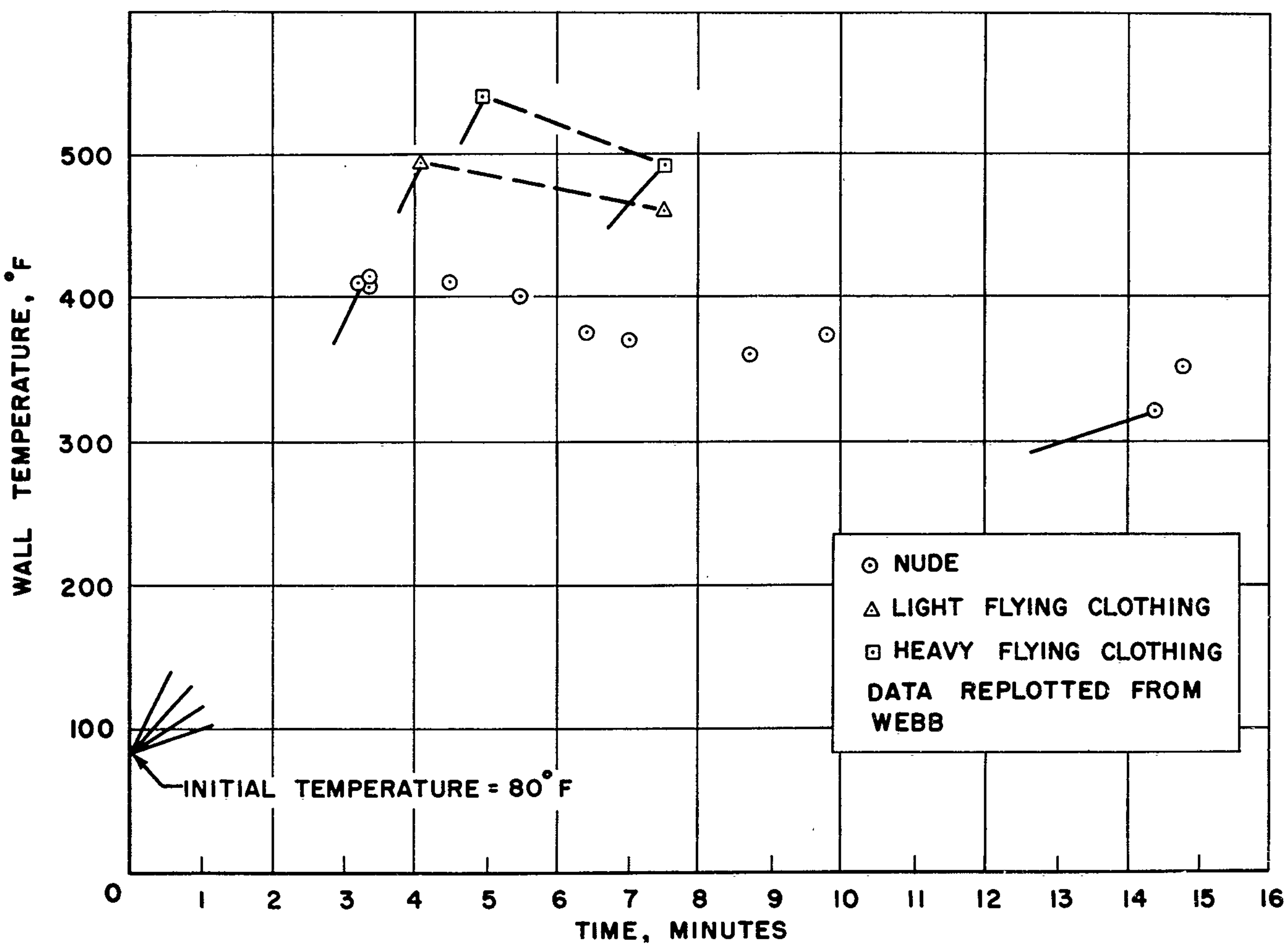
FIGURE 2

The possible extreme conditions outlined in the foregoing paragraphs should be considered in relation to presently known facts of human tolerance to such temperatures. This comparison may be expected to provide information concerning ranges and rates of change of temperature which will yield practical information.

B. <u>Human Tolerance Limits</u>

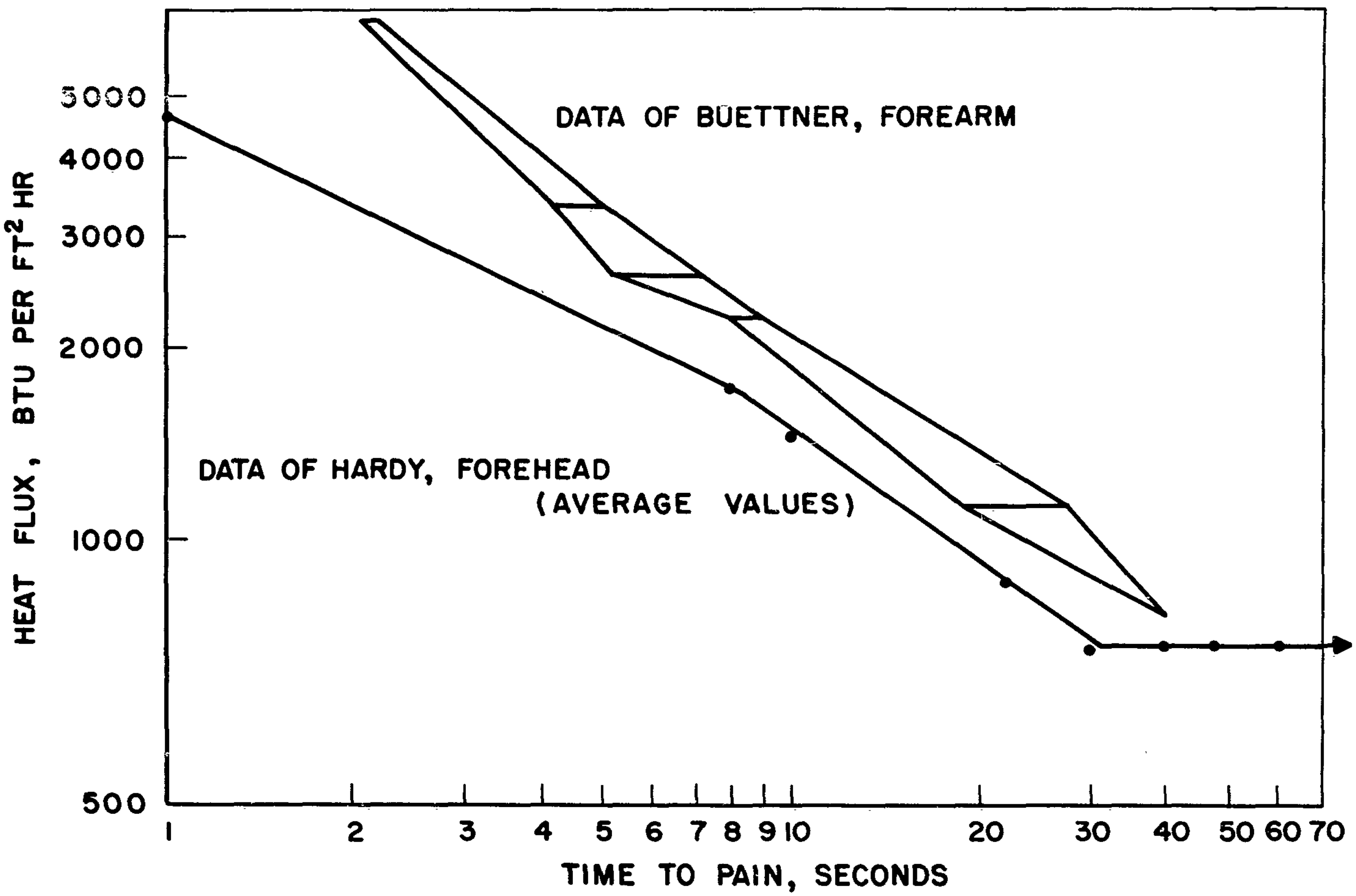
Webb[21] experimentally defined the approximate temperature limits for human subjects exposed to ramp functions of wall temperature between 100 and 160°F/min. These tolerance limits seem to be dependent upon the amount of clothing insulation and the rate of change of temperature. From the results of his work, summarized in Figure 3, it can be seen that further experiments to explore the protective effects of clothing for slower rates of temperature rise are indicated. Also, investigations of both nude and clothed tolerance at very high rates of change should probably be undertaken for small areas of skin.

Earlier work by Hardy [9] and Buettner[5] established pain thresholds for specified temperatures and the time required at a given flux rate to reach this temperature. Values extracted from these papers are presented in Figure 4.



