Report Number: NASA CR-72045
K-D-1870

SUMMARY REPORT

HEAT EXCHANGER TEST EXPERIMENTAL APPARATUS
SECTION I - HELIUM HEATER
SECTION II - HELIUM CIRCULATOR

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SUMMARY AND INTRODUCTION

The work to be performed under NASA authorization consisted of furnishing the services, labor and materials to design, fabricate and test a complete experimental apparatus to evaluate the characteristics of heat exchangers over a wide range of operating conditions. During the first phase of testing, the heat transfer characteristics and operational limitations of the heat exchangers were to be established for a wide range of pressures, flows and temperatures. The second phase of the testing consisted of a cyclic life test.

The principal components of the apparatus consists of a helium heater, helium-to-air heat exchanger, recuperator, and variable speed compressor with precooler. These components, installed in series, permit helium to be heated and circulated so as to transfer heat to an air stream passing through the heat exchanger being evaluated.

The design requirements for the principal components were as follows:

A. Helium Heater

A helium heater was designed to heat helium gas which enters at a pressure of from 800 to 1500 psi and a temperature which may be as low as 0°F or as high as 1000°F. Helium flow through the heater is variable from a minimum of 0.01 lb/second to a maximum of 0.25 lb/sec. Maximum heat imparted to the gas was set at 360 kilowatts.

B. Recuperator

An air recuperator was designed to serve two functions, (1) to cool the exhaust air below 1000°F so that the air back-pressure regulating valve can be expected to have reasonable life, and (2) to heat the incoming air over a temperature range which may be varied by changing the percentage of supply air by-passing the recuperator.

The recuperator was designed to have an effectiveness such that it could cool an air stream leaving a test heat exchanger at 1400°F to a temperature not higher than 600°F while heating incoming air from ambient temperature over a corresponding temperature range as the flow varies from a minimum of 0.25 lb/sec. up to a maximum of 1.10 lb/sec. The pressure drop through either side of the recuperator was limited to 10 psig. In addition, the recuperator must handle hot side inlet gas temperatures up to 1500°F, withstand hot side outlet temperatures up to 1000°F, and pressures on both sides up to 300 psig.

C. Recirculator or Compressor and Helium Cooler

Since the quantity of helium that would be required for the trailer mode of operation would be excessive, it was decided to include a compressor or recirculator. Use of the compressor made it possible to close the loop and recirculate the helium during testing. An existing design for a HECT-2 compressor was modified to make it capable of operating at 1500 psig and 700°F. The aerodynamic performance of the compressor was established to recirculate the gas at the flows, pressures and temperatures as specified for the heater. To assure that the compressor suction gas temperature was below 600°F, a helium cooler (He to water) was added between the discharge of the heat exchanger and the suction of the compressor.

D. Test Heat Exchanger

A test heat exchanger was designed to transfer energy from a heated, recirculated, helium stream to a once-through air stream. Design conditions for the test heat exchanger were as follows: With helium temperatures of 1800°F in and 800°F out, at a static pressure of 1500 psig and a pressure drop to static pressure ratio not greater than 0.005, heat was to be transferred to air raising its temperature from an inlet at 500°F to 1400°F at the outlet, at a static pressure of 71 psig and a pressure drop to static pressure ratio of not over 0.10. The air side was designed to withstand a pressure of 285 psig. Tube length used was not to exceed 6 feet.

After completion of the design of the helium heater, heat exchanger and compressor, the balance of the work on the other components of the system and cyclic life was deferred because of budgetary limitations.

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ABSTRACT

This portion of the report describes the design and analysis of a molybdenum tube which is to be used to heat helium to 2500°F for use in a high temperature experimental heat exchanger test facility. This heater tube is enclosed in a pressure vessel; the inner vessel walls are protected by radiation shields. The heater is energized at a maximum electrical capacity of 540 Kw at a voltage of 36 and an amperage of 15,000.

The tube is conservatively designed to operate at a constant axial wall temperature of 4000°R with a maximum operation temperature limit of 4400°R. The tube is 73 inches in length with an inside diameter of 0.875 inches and maximum outside diameter of 1.651 inches at the outlet end. The tube wall thickness tapers to a minimum outside diameter of 1.295 inches at the inlet end. There are four (4) radiation shields which limit the pressure vessel wall temperature to 1400°R at ambient conditions externally.

This tube can be heated safely to operating temperature before the helium gas is allowed to flow through the test heat exchanger, thus enabling the exchanger to be thermally shocked.

HEAT EXCHANGER TEST EXPERIMENTAL APPARATUS

HELIUM HEATER

I. OBJECTIVES

The heater system is intended to provide helium at a temperature of 2500°F and 1500 psia (maximum) to a helium-to-air heat exchanger. This heater is to be an integral portion of a test facility in which various heat exchangers can be evaluated.

This report deals with the heat transfer considerations for such a heater system, and will analyze some off-design operations.

II. CONCLUSIONS

This design study shows it is feasible to heat helium in the trailer mode of operation (0°F inlet temperature) to an exit temperature of 2500°F, without excessive pressure and power losses, in a heater that is of reasonable size.

The heater tube per se should be of the constant wall temperature type operating at the highest practical temperature possible. The material selected is molybdenum and the maximum wall temperature of operation is set at 4400°R, with design wall temperature of 4000°R. The wall thickness has been determined to be essentially linear with tube length, and thus machining will be simplified. In order to reduce the pressure casing wall temperature to a safe value, it will be necessary to install four (4) thermal radiation shields. This will ensure that the casing temperature will remain below 1000°F even at the maximum heating conditions. The design heater wall temperature for radiation shielding is set at 4400°R.

The summary of design dimensions is given in Section IX of this report. Mechanical details of the heater system are to be given in another report.

A study of the time to heat the metal walls of the heater tube has shown that it (the time) will be sufficiently large so that the flow of helium need not be started concurrently with the electrical energy to the tube. This will enable the test operator to thermally shock the heat exchanger with relatively high temperature helium.

III. RECOMMENDATIONS

It is recommended that the dimensional data given under Section IX of this report be used in the construction of a heater system for use as stated in the OBJECTIVES.

IV. DATA GIVEN AND NOMENCLATURE

A. The purpose of the heater is to provide gas at a temperature of 2500°F to a test heat exchanger.

B. The inlet temperature range to the heater is 0° - 1000°F. The 0°F inlet temperature is associated with a possible trailer-mode operation of the test facility. The 1000°F maximum inlet temperature does not appear to be attainable unless development work is performed on a compressor for the closed-loop operation.

C. The nominal pressure level range of operation is 800 - 1500 psia.

D. The helium flow rates are to be compatible with the desired temperature rise with a maximum power input to the helium of 360 kilowatts. The helium flow will be variable from a minimum of 0.01 lb/sec to a maximum of 0.25 lb/sec.

E. The maximum available electrical capacity to the test facility for the heater is 540 kilowatts at a voltage of 36 and an amperage of 15,000. The maximum allowable voltage drop from the point of availability to the heater tube per se is 6 volts. The heater design shall use a maximum of 30 volts as measured at the point of entry into the pressure shell.

F. The heater is to operate satisfactorily at a current input of 13,500 amperes. The helium flow rate is to be 0.071 lb/sec at an entrance temperature of 0°F, an exit temperature of 1800°F, and a system pressure of 1500 psia.

G. The following nomenclature is used in this report.

h_f = film coefficient of heat transfer, Btu/ft^2-sec-°R

C_p = coefficient of specific heat, Btu/lb-°R

ρ = gas density, lb/ft^3

ρ^* = electrical resistivity, ohms-ft

μ = coefficient of viscosity, lbm/ft-sec

A = flow area, ft^2

κ = thermal conductivity, Btu/ft-sec-°R

W = flow rate, lb/sec = GA

T_w = wall (metal) temperature, °R

T_t = gas total temperature, °R

T_f = film temperature = $(T_w + T_t)/2$, °R

P_t = total pressure, lb/ft^2

G = weight flow per unit area = ρV, lb/ft^2-sec

V = gas velocity, ft/sec

M = gas Mach number = V/C_s

C_s = velocity of sound, $\sqrt{\gamma g \mathcal{R} T}$, ft/sec

γ = ratio of specific heats

g = acceleration of gravity, ft/sec^2

$\mathcal{R}$ = gas constant = 1545/M* = 386.0 for helium

M^* = molecular weight of helium = 4.003

D = cylindrical diameter, ft, in particular, the inside diameter of heater tube

t = tube wall thickness, ft

f = friction factor

Q = heat input per unit length per unit time, Btu/ft-sec

P = electrical power, watts

I = electrical current, amperes

R = electrical resistance, ohms

Pr = Prandtl Number, $C_p\mu/\kappa$

x = distance, ft

L = total length, ft

ξ = x/L

D_H = hydraulic diameter for heating surface

D_h = hydraulic diameter for frictional effects

τ = time, seconds

Subscripts

- 0 = entering heater casing at $x = 0$
- 1 = entering heater tube
- 2 = leaving heater tube
- s = shield.

V. ASSUMPTIONS

A. The properties of the helium gas are given in the following table and Figures 1 and 2.

C_p = 1.2430, constant, at 1500 psia

M^* = molecular weight = 4.003

$\mathcal{R}$ = gas constant = 386.0

Pr = constant = 0.688.