HUMAN TOLERANCE LIMITS TO TRANSIENT HEATING

FIGURE 3



THRESHOLD FOR TEMPERATURE-INDUCED PAIN

FIGURE 4

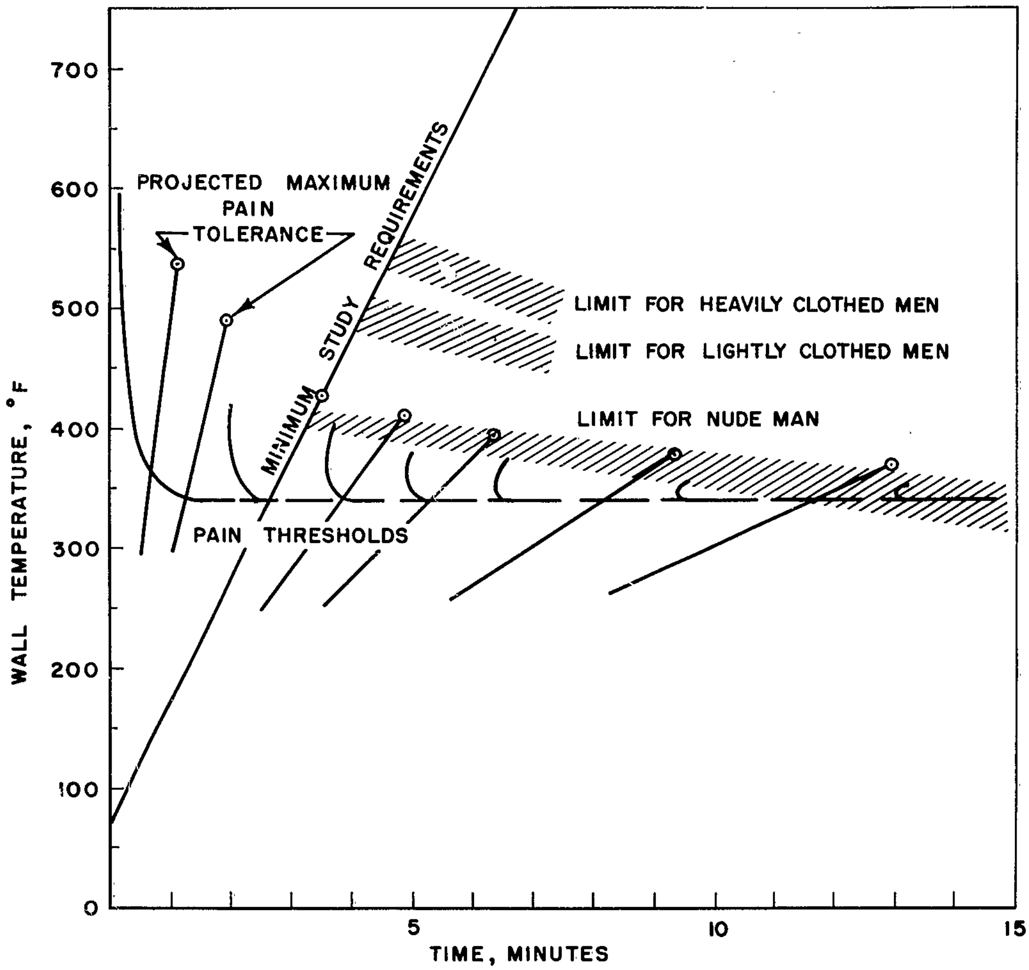
In both experimental series, time-to-pain was a definable relationship with some variation between subjects for flux rates above about 700 BTU per ft^2 hr (2000 Kcal per m^2 hr). Large random variations were found for flux rates below this range, and when the flux rate was lowered further, pain did not occur. Apparently increase in local circulation was sufficient to remove the heat. Human tolerance and performance at low rates of heating are reported by Blockley, McCutchan, and Taylor.(2) Although derived from temperatures below the maximum to be considered here, the time limits at various heating rates can be applied at higher temperatures when insulation is correspondingly increased. The amount of heat removed by ventilation can be computed by the equation given by Skilling, McCutchan and Taylor(19) -- subject to their assumptions -- in particular the insulation of exposed skin (face, hands, and feet).

With this knowledge of human response to high temperatures, it seems possible to decide how large a temperature range is meaningful, and where a limit should be set beyond which higher temperatures would not yield much additional information.

C. Chamber Performance Characteristics

Specifications for this study were informally outlined with Biomedical Laboratory personnel. A rate of temperature rise of 100°F per minute to a 750°F maximum temperature was suggested, and these study requirements form the basis of Figure 5. Superimposed are the Webb tolerance data which define nude tolerance to linear rises in temperature at rates of rise below 100°F per minute, and clothed tolerance for rates between 50 and 100°F per minute. Plotted also in Figure 5 is a pain threshold curve derived from the Buettner and Hardy data as follows: For a value of absolute temperature (T) on the ordinate, the emissive power (σT^4) was calculated, and the time-to-pain corresponding to a heat flux of that intensity was obtained from Figure 4 and plotted in Figure 5. As a result, this curve represents approximately the time at which a nude individual exposed suddenly to radiation from surroundings at a given temperature will feel pain in his skin.

Although this curve was derived for sudden exposures, several conclusions can be extrapolated for exposures produced by a linear rise in temperature. At temperatures below 340°F, which is represented by the horizontal portion of the curve, no pain will be felt. Once this temperature is exceeded, pain will be felt after a certain duration which will be governed by the hyperbolic segment of the curve along the vertical axis. The exact time relation is not known, although it can be expected not to exceed 1-1/2 minutes after crossing 340°F. One simplified method of treating this uncertainty is to pick a point upon a particular linear increase, and then, disregarding the effect of further increase, plot the time at which pain is expected to be felt. If this is done for a number of temperatures, an arc will be formed which possesses a minimum with respect to time. A vertical tangent to this minimum, intersecting the line of linear increase, then determines a point. Points obtained by this construction for various rates of increase are plotted in Figure 5, where they can be seen to form an upper limit to the experimental data in the range for which it exists. Points in the range above 100°F per minute indicate the maximum tolerance which might be expected.



CHAMBER STUDY REQUIREMENTS IN RELATION TO
THERMAL TOLERANCE LIMITS

FIGURE 5

In considering the limits plotted in Figure 5 for clothed men, it should be understood that although feet, hands, and head were protected, the face was unclothed, and its only protection was whatever shielding the subject could provide with his hands. If complete face protection and cool breathing air were provided, an even higher limit might be established. In particular, because of the thermal lag such clothing would provide, very high tolerance limits (1000°F) might be obtainable at high rates of rise (to perhaps 300°F per minute, for example).

Because of demands on power and materials, cost of a heat chamber will increase with the maximum temperature to be reached and with the rate of change to be achieved. Since change and range are interrelated when probing tolerance limits, cost mounts very rapidly. To set a bound to this, a design maximum of 200°F per minute and 600°F has been more or less arbitrarily taken for the present study. It deviates from the suggested range of 750°F limit at 100°F per minute, but should cover a useful practical range at a somewhat lower initial cost. This rate of change is adequate to simulate temperature changes in a space vehicle three times as rapid as presently foreseen, and the maximum temperature includes a range in which the clothing configuration will be an important variable, but not so high that it is the only one.

On the basis of these maximum conditions, feasible approaches to a design can now be selected.

III. FEASIBILITY STUDY OF VARIOUS METHODS OF HEATING

Important factors to be included in approaching a design are the available sources and sinks of heat, and the equipment necessary to supply and control this heat. In this section, certain cases are examined with respect to feasibility and cost.

A. Heat Sources and Sinks Available in Biotechnology Laboratory

Present heat sources and sinks available at a possible site for a chamber (Biotechnology Laboratory, Department of Engineering, UCLA) with the cost of each are shown in Table 1.

TABLE 1

COSTS OF SELECTED HEAT SOURCES AND SINKS

				Approximate Cost per Million BTU
1. GAS	2-inch line	Cost:	58¢ per 1000 cubic feet (34¢ if interruptable in winter)	.53 .31
		Heating Value:	1100 BTU per cubic foot	

TABLE 1 (Cont.)

COSTS OF SELECTED HEAT SOURCES AND SINKS

				Approximate Cost per Million BTU
2.	STEAM	2-1/2 inch line	Cost: 77¢ per 1000 pounds	.59
			Pressure: 30 to 35 PSIG	
			Temperature: 250° to 315°F	
			Heating Value: Around 1300 BTU per pound	
3.	ELECTRICITY		Cost: 1¢ per kilowatt hr., with 1¢ penalty to University if load exceeds approx. 100 kw maximum during peak hours (10:15 to 11 AM, 1:45 to 3 PM)	2.94
			Voltage: 220 VAC three phase delta connected	
4.	COMPRESSED AIR	1-1/2 inch line	Cost: Less than 0.1¢ per pound	
			Pressure: 100 PSIG nominal	
5.	COOLING WATER		Cost: 11¢ per 100 cubic feet (industrial water)	
		Used in liquid state (50°F temperature rise).		.35*
		Vaporized.		.017*

*Because of the large difference in cost between the two methods, a package unit cooling tower might be justified for a large, sustained load. This could be placed on the roof.

B. Means of Applying Heat Directly to Chamber Walls

Some readily available means of applying heat directly to chamber walls, including the costs and limiting temperatures, are shown in Table 2.

TABLE 2

COST AND POWER REQUIREMENTS OF SELECTED MEANS FOR APPLYING HEAT DIRECTLY TO CHAMBER WALLS

	Cost per sq. ft.	Max. Wall Temp.	Limit Watt/cm^2	Power Watt/in^2	Density BTU/ft^2hr
1. Conductor bonded to panel surface					
a. Electrofilm, Inc. "Electromesh"	$ 8.00	500°F	4.65	30	14,700
b. Electrofilm, Inc. "Electrofilm"	$10.00	340°F	7.75	50	24,400
c. Napier-PacAero "Spraymat"	$10.00	250°F	6.20	40	19,600
2. Conductor in flexible blanket					
a. Electrofilm, Inc.	$40.00	200°F	6.20	40	19,600
3. Radiant heating					
a. G.E. type T-3 lamps	$21.00 ($7 per linear foot of lamp, 4 inch lateral spacing)	2400°F	51.7	333	163,000

The temperature limits of the bonded conductors are due to the limits of the bonding materials themselves rather than the conductors. When a ceramic fiber is used, as in the blanket, very high temperatures can be attained.

Maximum temperature of a radiantly heated panel would be limited by the softening point if a non-ferrous material was used for the walls. With stainless steel, temperatures of 1400°F and above could be reached if certain parts of the lamps themselves are cooled.

The possibility of using gas to directly heat the chamber walls also exists, but the difficulty of control and non-uniformity of temperature, particularly on vertical walls, seriously offset its lower cost.

The maximum steam temperature of about 300°F makes heating with this medium unfeasible in the present application.

C. Means of Indirect Application of Heat

In general, any of the sources of energy listed above could be used to heat a secondary fluid which is circulated and in turn transfers its heat to the chamber walls. Such an approach would seem to lead to some excellent possibilities for rapid and closely controlled temperature change rates and limits. Consequently, it was decided to study this possibility in more detail.

IV. FLUID-WALL THERMAL CHAMBER DESIGN ASSESSMENT*

As stated elsewhere, the design objective for the thermal chamber was arbitrarily set to be able to adjust the heat exchanger wall panel temperature at a maximum rate of 200°F per minute up to a temperature of 600°F. From a performance standpoint, it would also be desirable for the rate of temperature change to be as uniform as possible over the entire surface of the thermal panel. To aid in following the rationale of the subsequent sections, a glossary of symbols, units, and relevant supporting equations is appended after the list of references.

Suppose a fluid is allowed to pass through a heat exchanger wall panel, as shown in Figure 6. The instantaneous panel temperature would rise very rapidly near the high temperature fluid inlet and proportionally less as the fluid temperature drops near the outlet. The reason for this is that the quantity of heat transferred per unit area per unit time, dq/dA, is directly proportional to the temperature difference or potential between the fluid and the panel.

The temperature differential across the panel is, of course, only transient, and the panel in time would approach temperature stabilization. The rate of stabilization would be a function of panel sizes, configuration, and thermal diffusivity, K/Cp. Because the nature of the proposed thermal tests is transient, the following panel design approach is suggested to alleviate the transient uneven panel temperature distribution.

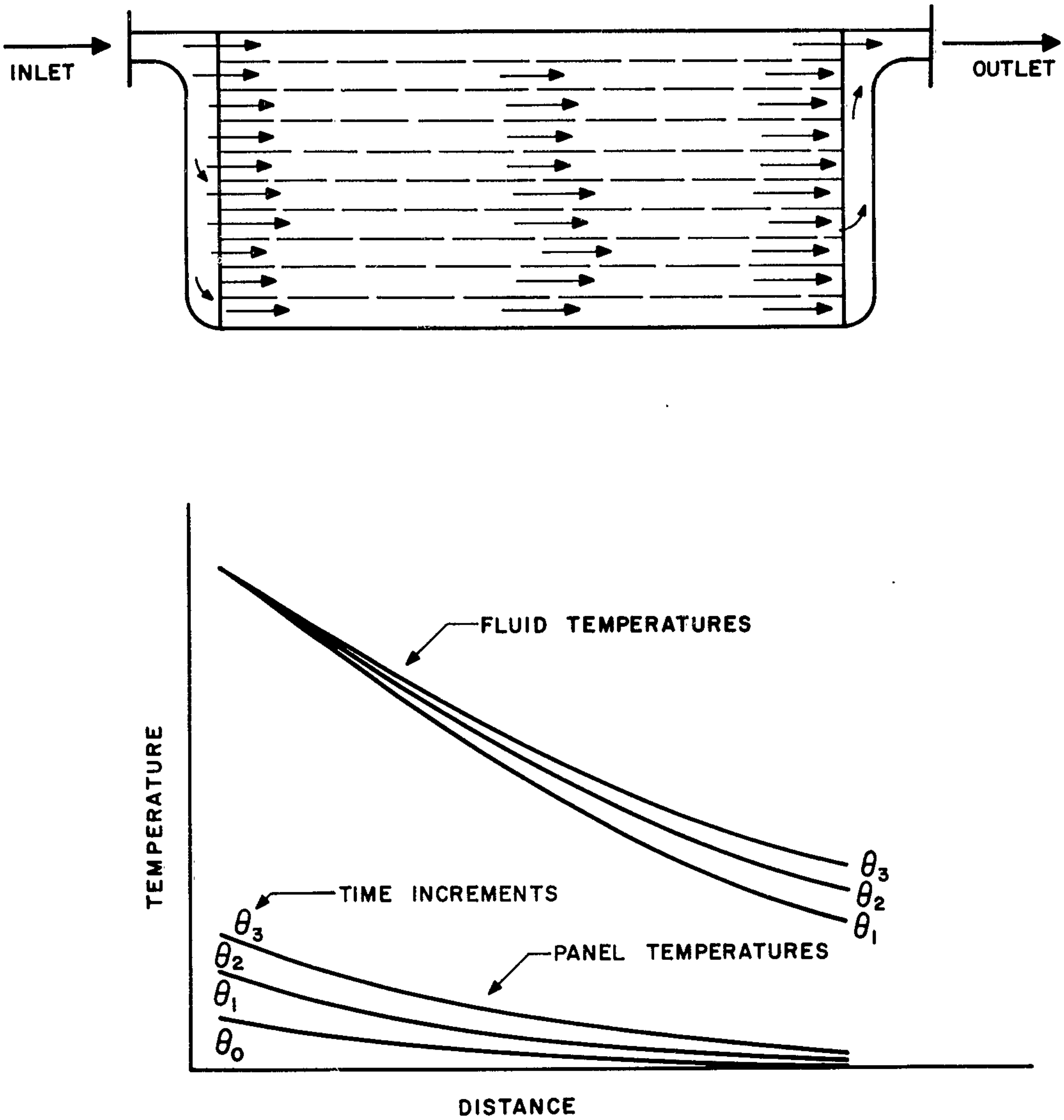
A. Uniform Temperature Thermal Panel

An essentially uniform temperature may be obtained by reversing the fluid flow alternately across the panel. This may be accomplished by two different methods as shown in Figures 7 and 8.

In the first method, fluid runs through alternate channels the length of a half panel and is returned in the opposite direction through alternate channels immediately adjacent to its prior direction. In this design, inlets and outlets are on the same side of the panel.

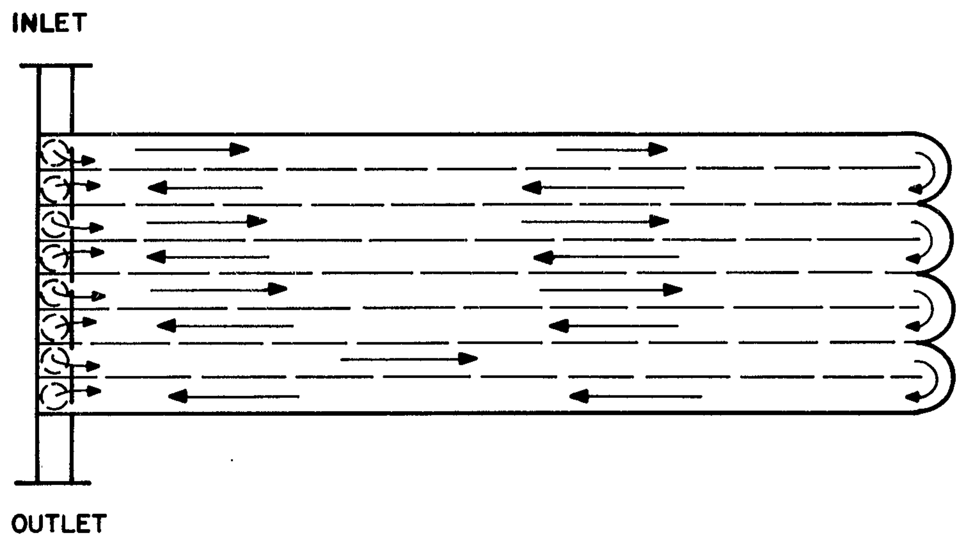
The rapid change in direction of the fluid may cause an excessive pressure drop. Therefore, a second approach is recommended. In this approach,