B. The heat transfer relation for turbulent flow is given by

$$h_f = 0.023\, C_p G \Big/ \left(\frac{D_H G}{\mu_f}\right)^{0.2} (Pr)^{2/3} .$$

The hydraulic diameter D_H is defined in a manner to emphasize the particular surface film from which heat is being supplied to the gas, or

$$D_H = 4\, \frac{\text{cross sectional area}}{\text{wetted perimeter of heated surface}} .$$

C. The friction factor relation for turbulent flow, $\dfrac{D_h G}{\mu_f} > 2000$ is given by

$$f = 0.00140 + 0.125 \left(\frac{D_h G}{\mu}\right)^{-0.32} .$$

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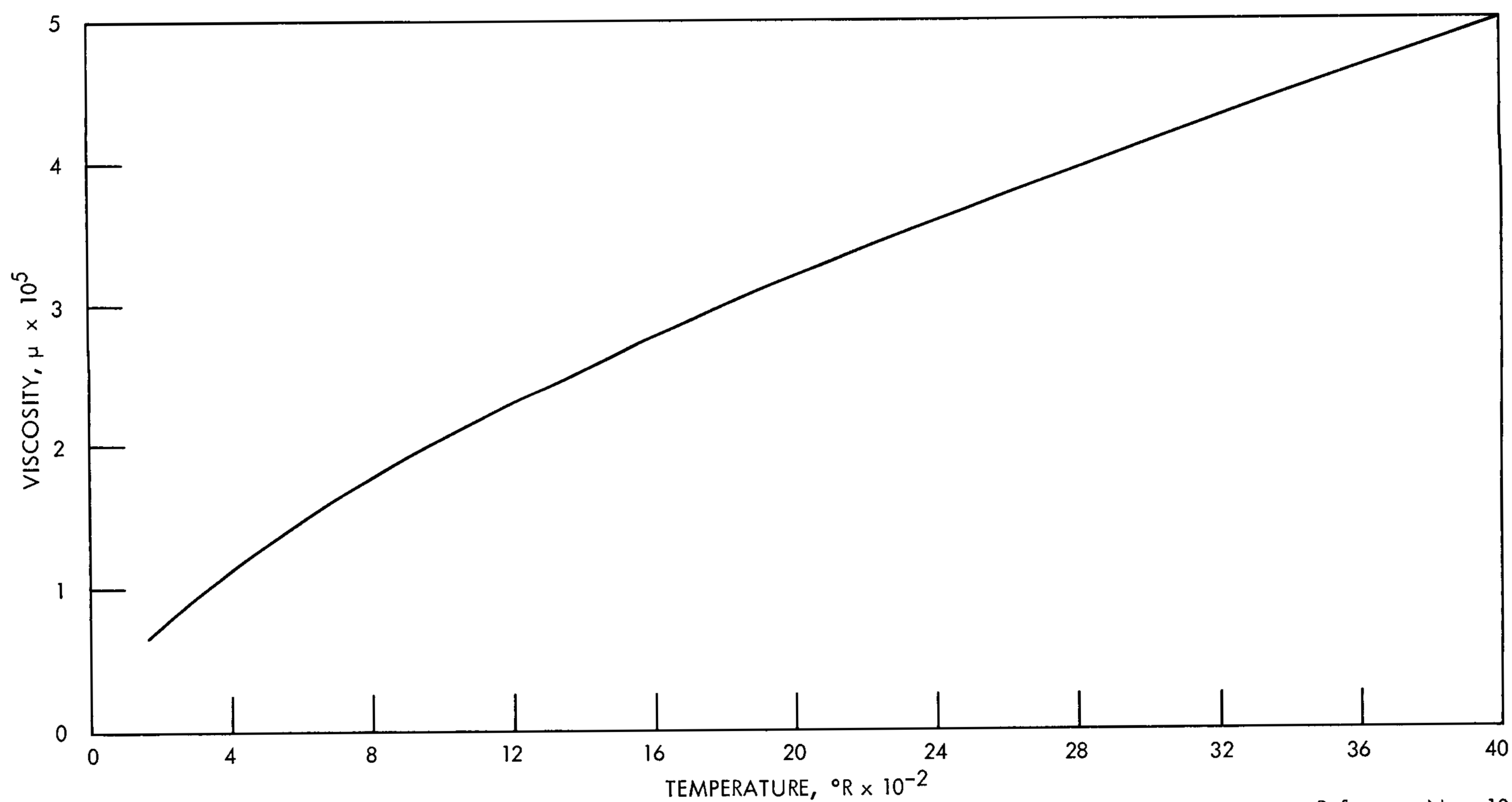

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Figure 1

VISCOSITY OF HELIUM

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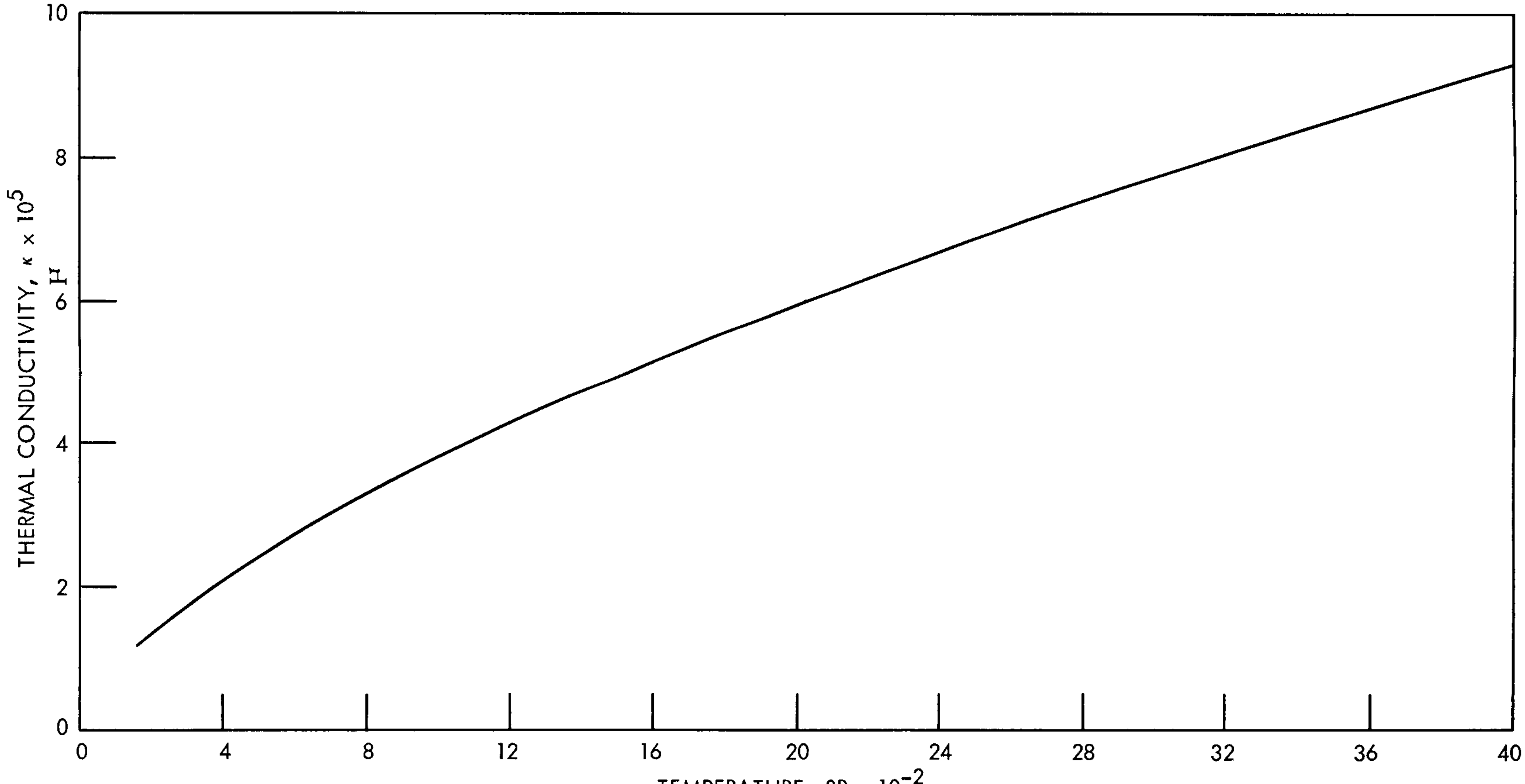

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Figure 2

THERMAL CONDUCTIVITY OF HELIUM

This relation is for a smooth tube, and D_h is the hydraulic diameter for friction effects. This hydraulic diameter is defined in the usual manner for frictional pressure drop as

$$D_h = 4 \frac{\text{cross-sectional area}}{\text{wetted perimeter}} \quad .$$

D. The fluid dynamics are considered to be described by means of conventional one-dimensional flow relations.

E. The relation between the metal temperature and the electrical power is given by Joule's Law of electric heating, viz, the heat produced in a conductor is proportional to the resistance of the conductor, to the square of the current, and to the time. From this the power expended in heat is given by

$$P = RI^2, \text{ watts.}$$

F. The value of electrical resistivity (ρ*) for molybdenum is given in Figure 3.

G. The axial heat transfer is neglected in the metal components and in the gas.

H. The heater assembly schematic is assumed for calculation purposes to be that shown in Figure 4.

VI. BASIC RELATIONS USED FOR CALCULATIONS

The helium flow region is divided into two separate regions, between the heater tube and the thermal shield and inside the heater tube. The convective heat transfer to the helium between the shield and the casing wall is neglected. This will allow the design to be conservative.