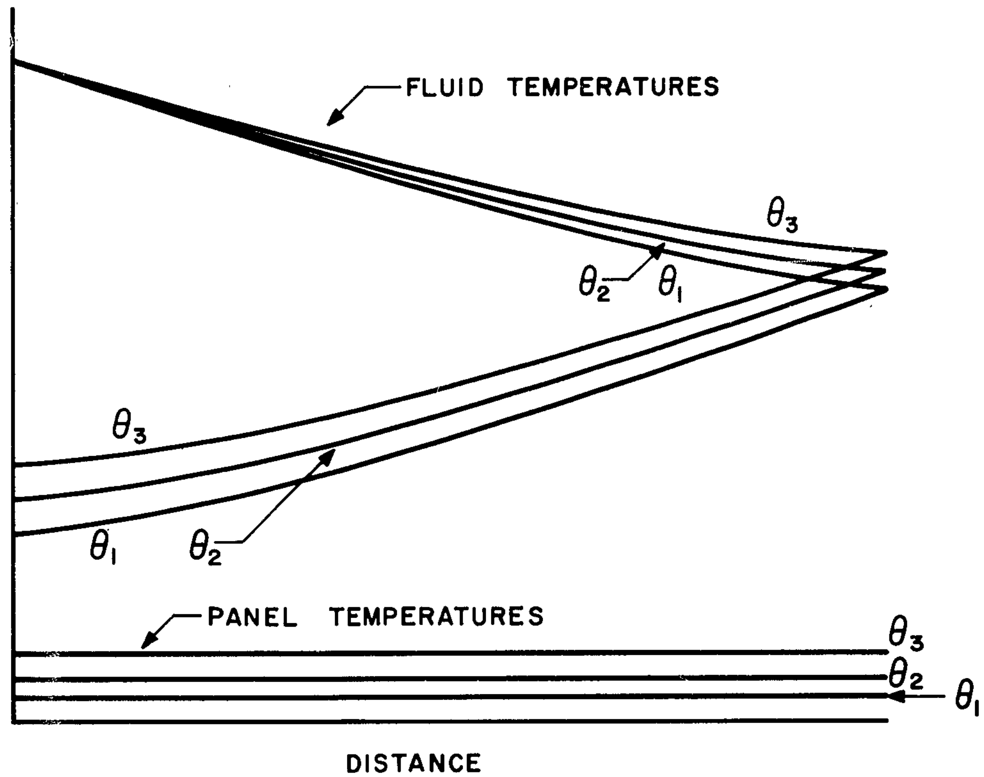
*The content of Sections IV and V was prepared principally by Mr. R.W. Maddock.



CONVENTIONAL PANEL DESIGN AND TEMPERATURE RELATIONS
WITH FLUID FLOW DISTANCE

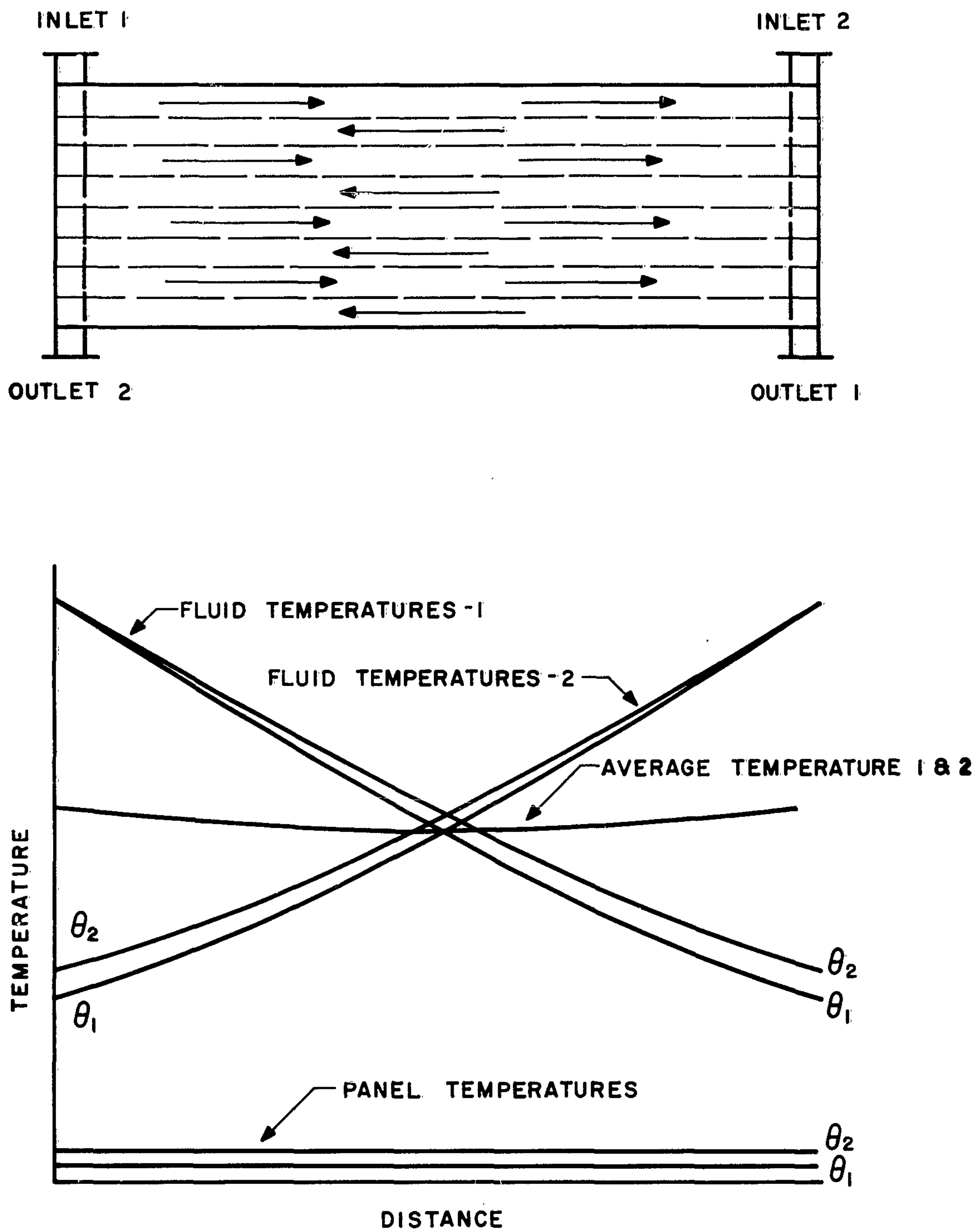
FIGURE 6





FIRST METHOD TO OBTAIN UNIFORM PANEL TEMPERATURE

FIGURE 7



SECOND METHOD TO OBTAIN UNIFORM PANEL TEMPERATURE

FIGURE 8

the fluid inlet is introduced through alternate channels from both ends of the panel and collected from both ends of the panel.

The theory is that heat flow will be average at any point along the panel since highest temperature potentials are added to lowest temperature potentials for any two adjacent channels. Figure 8 illustrates approximately how this averaging is accomplished. It is recognized that there will be a loss in heat exchanger effectiveness because some heat will bypass back into the outlet stream. The degree to which this happens will depend both on the panel configuration and the thermal diffusivity of the panel material. It is believed that this relatively small loss in efficiency is well worth the great advantage in having a thermal chamber with an even panel temperature distribution.

B. Power Requirement

The total power requirement for the fluid-wall thermal chamber may be divided into that derived from the natural gas requirement for fluid heating and from the electricity for fluid-pumping. The required values, especially that of fluid pumping power, depends on both the thermal chamber fluid-wall panel design configuration and the choice of heat-transfer fluid.

The heat-transfer coefficients of fluids in laminar flow are usually quite small because they (with the exception of liquid metals) have relatively small thermal conductivities. As the flow rate increases beyond a critical velocity range, laminar flow can no longer be maintained and turbulence begins. As turbulent flow begins, there is a sharp rise in the heat-transfer coefficient. The reason for this is that heat-transfer no longer is accomplished by conduction alone, and the cross currents and eddies rapidly mix temperature difference in the main body of the stream. The temperature of a fluid in turbulent flow through any given cross section will be essentially constant with the exception of a slow-moving laminar region next to an almost stagnant film on the heat-transfer surface.

The turbulent flow heat-transfer coefficient may further be increased by increasing the flow rate, thus increasing the turbulence of the main stream. The increased turbulent eddies penetrate deeper into the laminar layer, thus reducing the thermal resistance of the film. This, however, is accomplished only by larger energy losses and increasing frictional pressure drop.

Inspection of the Dittus and Boelter heat-transfer coefficient equation (see page 34) shows that in the magnitude of the design of the fluid wall panel two factors will be in control of the design. These are the velocity of the fluid and the diameter or hydraulic diameter of the channels. By increasing the velocity of the fluid, an increase in the heat-transfer film conductance proportional to $V^{0.8}$ is achieved.

An increase in hydraulic diameter will reduce the heat-transfer film conductance by a ratio of $1/D^{0.2}$. Where the flow rate remains constant, an increase in the cross sectional area of the channels will further lower the film conductance.[3] Therefore, it is seen that use of small cross-sectional areas are conducive to high rates of heat-transfer, but not without penalties, since small cross-sectional areas call for increased pumping power to overcome increased frictional resistance.