A. Determination of Flow Rate

The total power transferred to the gas stream, in kilowatts, is given by the expression,

$$P = 1.054 \, W \, C_p \, (\Delta T_t) \quad , \tag{1}$$

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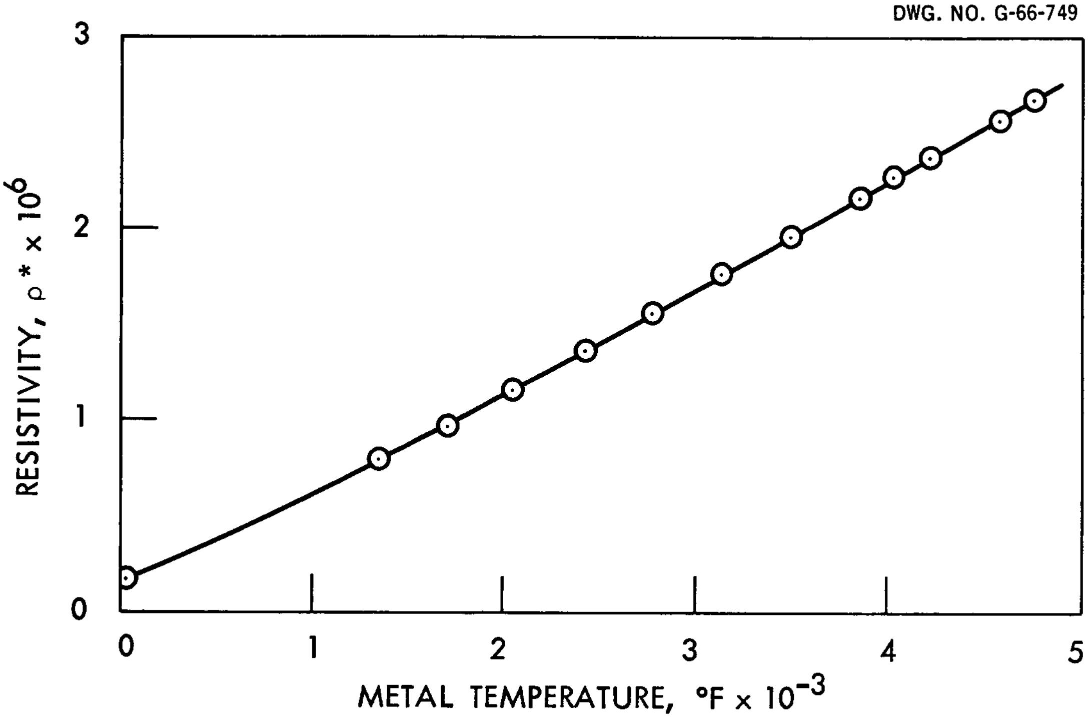

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Figure 3

ELECTRICAL RESISTIVITY OF MOLYBDENUM

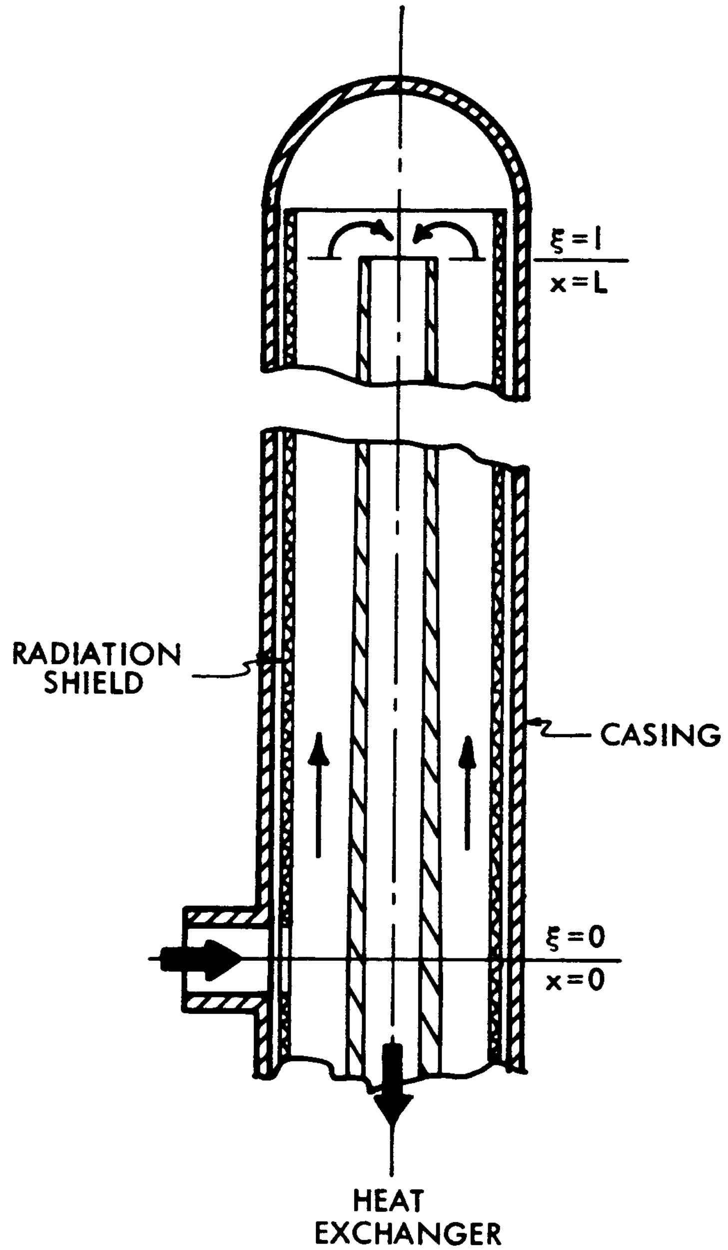


Figure 4

SCHEMATIC OF HEATER ASSEMBLY

where ΔT_t is the desired total temperature rise. The maximum value of P is set at 360 kilowatts.

B. Heat Transfer Relations

The heat transfer from the shield to the helium gas per unit length is given by

$$Q_s = h_{f,s} (T_s - T_t) \pi D_s \tag{2}$$

where the heating hydraulic diameter D_{Hs} is taken to be

$$D_{Hs} = \frac{D_s^2 - (D + 2t)^2}{D_s}$$

and therefore

$$\frac{\pi D_s h_{f,s}}{W C_p} = \frac{0.1125(\mu_{f,s})^{0.2}}{D_s^{0.8}(1 - \zeta^2)W^{0.2}}$$

ζ = $(D + 2t)/D_s$

T_s = metal temperature of shield

and $\mu_{f,s}$ = viscosity in film evaluated at $T_{f,s} = (T_s + T_t)/2$.

The heat transfer per unit length from the outside of the heater tube to the gas is represented by

$$Q_w = h_{f,w} (T_w - T_t) \pi (D + 2t) \tag{3}$$

where the heating hydraulic diameter $D_{H,w}$ is given by

$$D_{H,w} = \frac{D_s^2 - (D + 2t)^2}{D + 2t}$$

so that

$$\frac{\pi(D + 2t) h_{f,w}}{W C_p} = \frac{0.1125 \zeta^{1.2}(\mu_{f,w})^{0.2}}{D_s^{0.8}(1 - \zeta^2)W^{0.2}}$$

with T_w = heater tube wall temperature

and $\mu_{f,w}$ = helium viscosity in film evaluated at $T_{f,w} = (T_w + T_t)/2$ outside the tube.

The heat transfer per unit length inside the heater tube is

$$Q = h_f(T_w - T_t)\pi D \qquad (4)$$

where
$$\frac{\pi D}{W\ C_p} h_f = \frac{0.1125(\mu_f)^{0.2}}{D^{0.8}\ W^{0.2}} ,$$

and μ_f = viscosity in film evaluated at $(T_w + T_t)/2$ inside the tube.

The gas bulk temperature variation with respect to distance and heat input is given by:

1. Outside heater tube

$$W\ C_p \frac{dT_t}{dx} = Q_s + Q_w$$

or

$$\frac{dT_t}{d\xi} = \frac{0.1125L}{D_s^{0.8}(1-\zeta^2)W^{0.2}} \left[(\mu_{f,s})^{0.2}(T_s - T_t) + \zeta^{1.2}(\mu_{f,w})^{0.2}(T_w - T_t)\right] \qquad (5)$$

2. Inside heater tube

$$- W\ C_p \frac{dT_t}{dx} = Q$$

or

$$- \frac{dT_t}{d\xi} = \frac{0.1125L}{D^{0.8}W^{0.2}} \mu_f^{0.2}\ (T_w - T_t) \qquad (6)$$

The interest lies in minimum length to achieve the necessary design $\Delta T_t = 2500°$. This is accomplished by keeping the wall temperature at as high a level as possible and constant axially.

For this case, relation (6) can be integrated to yield

$$\frac{L}{D^{0.8}} = \frac{8.89\, W^{0.2}}{(\overline{\mu_f})^{0.2}} \ell n \left[\frac{T_w - T_{t,1}}{T_w - 2960}\right] \tag{7}$$

where $T_{t,1}$ is the gas bulk temperature entering the heater tube, °R

and $(\overline{\mu_f})^{0.2} = \int_0^1 (\mu_f)^{0.2}\, d\xi$, which is approximated closely by using the viscosity at a temperature

$$T_{f,av} = (T_w + T_{av})/2,$$

where

$$T_{av} = T_w - \frac{2960 - T_{t,1}}{\ell n\left[\frac{T_w - T_{t,1}}{T_w - 2960}\right]} \quad . \tag{8}$$

For assumed values of $T_{t,1}$ and T_w the necessary $L/D^{0.8}$ value can be determined quite accurately from (7). Maximum and minimum values are also easily calculated from the relation.

Relation (5) can be integrated into the following form:

$$\frac{T_w - T_{t,1}}{T_w - T_{t,o}} = \frac{1}{\Phi(1)} + \frac{1}{(T_w - T_{t,o})\Phi(1)} \int_o^1 B(\xi)\Phi(\xi)(T_w - T_s)d\xi \tag{9}$$

with

$$\Phi(\xi) = \exp \int_o^{\xi} A d\xi \quad ,$$

$$A(\xi) = \frac{0.1125L}{D_s^{0.8} W^{0.2}(1-\zeta^2)} \left[(\mu_{f,s})^{0.2} + \zeta^{1.2}(\mu_{f,w})^{0.2}\right]$$

$$B(\xi) = \frac{0.1125\, L}{D_s^{0.8} W^{0.2}(1-\zeta^2)} (\mu_{f,s})^{0.2}$$

Because of the small variations in the temperature outside the tube and the heater tube thickness with distance, a good approximation of (9) can be obtained by using average values of the variables. Thus

$$\frac{1}{\Phi(1)} = \exp(-\bar{A})$$

where

$$\bar{A} = \frac{0.1125L}{D_s^{0.8} W^{0.2} (1-\zeta^2)_{av}} \left[(\overline{\mu_{f,s}})^{0.2} + (\zeta_{av})^{1.2} (\overline{\mu_{f,w}})^{0.2} \right] ,$$

etc, and (9) can be written in the form

$$\frac{T_w - T_{t,1} - (\bar{B}/\bar{A})(T_w - T_s)_{av}}{T_w - T_{t,o} - (\bar{B}/\bar{A})(T_w - T_s)_{av}} = \exp(-\bar{A}) \qquad (10)$$

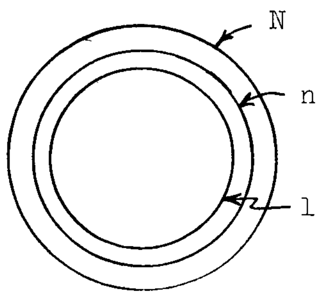
However, the average value of T_s is unknown at this point and must be determined by a radial heat balance (axial heat transfer being neglected).

C. Radiant Heat Transfer

For practical purposes, helium is transparent to thermal radiation so that only radiation between metallic surfaces is considered. It shall be assumed that all surfaces are gray in nature, and that the end effects are negligible.

The heat transfer from the casing to the ambient is relatively small so that more than one shield may be required to keep the casing temperature within safe limits. An expression will be derived for the radiant heat transfer between N shields.

Let T_n be the temperature of the nth shield with diameter D_n and emissivity ϵ_n. Then the radiant heat transfer between shield (n) and shield (n+1) is given by



$$Q_{n \to n+1} = \frac{0.481\pi}{\alpha_n}\left[\left(\frac{T_n}{1000}\right)^4 - \left(\frac{T_{n+1}}{1000}\right)^4\right]\frac{\text{Btu}}{\text{ft-sec}} \tag{11}$$

with

$$\alpha_n = \frac{1}{D_n\epsilon_n} + \frac{1-\epsilon_{n+1}}{D_{n+1}\epsilon_{n+1}} \quad .$$

It can be readily shown that the heat conducted between the shields through the helium is negligible in the determination of the shield temperatures, so that the steady radiant transfer between each shield must be equal, say $Q_n = Q$, and then the following can be derived from (11),

$$Q = \frac{0.481\pi}{\sum\limits_{n=1}^{N-1}\alpha_n}\left[\left(\frac{T_1}{1000}\right)^4 - \left(\frac{T_N}{1000}\right)^4\right] \tag{12}$$

If

$$Q^* = \frac{0.481\pi}{\alpha}\left[\left(\frac{T_1}{1000}\right)^4 - \left(\frac{T_N}{1000}\right)^4\right]$$

with

$$\alpha = \frac{1}{D_1\epsilon} + \frac{1-\epsilon_N}{D_N\epsilon_N} \quad ,$$

i.e., Q^* is the radiant heat transfer if only the first and (Nth) shield are there, the transfer Q can be related to this by

$$Q = \alpha Q^*\left(\sum_{n=1}^{N-1}\alpha_n\right)^{-1} . \tag{13}$$

If the shields are closely spaced, of the same emissivity ϵ, and D_1 is relatively large so that all D_n are approximately equal to D_1, then (13) reduces to

$$Q = \frac{1}{N-1}Q^* \quad , \tag{14}$$

a useful first approximation when determining the number of shields required between any two temperatures for a given heat transfer load.

The net radial radiant thermal transfer per foot per second from the heater tube to the first shield is given by the expression

$$Q_r = \frac{0.481\pi\left[\left(\frac{T_w}{1000}\right)^4 - \left(\frac{T_{s,1}}{1000}\right)^4\right]}{\frac{1}{(D + 2t)\epsilon_w} + \frac{1 - \epsilon_s}{\epsilon_s D_{s,1}}} \tag{15}$$

where ϵ_w = emissivity of heater tube wall

ϵ_s = emissivity of shield.

The net transfer (radiant) from the last shield (Nth) to the casing wall per foot per second is

$$Q_{r,s} = \frac{0.481\pi\left[\left(\frac{T_{s,N}}{1000}\right)^4 - \left(\frac{T_{c,i}}{1000}\right)^4\right]}{\frac{1}{\epsilon_s D_{s,N}} + \frac{1 - \epsilon_c}{\epsilon_c D_{c,i}}} \tag{16}$$

with $T_{c,i}$ = the temperature of the inside casing surface, °R

$D_{c,i}$ = the internal diameter of the casing, ft

ϵ_c = emissivity of the casing surface.

D. Heat Transfer to Ambient

The heat transfer from the casing to the ambient is considered to be of two types, natural convection and thermal radiation. Thus

$$Q_{amb} = Q_{n.c.} + Q_{r,a}$$

where

$$Q_{n.c.} = 0.000075\pi\,(D_{c,o})^{0.75}(T_{c,o}-T_a)^{1.25},\ \frac{\text{Btu}}{\text{ft-sec}}^{*} \tag{17}$$

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* See Reference 2, 3rd Edition, page 240-241, or Reference 2, 3rd Edition, page 173.

with $D_{c,o}$ = outside diameter of casing, ft,
$T_{c,o}$ = temperature of outside casing surface, °R,
T_a = ambient temperature, °R,

and
$$Q_{r,a} = 0.481\pi D_{c,o}\epsilon_c \left[\left(\frac{T_{c,o}}{1000}\right)^4 - \left(\frac{T_a}{1000}\right)^4\right] \frac{\text{Btu}}{\text{ft-sec}} \qquad (18)$$

The heat transferred across the casing by conduction must be equal to the heat transferred to the ambient. Therefore

$$Q_{amb} = Q_{cond} = \frac{2\,\pi\,k_c\,(T_{c,i} - T_{c,o})}{\ell n\,(D_{c,o}/D_{c,i})} \qquad (19)$$

where k_c = thermal conductivity of casing material.

E. Heat Balances

The total heat transfer must balance with the power input and the heat loss to the ambient. A subsidiary heat balance, which must obtain, is

$$\int_o^L Q_r\,dx = \int_o^L Q_s\,dx + \int_o^L Q_{amb}\,dx \quad . \qquad (20)$$

The left-hand side of (20) represents the heat radiated to the shield from the heater tube. On the right side of (20), the first integral represents the heat convected from the shield to the gas, while the second integral is the heat removed from the casing to the ambient surroundings. This balance will enable the shield temperatures to be determined for fixed subsidiary temperatures T_w, T_a and $T_{c,o}$. In calculations, relation (20) will be used with average values of temperature. Thus only on the average will the heat transfer relations be balanced. However, even locally, the balance is probably not too undesirable.