The designer is then confronted with trade-offs between the gain in heat-transfer coefficient with a reduction in the size and initial cost of equipment, and a larger pump, with increased operating and pumping-power costs. The additional operating costs for high-velocity equipment often outweigh the initial savings.

Commercial heat-exchangers generally have velocities which correspond to a Reynolds number of no more than 50,000. Laminar flow, because of its very low heat-transfer conductance, is generally avoided.[13]

C. Heat-Transfer Fluid Evaluation

The most essential requirement of any heat-transfer fluid for the fluid-wall thermal chamber is high temperature (600°F +) operational performance capability.

In order to make a comparative evaluation between the most suitable heat-transfer fluids, certain physical and chemical properties of the fluids must be known. The most important of these are boiling point, viscosity, specific heat, thermal conductivity, density, vapor pressure, flash point, chemical stability, and compatability.

Whenever possible, such information should be secured from experimental data. These data are available from fluid manufacturers, processors, trade publications, and scientific literature.[4,7,8,15-18,20,22-25] When such data are nonexistent, empirical and theoretical methods can be used to estimate some of these physical properties. C.J. Geankoplis et al[8] gives a rather complete discussion on the merits of various methods of estimating these values. In addition to purely physical and chemical properties of fluids as they affect heat-transfer and pumping cost, items such as fluid cost per gallon, personnel safety, reliability, dependability, and physiological properties should also be considered carefully before final selection of the best fluid for the thermal chamber.

Some of these fluid properties bear more careful consideration in the comparative evaluation of the most promising fluids. These properties are discussed below.

D. Vapor Pressure, Flash and Fire Points, Density and Specific Heat

Because of the relatively high operating temperatures, vapor pressure is one of the most important design considerations.

Vapor pressure is primarily a measure of the molecule escape-tendency in a liquid expressed in terms of pressure. This escape tendency depends on the translational kinetic energy. The higher the temperature, the higher the kinetic energy of each molecule. This gives molecules an increased ability to overcome the forces tending to draw them together, thereby preventing them from reforming as other molecules in a liquid or a solid.

Vapor formation on the heat transfer surfaces will cause marked reduction in the heat-transfer coefficient. The formation of vapor at the pump may cause pump damage by overheating and by the lack of circulation through the system.

Vapors escaping from the system may cause a safety hazard because of their toxicity or flammability. Furthermore, the fluid depletion represents a financial loss. Although a pressurized system may solve this problem, large pressure requirements will increase the cost of the thermal chamber design. Here again there must be trade-offs.

Because of the safety factor, the flash and fire points must be carefully evaluated. The higher the value for these quantities, the less the danger in the event of accidental spillage or system leakage. For cold operations of the chamber, low freezing and low pour points of the fluid are extremely desirable. A high fluid density will reduce the overall volume or size of the equipment. It also will provide some reduction in pumping power.(17) A low viscosity will not only increase the heat-transfer coefficient, but will also substantially reduce the frictional pressure drop, thus reducing pumping power.

A fluid with high specific heat will have the effect of lowering the required size of the system. The Dittus and Boelter heat-transfer coefficient equation (5) shows that thermal conductivity will effect heat-transfer from fluid directly. An increase of twenty percent would decrease the size of the equipment by approximately one-sixth.

V. ANALYTICAL METHOD FOR COMPARING HEAT-TRANSFER CAPABILITIES AND HORSEPOWER REQUIREMENTS OF FLUIDS

This section is concerned with a method for comparative analytical evaluation of heat-transfer fluids, rather than specific numerical results. The objective is to compare one reference fluid to other fluids with respect to heat-transfer capabilities and horsepower or pumping power requirements. In order to compare horsepower, the fluid weight flow rate and average mean operational temperature for evaluating the fluid's physical constants are required. The average mean operational fluid temperature and average time rate of change of panel temperature are required to evaluate the fluid weight flow rate. The time rate of change of panel temperature must be given in the design criteria. The average mean fluid temperature must be evaluated, but this requires the fluid weight flow.

As indicated previously, up-to-date lists of qualified high temperature fluids and their properties may be obtained from the fluid manufacturers, such as Minnesota Mining and Manufacturing Co., Dow-Corning, General Electric Co. and Union Carbide Corp.

Water is recommended for use as the reference fluid, to evaluate the necessary relationships since accurate physical property values can be obtained from the steam tables.(11) High vapor pressure is the main disadvantage of actually using water as the heat-transfer fluid for the thermal chamber. The vapor pressure of water at 600°F is 1528.2 psi, 2193.5 psi at 650°F and 3079 psi at 700°F. Furthermore, the chamber would be limited to high temperature operation. No cold chamber work could be accomplished because of the high freezing point of water. Nevertheless it is useful as a reference point for comparative evaluations of all other fluids.

A. Temperature Considerations

The heat-transfer fluid used in the thermal chamber will be subject to rapid changes over a wide range of temperatures. In the warm-up process during the first increments of time, the temperature differences between the fluid and the chamber panel will be at their maximum. In the analytical evaluation presented here, the fluid weight flow rate (for the given fluid) is considered constant. The overall heat-transfer coefficient is then essentially dependent on temperature only. The overall heat-transfer rate is directly proportional to the temperature difference. Thus the initial heat-transfer rates are highest, resulting in the lowest fluid outlet temperatures. As the temperature of the panels rises, the temperature difference diminishes, and the heat-transfer rate drops, resulting in a rise in outlet fluid temperature. This leads to a steady decrease in the rate of panel temperature rise. The converse is true of the log mean temperature of the fluid.

The evaluation of the mean fluid temperature value for a given warm-up period is somewhat complicated by the number of temperature variables. For this reason it is believed that the mean temperature of a given fluid can be evaluated most conveniently by an incremental approach.

B. Method for the Evaluation of the Average Mean Temperature of the Fluid

Using an incremental approach, values of mean temperature are calculated over a small even increment of time, $\Delta\theta_j$. The resulting mean temperatures can be plotted against time. Each increment of time is given its average mean fluid temperature. For a particular time period or number of increments the average mean temperature may be obtained arithmetically.

C. Computation of Panel Temperatures

In order to compute the incremental average mean fluid temperature, the incremental panel temperature must first be calculated.

The following assumptions are made:

1. The fluid mean temperature, T_M, over a small finite increment of time is assumed to be constant.

2. Temperature of the panel is considered essentially uniform due to the counter-current uniform temperature design previously discussed.

3. The panel at the start is step-wise suddenly exposed to the high temperature fluid.

4. It should be noted that for the first cut an estimation must be made of the average mean temperature so the physical values of the fluid can be determined for use in the formulations.

5. It is assumed in this initial analysis that the weight flow rate of any given fluid is constant.

To proceed with the development of the necessary design equations, the change in internal energy of the thermal chamber during a time interval $d\theta$ = The net heat flow to the thermal chamber during a time interval $d\theta$ from the fluid.

Thus,

$$C_p W_p \, dT_p = -h \, A_s \, (T_p - T_M) \, d\theta \tag{1}$$

dT_p = temperature change of panel during $d\theta$

Variables T_p and θ are separated, for a differential time interval $d\theta$, leading to,

$$-\frac{dT_p}{T_p - T_M} = -\frac{d(T_p - T_M)}{(T_p - T_M)} = \frac{h \, A_s}{C_p W_p} \, d\theta \tag{2}$$

If T_M is assumed constant during a $\Delta\theta$, then $d(T - T_M) = dT$. Integrating Equation (2)

$$\ln \frac{T_p - T_M}{T_{pj} - T_M} = \frac{-h \, A_s}{C_p W_p} \theta$$

Where T_{pj} is the initial temperature of the panel during the time increment $(\Delta X)_j$,

$$\frac{T_p - T_M}{T_{pj} - T_M} = e^{(\bar{h} \, A_s / C_p W_p) \, \theta} \tag{3}$$

Equation (3) can also be expressed for convenience in terms of dimensionless parameters.

$$\frac{V_p}{A_s} = L = \frac{W_p}{\rho_p A_s}$$

Where L = significant length = $\frac{\text{Volume}}{\text{Surface Area}}$

$$\frac{h \, \theta \, L K_p}{C_p \rho_p L^2 K_p} = \frac{HL}{K_p} \frac{K_p}{C_p \rho_p} \frac{\theta}{L^2} = (Bi) \; (Fo) \tag{4}$$

from Reference 14,

$$\frac{hL}{K_p} = \text{Biot number} = Bi$$

$$\frac{a\theta}{L^2} = \text{Fourier Modulus} = Fo$$

$$a = \frac{K_p}{C_p \rho_p} = \text{Thermal diffusivity}$$

Equation (4) is plotted in Figure 9 and may be used as time saver in calculations.

D. Time Rate of Change of the Thermal Panel Temperature

The resulting panel temperature values can now be plotted incrementally with time. The slope of this curve $\frac{\Delta T_j}{(\Delta\theta)_j}$, the time rate of change of thermal chamber panel temperature provides valuable design information. If it is desired to have a constant rate of panel temperature increase, then $\frac{dt}{d\theta}$ must equal a constant. This can be accomplished by maintaining the heat-transfer rate constant. Inspection of Equation (1) shows that where A_s, the heat-transfer area is constant, the heat-transfer coefficient, h, must be increased proportionally to the decrease in temperature difference, (T_p-T_M), in order to maintain the heat-transfer rate constant.