F. Pressure Drop Relations

For the one-dimensional flow assumed one can derive the relation

$$\frac{dP_t}{P_t} = -\frac{\gamma M^2}{2}\left[\frac{dT_t}{T_t} + \frac{4fL}{D_h}\,d\xi\right] \tag{21}$$

where now $d\xi$ is always taken in the direction of flow. One can write this in the form

$$-P_t\,dP_t = \frac{W^2\mathcal{R}}{2gA^2}\left[dT_t + \frac{4fL\,T_t}{D_h}\,d\xi\right] \tag{22}$$

since the flow Mach number for this design is extremely small (less than 0.02). Integration of (22) over the two flow regions, the outside annular region and inside the heater tube, and anticipation of the fact that the relative pressure drop $(\Delta P_t/P_{t,o})$ is extremely small, leads to the relation

$$\frac{\Delta P_t}{P_{t,o}} = a_1(T_{t,1}-T_{t,o}) + a_2(T_{t,2}-T_{t,1}) + b_1(fT_t)_{av,1} + b_2(fT_t)_{av,2} \tag{23}$$

where

$$a_1 = \frac{8\,W^2\mathcal{R}}{\pi^2\,g\,P_{t,o}^2\,D_s^4(1-\zeta^2)^2_{av,1}}$$

$$a_2 = \frac{8\,W^2\mathcal{R}}{\pi^2\,g\,P_{t,o}^2\,D^4}$$

$$b_1 = \frac{4\,a_1L}{D_s(1-\zeta)_{av,1}}$$

$$b_2 = \frac{4\,a_2L}{D} = \frac{4\,a_2}{D^{0.2}}\,(L/D^{0.8})$$

$$(fT_t)_{av,1} = T_{t,av,1}\,f(T_{t,av,1})$$

$$T_{t,av,1} = \frac{1}{2}\,(T_w + \overline{T}_s) - \frac{T_{t,1} - T_{t,o}}{\ln\left[\dfrac{T_w + \overline{T}_s - 2T_{t,o}}{T_w + \overline{T}_s - 2T_{t,1}}\right]}$$

$$(fT_t)_{av,2} = T_{t,av,2}\, f(T_{t,av,2})$$

$$T_{t,av,2} = T_w - \frac{T_{t,2} - T_{t,1}}{\ln\left[\dfrac{T_w - T_{t,1}}{T_w - T_{t,2}}\right]}$$

The friction factor relation can be rewritten in the form

$$f = 0.00140 + 0.1157\, D_h^{0.32} \left(\frac{\mu}{W}\right)^{0.32} ,$$

so that the average friction factor is evaluated at $T_t = T_{t,av}$ in the viscosity term for the proper region, and D_h is given by the following expressions:

For the annular region:

$$D_h = \frac{D_s^2 - (D + 2t)^2}{D_s + (D + 2t)} = D_s\,(1 - \zeta)$$

For the heater tube:

$$D_h = D .$$

Entrance and discharge losses are to be added to the above pressure drop along the tube.

G. Tube Wall Thickness

From Joule's Law of electrical heating and the definition of electrical resistivity,

$$dR = \rho^* \frac{dx}{A}$$

where A is the cross-sectional area of the tube, one can derive the following relation:

$$1054\, Q_{tot} = \frac{\rho^*\, I^2}{\pi t(D + t)} , \qquad (24)$$

where Q_{tot} is the total heat per unit length transferred from the tube including radiation to the shield and forced convection to the helium inside and outside.

From (24) one obtains

$$(1 + 2t/D)^2 = 1 + \frac{(\rho^* I^2/1054)}{(\pi D^2/4) Q_{tot}}$$

or $$\frac{t}{D} = \frac{1}{2}\left\{\left[1 + \frac{(\rho^* I^2/1054}{(\pi D^2/4)(Q_{tot})}\right]^{\frac{1}{2}} - 1\right\} \qquad (25)$$

Thus the thickness required versus Q_{tot}, total heat transfer, for a given T_w and diameter D, can be determined quite easily. Once the relation between distance, x, and Q_{tot} is established, and the material is selected, the schedule of t, thickness, versus x, distance, can be completed. This is an iterative procedure as the heat transfer depends somewhat on the factor (1 + 2t/D).

VII. DISCUSSION

The heater tube should be as short as practical. Therefore the temperature differential between the wall and the gas stream should be as large as possible, which implies that the tube wall temperature should be constant at the practical maximum, and the basic equations have been derived and solved for this case in Section VI. The shortest possible length is then established by the smallest possible diameter which satisfies the maximum permissible pressure drop.

Figure 5 is a chart for the determination of the amount of power input to the gas to achieve ΔT_t = 2500° for various flow rates, and conversely the ΔT_t attainable at a 360-kilowatt input power

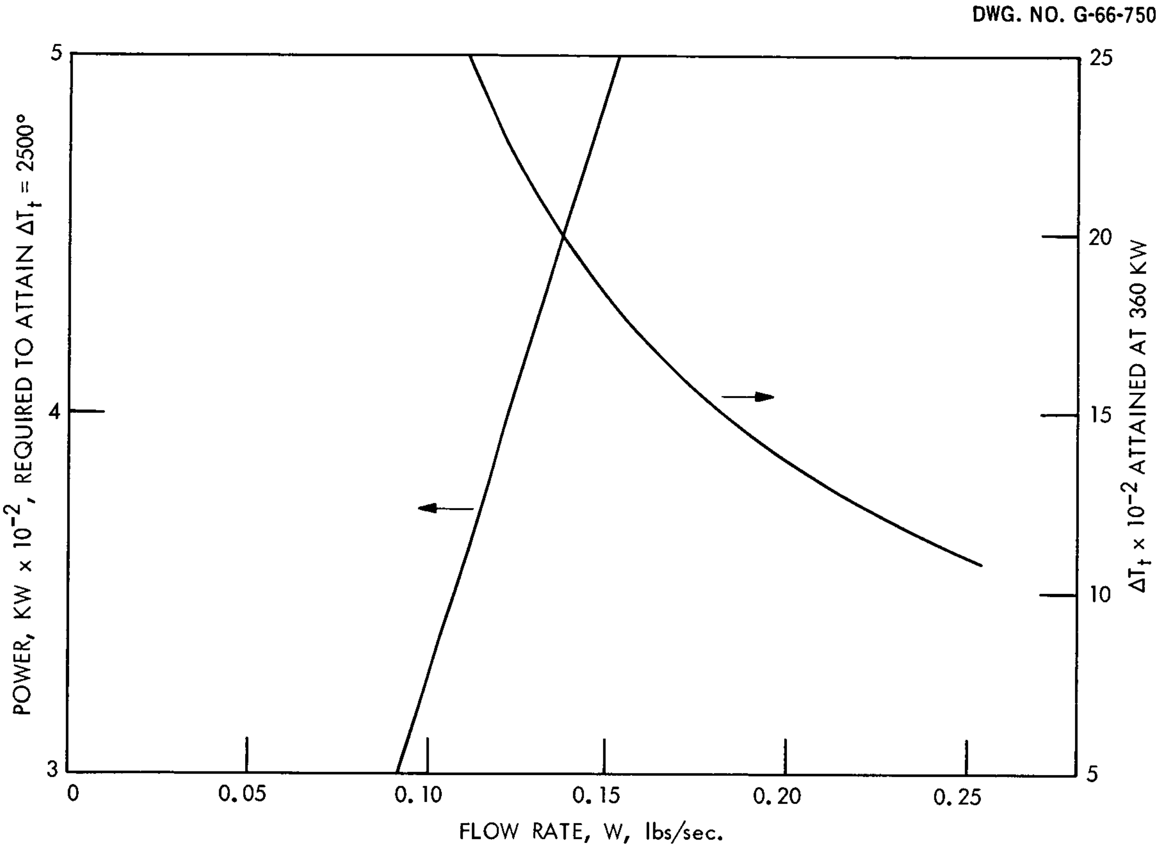

Figure 5

POWER AND FLOW REQUIREMENTS

level to the gas for various flow rates. From the chart the flow rate at $\Delta T_t = 2500°$ and power level of 360 kilowatts is 0.1102 lb per second.

Figure 6 is a carpet plot of $L/D^{0.8}$ for various tube inlet temperatures, $T_{t,1}$, and tube wall temperatures, T_w, as derived from (7) of Section IV. The outlet temperature is constant at $T_{t,2} = 2960°F$ with a flow rate $W = 0.1102$ lb/sec. This graph was constructed for a Prandtl number of 0.688. One can obtain values of $L/D^{0.8}$ for other Prandtl numbers by utilizing the relation

$$(L/D^{0.8})_{Pr} = (L/D^{0.8})_{0.688} \left(\frac{Pr}{0.688}\right)^{2/3} \tag{26}$$

The pressure drop is calculated at maximum flow conditions and minimum pressure level as this yields the largest pressure drop expected. The pressure drop through the heater tube is set at 5 psi. This is admittedly arbitrary, but the compressor that is contemplated for the test facility will be able to handle that drop (along with the other loop pressure drops) quite easily, and the higher the pressure drop the smaller the tube internal diameter and the lesser the tube length. The pressure drop consists of two parts, that due to the heat addition and that due to the friction. Define

$$\left(\frac{\Delta P_t}{P_t}\right)_h = a_2(T_{t,2} - T_{t,1}) = a_2(\Delta T_t') \tag{27}$$

$$\left(\frac{\Delta P_t}{P_t}\right)_F = b_2(fT_t)_{av,2} \tag{28}$$

Figure 7 is a graph of the heating drop along the inside of the tube for the case of maximum flow and minimum pressure level. From the heating pressure drop along, it is seen that the tube diameter will range between 0.55 - 0.70 inches for the given temperature differentials at the 5 psi restriction.

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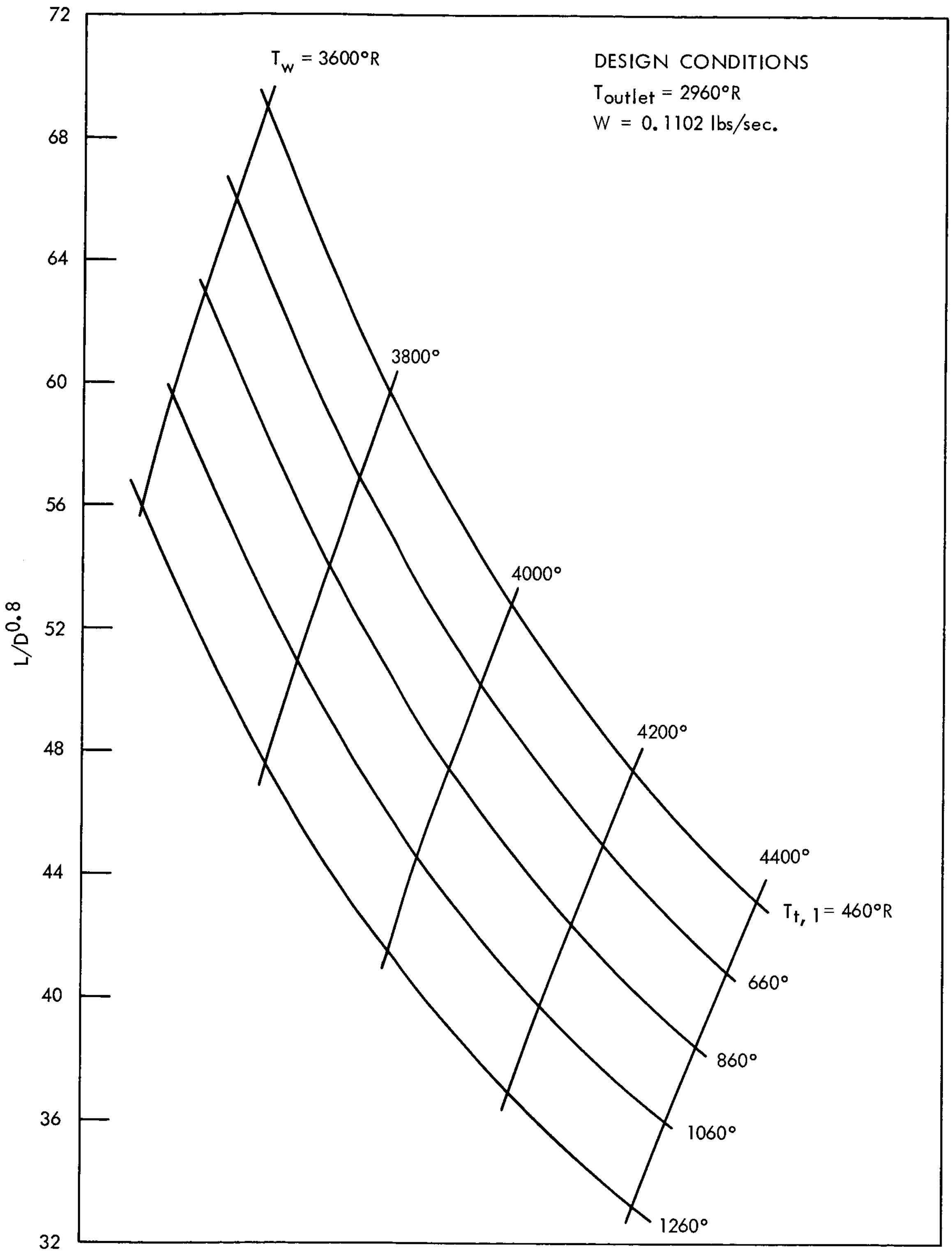

Figure 6

L/D$^{0.8}$ REQUIRED FOR DESIGN CONDITIONS

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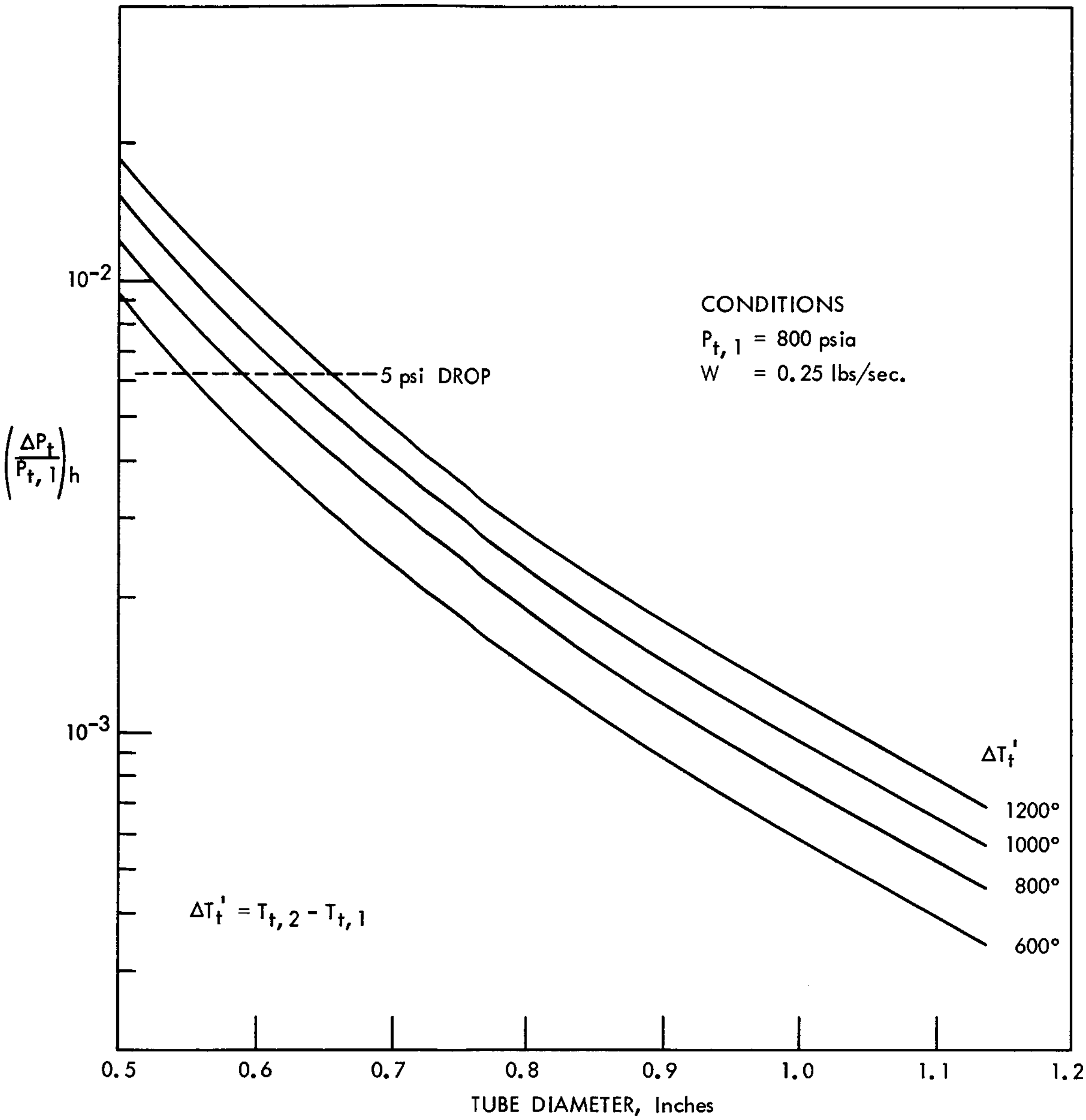

Figure 7

PRESSURE DROP DUE TO HEATING EFFECT

Figure 8 is a graph of the frictional drop per unit $(L/D^{0.8})$ for the same condition as Figure 7. In the calculation of the average temperature, a tube outlet temperature of 2460°R (consistent with 360 kw and an inlet temperature of 1460°R) was assumed. The average temperature is not strongly dependent upon the tube wall temperature at a given temperature rise. Therefore, the calculations were performed for a tube wall temperature of 3600°R. From the graph it is seen that the frictional pressure drop is but slightly influenced by the temperature rise $(\Delta T_t')$ across the tube per se.

Figure 9 is a plot of entire stagnation (total) pressure drop versus heater tube diameter for a loop pressure of 800 psia. At the maximum $L/D^{0.8}$ ratios the temperature rise across the tube is not influential on the total pressure drop. From the graph it can be seen that the diameter must lie somewhere between 0.80 - 0.95 inches. For conservatism, it has been decided to operate the tube at a design wall temperature of 4000°R and to set the internal diameter of the heater tube at 0.875 inches. This extra increase in diameter is to allow for the pressure drops at the tube entrance and exit and along the outside annular space. The three additional curves (dashed) drawn on Figure 9 are consistent with the $L/D^{0.8}$ requirements for three different tube wall temperatures, the $T_{t,1}$ agreeing approximately with the match curve of Figure 14. It should be noted that $\Delta T_t' = 1000$ is approximately consistent with 360 kw and W = 0.25 lb/sec.