Within limits, the heat-transfer coefficient may be increased by an increase in the fluid weight flow rate. The heat-transfer rate can also be controlled by variations in the inlet fluid temperature. By the proper scheduling of the fluid weight flow rate and/or the inlet fluid temperature, many variations in temperature programs can be accomplished.

E. Computation of the Average Mean Fluid Temperature, T_M

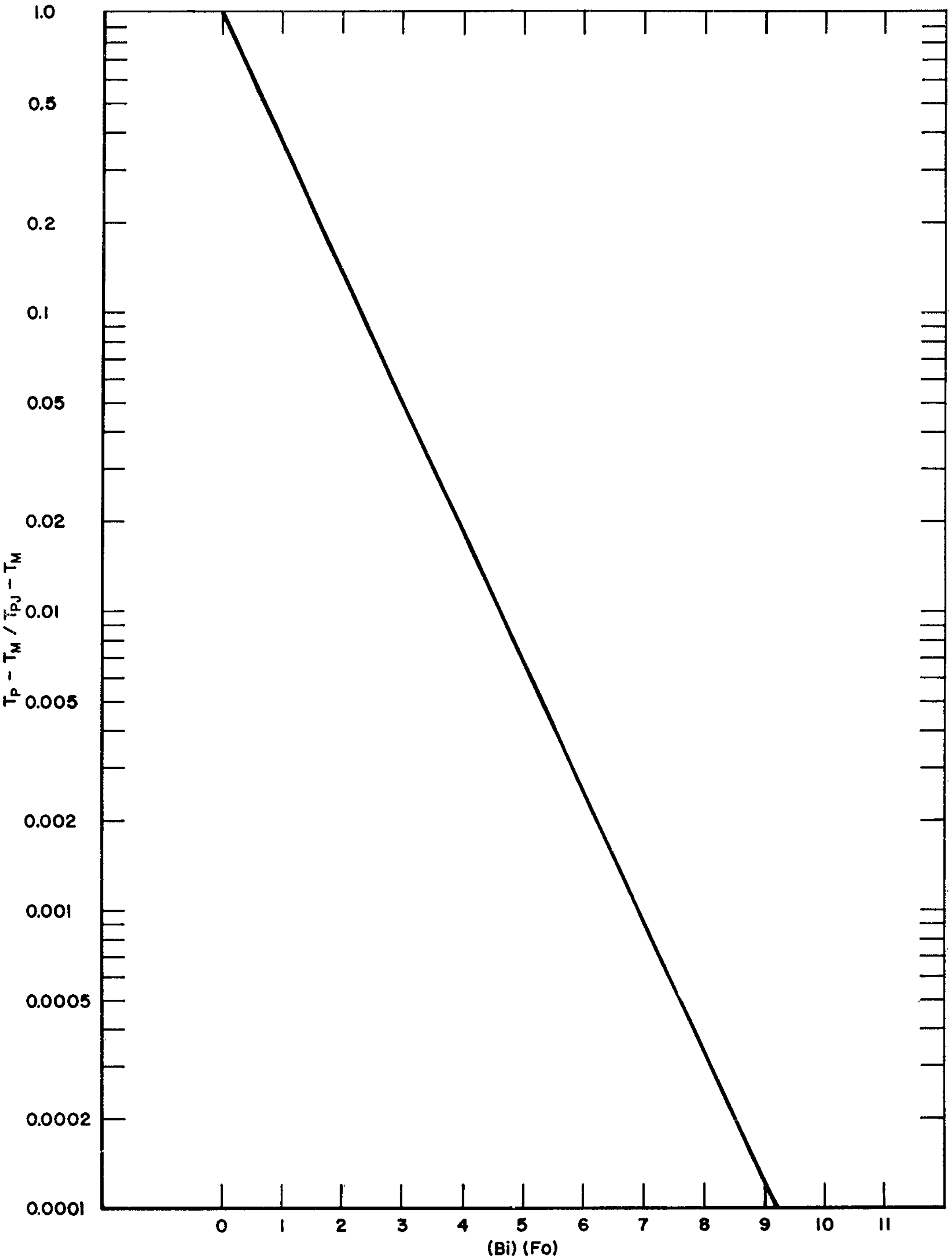
As each new panel temperature, T_j, is calculated, it will be necessary to calculate a new mean fluid temperature, T_M, to be used in Equation (3) to calculate the panel temperature, $T_j + 1$, for the next increment of time.

The mean fluid temperature, T_M, may be calculated from the mean temperature difference, ΔT_M, for each increment of time, $(\Delta\theta)_j$. ΔT_M is equal to the temperature difference between the mean fluid temperature and the panel temperature.

Thus,

$$\Delta T_M = T_M - T$$

$$T_M = T + \Delta T_M$$



NOMOGRAPH FOR DETERMINING TEMPERATURE RATIO

FIGURE 9

Now let

q = The net heat flow to the thermal chamber during a time interval $(\Delta\theta)_j$ = The change in internal energy of the fluid during a time interval $(\Delta\theta)_j$

$$q = \Delta T_M \, h \, A_s = (T_1 - T_o)\, CW \tag{5}$$

$$T_1 - T_o = \Delta T_M \frac{h\,A_s}{CW} = \frac{q}{CW}$$

$$T_o = T_1 - \Delta T_M \frac{h\,A_s}{CW} = T_1 - \frac{q}{CW}$$

$$\Delta T_1 = (T_1 - T)$$

$$\Delta T_M = \frac{(T_1 - T) - (T_o - T)}{\ln\left(\dfrac{T_1 - T}{T_o - T}\right)}$$

$$\Delta T_M = \frac{T_1 - T_o}{\ln\left(\dfrac{T_1 - T}{T_o - T}\right)}$$

$$\Delta T_M = \frac{\dfrac{q}{CW}}{\ln\left(\dfrac{\Delta T_1}{\left(T_1 - \dfrac{q}{CW}\right) - T}\right)}$$

$$\Delta T_M = \frac{q}{WC\ \ln\left(\dfrac{\Delta T_1}{T_1 - \dfrac{q}{WC}}\right)}$$

$$\Delta T_M = \frac{q}{WC\ \ln\left(\dfrac{\dfrac{\Delta T_1}{\Delta T_1}}{\dfrac{\Delta T_1}{\Delta T_1} - \dfrac{q}{WC\,\Delta T_1}}\right)}$$

$$\Delta T_M = \frac{q}{WC\ \ln\left(1 - \dfrac{q}{WC\ \Delta T_1}\right)^{-1}}$$

$$\Delta T_M = \frac{-\,q}{WC\ \ln\left(1 - \dfrac{q}{\Delta T_1\ WC}\right)} \tag{6}$$

Since,

$$q = \Delta T_M\, h\, A_s$$

$$\Delta T_M = \frac{-\Delta T_M\, h\, A_s}{WC \ln \left(1 - \dfrac{\Delta T_M\, h\, A_s}{\Delta T_1\ \ WC}\right)}$$

and,

$$R = \frac{h\, A_s}{WC} \tag{7}$$

can be defined,

$$\Delta T_M = \frac{-\Delta T_M\, R}{\ln\left(1 - \dfrac{\Delta T_M}{\Delta T_1}(R)\right)}$$

$$\ln\left(1 - \frac{\Delta T_M}{\Delta T_1} R\right) = -R$$

$$\left(1 - \frac{\Delta T_M}{\Delta T_1} R\right) = e^{-R}$$

$$\Delta T_M = \frac{\Delta T_1}{R}\left(1 - e^{-R}\right) \tag{8}$$

$$T_M = T + \Delta T_M \tag{9}$$

F. "First Cut" Estimation

To calculate the mean fluid temperature difference, ΔT_M, it is necessary to evaluate the constant R. To do this the values for weight flow, heat-transfer coefficient, area of heat-transfer surface, and fluid specific heat must be known.

These values cannot be given accurately until the average mean fluid temperature, T_M, and the average mean fluid temperature difference, ΔT_M, are known. As a first cut, then, the fluid temperatures must be approximated.

Computations

1. Compute q

$$q = C_p\, W_p \left(\frac{\Delta T}{\Delta \theta}\right) \tag{10}$$

Where $\left(\frac{\Delta T}{\Delta \theta}\right)$ equals the average time rate of change of panel temperature during the total time period for temperature increase range required. This value must be given in the design criteria.

2. Compute h

$$h = 0.023 \frac{K}{D} \left(\frac{DW}{A\mu}\right)^{0.8} \left(\frac{C\mu}{K}\right)^{0.4} \quad \text{(Only turbulent flow to consider - see Page 34)} \qquad (11)$$

In this equation the fluid weight flow, W, is not known, but h can be solved in terms of W.

3. Compute W

$$q = h\, A_s\, \Delta T_M \quad \text{(Assuming } h = U\text{)}$$

Substituting Equation (11) for h, and Equation (6) for ΔT_M, the following equation may be written.

$$q = 0.023 \frac{K}{D} \left(\frac{DW}{A\mu}\right)^{0.8} \left(\frac{C\mu}{K}\right)^{0.4} A_s \quad \frac{-q}{WC \ln \left(1 - \dfrac{q}{\Delta T_1\, WC}\right)}$$

$$-WC \ln \left(1 - \frac{q}{\Delta TW\, C}\right) = 0.023 \frac{K}{D} \left(\frac{DW}{A\mu}\right)^{0.8} \left(\frac{C\mu}{K}\right)^{0.4} A_s$$

$$-W^{0.2} \ln \left(1 - \frac{q}{\Delta T' W\, C}\right) = \frac{0.023}{C} \frac{K}{D} \left(\frac{D}{A\mu}\right)^{0.8} \left(\frac{C\mu}{K}\right)^{0.4} A_s \qquad (12)$$

Solve for W.

It is suggested that this equation be solved on a computer for W because of the trial and error method required in its solution.

The value W can now be returned to Equation (7) so that a "first-cut" can be made to evaluate the average fluid mean temperature, T_F, and the average mean temperature difference, ΔT_M. The number of iterations required depends on the accuracy desired. The percentage change in the calculated value of weight flow, W, can be used as an accuracy indicator.

Once the comparative values of weight flow have been established for different fluids, these values can be used either for calculating the pumping power requirement directly or evaluating pumping power on a comparative basis, from the equations on the last page of the appendix to this report.

The actual calculations for representative fluids are beyond the scope of this report. However, presentation of some aspects of specific fluid properties and some further considerations for chamber design based on the approaches examined seems worthwhile as a further preliminary step to establish the feasibility in terms of approximate costs.

VI. SOME PRACTICAL ASPECTS IN THE DESIGN OF A FLUID WALL CHAMBER

A. Suitable Liquids and Wall Materials

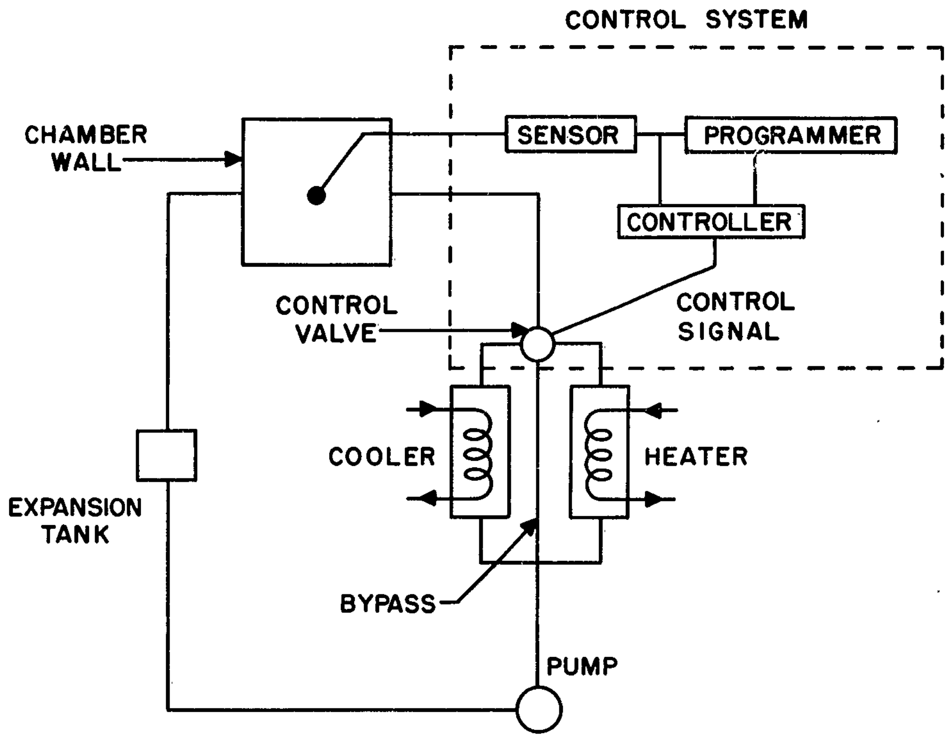
Although liquids in general are limited in temperature by their boiling points, certain silicon-derived liquids were found to possess sufficiently high boiling points and low enough change in viscosity with temperature to be suitable for heat transfer through a wide range in temperature. Fluids that appear to be particularly suitable are dimethyl siloxane fluid of approximately 50 centistoke viscosity at room temperature, which is marketed under trade names by Dow Corning, General Electric, and Union Carbide, and a tetra-aryl silicate fluid, marketed under a trade name by American Hydrotherm Corp.

A factor to be considered in addition to the liquid itself is the ducting which will make up the walls of the chamber and contain the liquid. For efficiency of heat transfer, it is desirable to minimize the product of mass per unit area and heat capacity, since the amount of heat required for a given rate of increase in temperature depends directly on this. Conflicting with this requirement, which implies using a thin, low density material, is the internal pressure developed both hydrostatically and frictionally within the duct. An approach suitable for quite high pressures is to form the wall of small diameter tubes joined side by side. One can visualize a large variety of such shapes; however for the same dimensions, large differences in either heat transfer properties or construction cost would not be expected. One example of a type of construction that appears to be suitable is manufactured commercially as an aircraft structural material, by Ryan Aeronautical Company under the name "MiniWate". It is a sandwich similar to corrugated cardboard formed of stainless steel sheet. The price of this ranges from $25 per square foot in quantities of 100 square feet to below $20 per square foot in large quantities.

B. Equipment Required for Indirect Application of Heat

If a test chamber is to be built which depends for heating upon the flow of a fluid through the ducts forming its walls, certain additional equipment must be supplied to circulate and to heat or cool the fluid. This equipment is generalized in Figure 10 and can be seen to include a pump, heat exchangers for heating and cooling, a control valve, and a control system whose function is to maintain actual panel temperature equal to the programmed temperature. The system shown is assumed to be applied to a liquid working fluid. Liquids possessing good heat transfer properties, such as the silicon-derived fluids discussed earlier, required much less heat exchanger surface area to transfer the same amount of heat as a gas. Duct or pipe cross-sections, because of the greater density and heat capacity, are likewise much smaller than for gases.

Considering the system of Figure 10 for liquids, details of the various components can be filled in. Since the pressure rise is low, a centrifugal (rather than positive displacement) pump can be used. Temperature of the cold



POSSIBLE SYSTEM FOR HEATING CHAMBER WALL WITH CIRCULATED LIQUID

FIGURE 10

and hot side heat exchangers can be relatively constant, and thus their thermal inertia need not be low. The control valve should proportion between hot and "bypass", depending on the sign of the error signal, thus minimizing the amount of energy used. Since the liquid is relatively incompressible, and since large changes in volume occur as a result of temperature changes, an expansion tank must be provided.

C. Equipment for Controlling the Application of Heat

A control system of the type shown as part of Figure 10 can be used with little modification whether heat is applied directly or indirectly. Basic elements of this control system consist of a sensor, programmer, controller, and control device. For example, these might be a resistance bulb or thermocouple type temperature sensor, a cam or curve-follower type programmer, and a power amplifying controller which compares actual temperature with programmed temperature and transmits the resulting error signal to the control device. Principal difference between a direct and an indirect system would be in the type of control device used.

The most feasible direct systems use electrical energy proportionally controlled with a saturable reactor. The indirect systems considered could also be controlled by varying the temperature of an electric heater in the fluid. Because heaters tend to have a high thermal inertia, and particularly if cooling is also desired, a system in which the temperature of heater and cooler are constant and control is obtained by proportional mixing with a valve is preferable.

VII. DISCUSSION AND CONCLUSION

Integrating current knowledge of human limitations when exposed to a rapid increase in temperature, with possible means of generating rapid changes in temperature and the relative costs of such systems, possible designs can be synthesized. Cost divides these possibilities into two groups, one in which only temperatures above room temperature are to be produced and controlled, and another which, when supplied with a cold source, can produce and control temperatures above and below room temperature.

Considering first the heating-only case, electric heating elements bonded to the surface possess simplicity and lowest cost. Of the various types investigated, "Electromesh", with higher temperature capability and slightly lower cost is to be preferred. If it is decided that a compromise to 500^{o}F is not allowable, radiant heating by means of the type T-3 quartz-tube infra-red lamps may be used. Since in both cases cost of the thin supporting metal wall is negligible, cost of the chamber can be approximated as $8 and $21 per square foot, respectively.

In the heating-and-cooling case, using a fluid wall, cost of the wall structure, as discussed earlier, is on the order of $25 per square foot. Cost of the liquids, the silicone derivatives ranges between $3 and $5 per pound. Since the ducting is about 1/4 inch thick, 0.021 cubic feet of fluid are required for each square foot of panel. Fluid density is on the order of 63 pounds per cubic foot, so that approximately 1.3 lb per square foot is required, and fluid cost will range from about $4 to $6.50 per square foot.

Cost per unit area of the two alternatives are summarized in Table 3. In addition to these costs which can be expressed on an area basis, the control system, which is essentially area-independent, must be considered. The sensor, programmer, controller, and control device combination was considered earlier. From this group, the control device (reactor or valve), whose size is dependent on area, has been listed with the alternatives above. The remaining three components cost approximately $400 if a cam programmer (West Instrument Corp.) is used. If an X-Y plotter (F. L. Moseley Co.) is desired, approximately $1200 must be added.