Figure 10 is a plot of the heat transfer to the ambient for variable $T_{c,i}$, the inner casing temperature, at constant ambient temperature T_a. Also plotted on this graph are curves of constant outer shield temperature. These were calculated considering heat conduction across a stagnant helium film along with the thermal radiation transfer. The assumed values of the diameters and emissivities are listed on the graph. The maximum allowable

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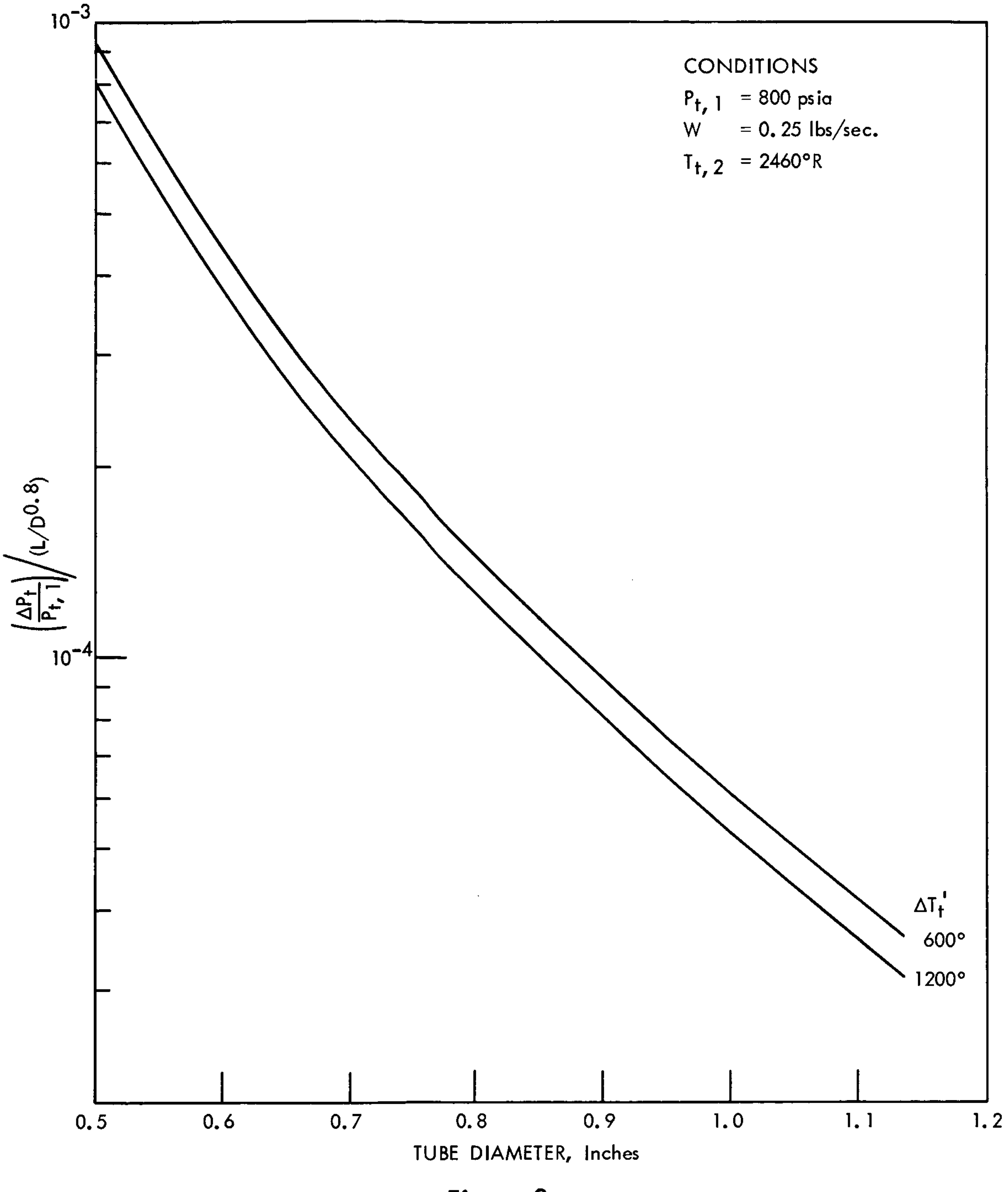

Figure 8

FRICTIONAL PRESSURE DROP

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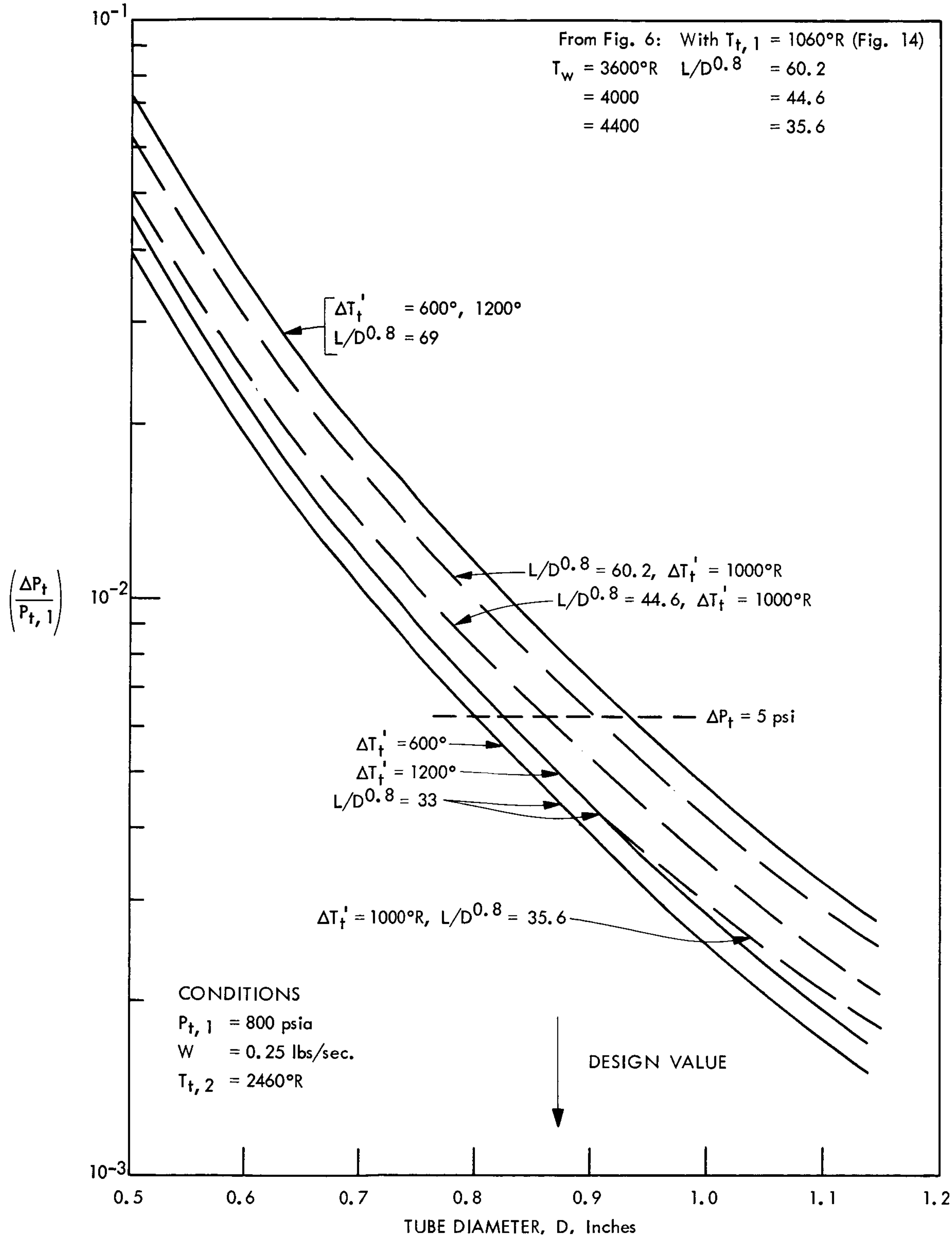