From the information developed in this study, it appears feasible that a cubical configuration chamber meeting the required specifications could be built on the basis of an 8 x 8 ft. size wall unit for well under $50,000. If the fluid wall approach was used such a chamber would have the possibility for further development into below room temperature ranges.

TABLE 3

COST SUMMARY

	Cost per square foot	
Alternative 1. Heating only. (Electric heaters)		
a. bonded resistance elements	$ 8	
b. radiant heating with lamps		$21
power moderator (reactor)	$ 6	$ 6
Total	$14	$27
Alternative 2. Heating and cooling capability. (Liquid)		
wall structure	$25	
silicon derivative fluid	$ 4	
circulation pump	$ 3	
heat exchangers and piping	$ 5	
control valve	$ 2	
Total	$39	

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APPENDIX

GLOSSARY OF DIMENSIONAL SYMBOLS AND UNITS

SYMBOL	QUANTITY	UNITS
A	AREA NORMAL TO DIRECTION OF FLUID FLOW	$Ft.^2$
A_s	AREA OF HEAT-TRANSFER SURFACE	$Ft.^2$
A	THERMAL DIFFUSIVITY $\frac{K}{C\rho}$	$\frac{Ft.^2}{HR.}$
C	FLUID SPECIFIC HEAT	BTU/LB - ^oF
C_p	PANEL SPECIFIC HEAT	BTU/LB - ^oF
D	HYDRAULIC DIAMETER $4\ \frac{\text{flow cross-sectional area}}{\text{wetted perimeter}}$	Ft.
g	ACCELERATION OF GRAVITY (4.17×10^8 ft/Hr^2)	Ft./HR^2
h	HEAT-TRANSFER SURFACE COEFFICIENT	BTU/HR-Ft^2 - ^oF
K	FLUID THERMAL CONDUCTIVITY	BTU/HR-Ft^2 - ^oF/Ft
K_p	PANEL THERMAL CONDUCTIVITY	BTU/HR-Ft^2 - ^oF/Ft
L	LENGTH	Ft.
q	TIME RATE OF HEAT TRANSFER	BTU/HR
Q	VOLUMETRIC FLOW RATE	Ft^3/HR
ΔP	PRESSURE DIFFERENCE	LB/$Ft.^2$
T	TEMPERATURE	^oF
T_1	FLUID TEMPERATURE AT INLET	^oF
T_o	FLUID TEMPERATURE AT EXIT	^oF
T_M	FLUID MEAN TEMPERATURE	^oF
T_F	FLUID TEMPERATURE	^oF
T_P	PANEL TEMPERATURE	^oF
ΔT_i	INLET TEMPERATURE DIFFERENCE (T_1-T_P)	^oF

GLOSSARY OF DIMENSIONAL SYMBOLS AND UNITS (Cont.)

SYMBOL	QUANTITY	UNITS
ΔT_o	OUTLET TEMPERATURE DIFFERENCE (T_o-T_p)	^oF
ΔT_m	LOG-MEAN-TEMPERATURE DIFFERENCE OF (T_F-T_p)	^oF
U	OVER-ALL COEFFICIENT OF HEAT TRANSFER	BTU/HR-Ft^2 - ^oF
V	VELOCITY	FT/HR
V_D	VOLUME OF THE PANEL	$Ft.^3$
W	WEIGHT FLOW RATE OF FLUID	LB/HR
W_p	PANEL WEIGHT	LB
ρ	DENSITY OF FLUID	LB/HR
ρ_p	DENSITY OF PANEL	LB/HR
μ	ABSOLUTE VISCOSITY	LB/Ft. - HR.
θ	TIME	HR.

GLOSSARY OF DIMENSIONLESS QUANTITIES

SYMBOL	QUANTITY	FORMULA
Re	REYNOLDS NUMBER	$\frac{DV\rho}{\mu} = \frac{DW}{A\mu}$ *
Nw	NUSSELT NUMBER	$\frac{hc\ D}{K}$
Pr	PRANDTL NUMBER	$\frac{Cp\mu}{K}$
St	STANTON NUMBER	$\frac{hc}{V\ Cp\ \rho}$
f	FRICTION FACTOR	$\Delta P \left(\frac{D2gc}{L\rho V^2}\right)$
f_F	FRICTION FACTOR APPLIED TO FANNING EQUATION	$\frac{f}{4}$
Bi	BIOT NUMBER	$\frac{hL}{K_p}$
Fo	FOURIER MODULUS	$a\ \frac{\theta}{L^2}$

---

*

$$V = \frac{Q}{A}$$

$$Re = \frac{DV\rho}{\mu} \qquad Q = \frac{W}{\rho} \qquad Re = \frac{D}{\mu}\left(\frac{W}{A}\right)$$

$$V = \frac{W}{A\rho}$$

1. Heat Transfer

$$q = UA_s \Delta T_m$$

2. Fluid Heat Gain

$$q = WC\ (T_{out} - T_{in})$$

3. Heat Transfer Coefficient

a. Turbulent Flow, "Dittus and Boelter" Equation (5)

$$h_c = 0.023\ \frac{K}{D}\ Re^{0.8}\ \ Pr^{0.4}$$

$$h_c = 0.023\ \frac{K}{D} \left(\frac{DW}{AM}\right)^{0.8} \left(\frac{C\rho\mu}{K}\right)^{0.4}$$

4. Mean Temperature Difference[2]

$$\Delta T_m = \frac{\Delta T_a - \Delta T_b}{\ln(\Delta T_a / \Delta T_b)}$$

5. Continuity Equation

$$W = \rho VA$$

6. Pressure Loss in a Circular Pipe[2]

$$\Delta P = f\ \frac{L\rho\ V^2}{8g_c m} \quad \text{(Darcy - Weisbach formula)}$$

$$\Delta P = f\ \frac{L\rho\ V^2}{2Dg_c}$$

7. Horsepower (Supplied to Fluid)

$$HP = \frac{Q\Delta P}{550 \times 3{,}600} = \frac{Q\Delta P}{1{,}980{,}000}$$

8. Friction Factor for Darcy Equation

a. Turbulent Flow[1]

$$f = \frac{0.184}{(Re)^{0.2}} \qquad \left(f_F = \frac{0.046}{(Re)^{0.2}}\right)$$

$$HP = \frac{Q\,\Delta P}{550\ (3600)}$$

$$\Delta P = f\left(\frac{L\rho V^2}{D2g_c}\right) \qquad g_c = 4.17 \times 10^8\ \text{ft/hr}^2$$

$$f = .184\ (Re)^{-0.2}$$

$$\frac{HP_1}{HP_2} = \frac{Q_1\ \ \Delta P_1}{Q_2\ \ \Delta P_2}$$

$$= \frac{Q_1}{Q_2}\left(\frac{f_1\ \dfrac{L\rho_1\ V_1^{\ 2}}{D2g_c}}{f_2\ \dfrac{L\rho_2\ V_2^{\ 2}}{D^2g_c}}\right) = \frac{Q_1}{Q_2}\ \frac{f_1}{f_2}\ \frac{\rho_1\ V_1^{\ 2}}{\rho_2\ V_2^{\ 2}}$$

$$= \frac{\dfrac{W_1}{\rho_1}\ \ f_1\ \rho_1\left(\dfrac{W_1}{\rho_1\ A}\right)^2}{\dfrac{W_2}{\rho_2}\ \ f_2\ \rho_2\left(\dfrac{W_2}{\rho_2\ A}\right)^2} \qquad V = \frac{Q}{A} \quad Q = \frac{W}{\rho} \quad V = \frac{W}{\rho A}$$

$$\frac{HP_1}{HP_2} = \left(\frac{W_1}{W_2}\right)^3 \left(\frac{\rho_2}{\rho_1}\right)^2 \left(\frac{Re_2}{Re_1}\right)^{0.2}$$

$$= \left(\frac{W_1}{W_2}\right)^3 \left(\frac{\rho_2}{\rho_1}\right)^2 \frac{\left(\dfrac{D\ \ W_2}{\mu_2\ A}\right)^{0.2}}{\left(\dfrac{D\ \ W_1}{\mu_1\ A}\right)^{0.2}}$$

$$\frac{HP_1}{HP_2} = \left(\frac{W_1}{W_2}\right)^{2.8} \left(\frac{\rho_2}{\rho_1}\right)^2 \left(\frac{\mu_1}{\mu_2}\right)^{0.2}$$

Aerospace Medical Division, 6570th Aerospace Medical Research Laboratories, Wright-Patterson AFB, Ohio. Rpt. No. AMRL-TDR-63-15. FEASIBILITY STUDY OF A SMALL THERMAL CHAMBER FOR PROGRAMMED TRANSIENT INCREASES AND DECREASES IN TEMPERATURE. Final report, Mar 63, iv + 36 pp incl. illus., tables, 25 refs. Unclassified report

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1. Thermal chamber
2. Temperature control
3. Heat tolerance
I. AFSC Project 7222, Task 722204
II. Biomedical Laboratory
III. Contract AF 33(616)-6763
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