Figure 9

PRESSURE DROP IN HEATER TUBE

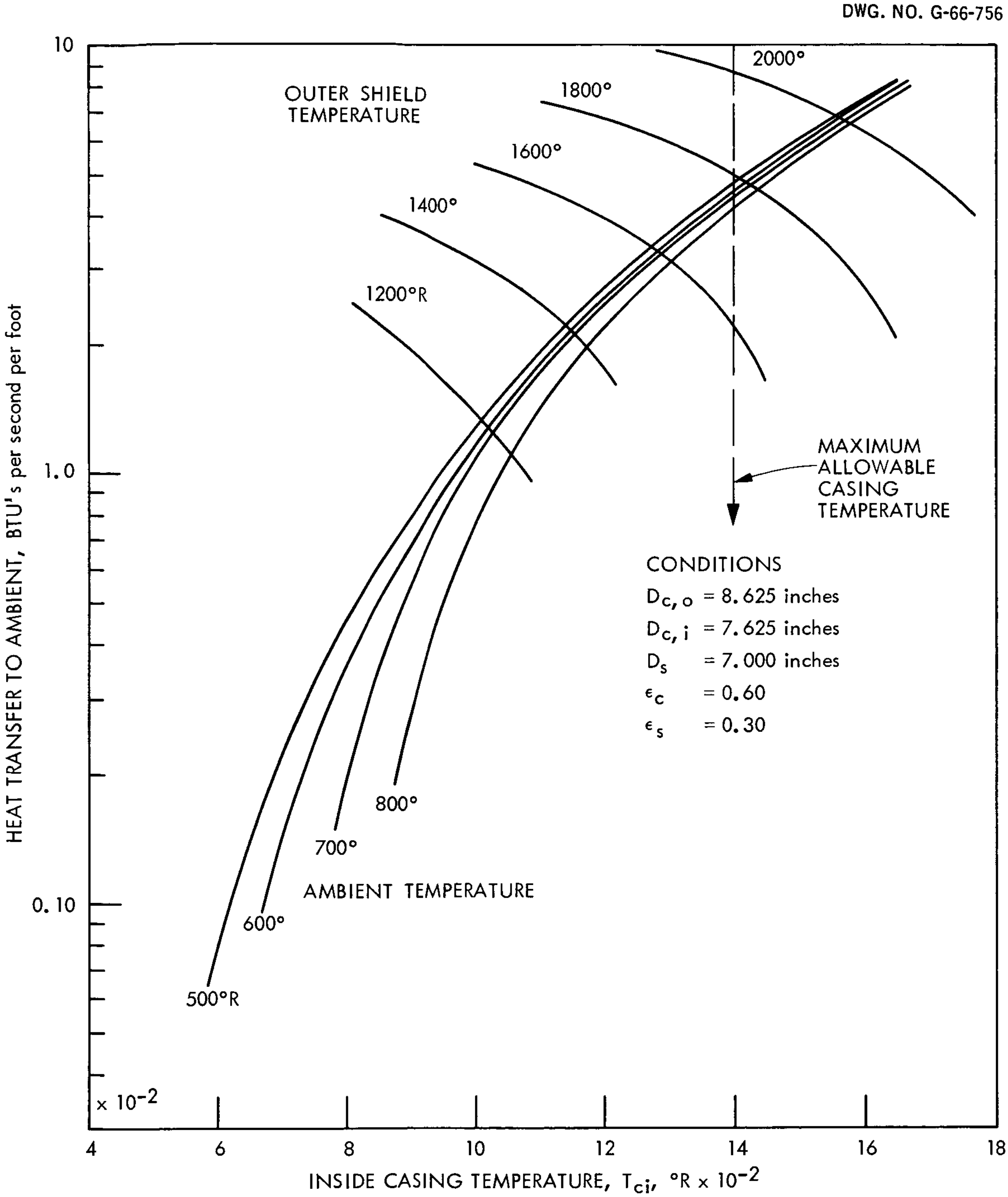

Figure 10
HEAT TRANSFER TO AMBIENT

casing temperature, for reasons of safety, is 1400°R as shown. This effectively restricts the heat transfer to the ambient to 4 - 5 Btu's per second per foot of length, on the average. The outer shield temperature is then restricted to 1750°R.

Next, the average Q_s transfer is determined for assumed values of the inner shield temperature. The innermost shield is placed at $D_{s,i}$ = 0.5 ft. From relation (2), averaged over the length L, and knowing that $(T_s - T_t)$ is exponential in nature, one obtains, to good approximation.

$$\overline{Q}_s = 0.00467 \, (\mu_{f,s} \times 10^5)_{av}^{0.2} \, (T_s - T_t)_{av} \tag{29}$$

where

$$(T_s - T_t)_{av} = \frac{T_{t,1} - T_{t,o}}{\ln\left[\dfrac{T_{s,av} - T_{t,o}}{T_{s,av} - T_{t,1}}\right]} \tag{30}$$

A value has been chosen for ζ_{av}^2 = 0.10 and W = 0.1102 lb/sec. Figure 11 shows a plot of $\overline{Q}_s$ for assumed values of $T_{t,1}$. Cross-plotted on the graph are values of temperature T_w for an assumed value of Q_{amb} = 4 Btu's per foot. This utilizes thermal emissivity values as given in Figure 12.

From Figure 11, together with (20), one can determine the average equilibrium shield temperature $T_{s,1}$ for constant T_w versus the heater tube inlet temperature $T_{t,1}$. This is shown in Figure 13. With these equilibrium temperatures determined, the length, in feet, can be determined in two ways for an assumed value of $T_{t,1}$. One length is determined from the inside $L/D^{0.8}$ ratio given in Figure 6 and the chosen D = 0.875 inches and the other is given by a rearrangement of relation (10) to determine L for the given $D_{s,i}$ = 0.5 ft. Three wall temperature cases were calculated and are shown in Figure 14. The match curve of $T_{t,1}$ is quite constant,

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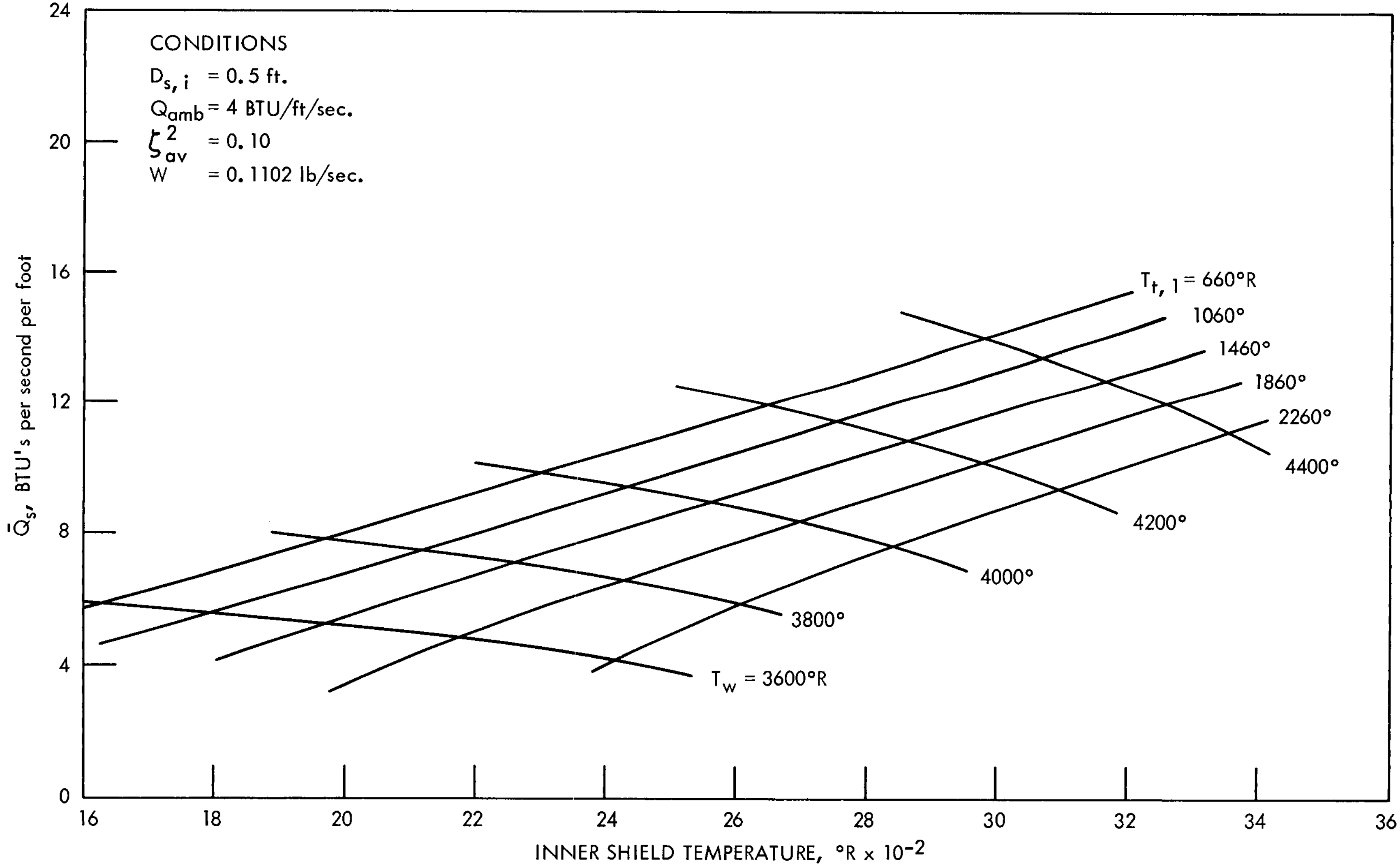

Figure 11

HEAT TRANSFER TO GAS FROM INNER RADIATION SHIELD varying only from 1020° - 1080°R. This match curve has also been plotted on Figure 13, so that the inner shield temperatures can be easily determined.

The last determinations concern the number of shields required and the schedule of thickness versus distance.

The maximum number of shields is determined by the maximum tube wall temperature of 4400°R. From Figure 13 the first shield temperature is 3060°R, while from Figure 10 the outer shield temperature is 1750°R. With these values $Q^* = 11.1$, and since $Q = Q_{amb}$ one obtains

$$N = 1 + \frac{11.1}{4} = 4.$$

Thus, to ensure that the casing temperature does not exceed allowable values at the 1500 psia level, one must install four (4) radiation shields. These shields are installed with the innermost one no smaller than six (6) inches in diameter.

Calculations of t versus x (see Table I) showed that the variation with distance is essentially linear. The inlet and exit thicknesses are given in Table II. The values of the thicknesses are in inches.

Discussions with the metallurgical people and mechanical design analysts determined that heater tube operation at 4400°R is feasible. For conservatism, however, a wall temperature of 4000°R was used as the final design parameter. The length of the tube was increased by 10% to compensate for the actual non-constant wall temperatures at the tube ends and any cooling effect that is present before the gas is delivered to the heat exchanger proper.

From Figure 14 the necessary length is 5.5 feet, or, with the 10% increase, the actual length is 6.05 feet.

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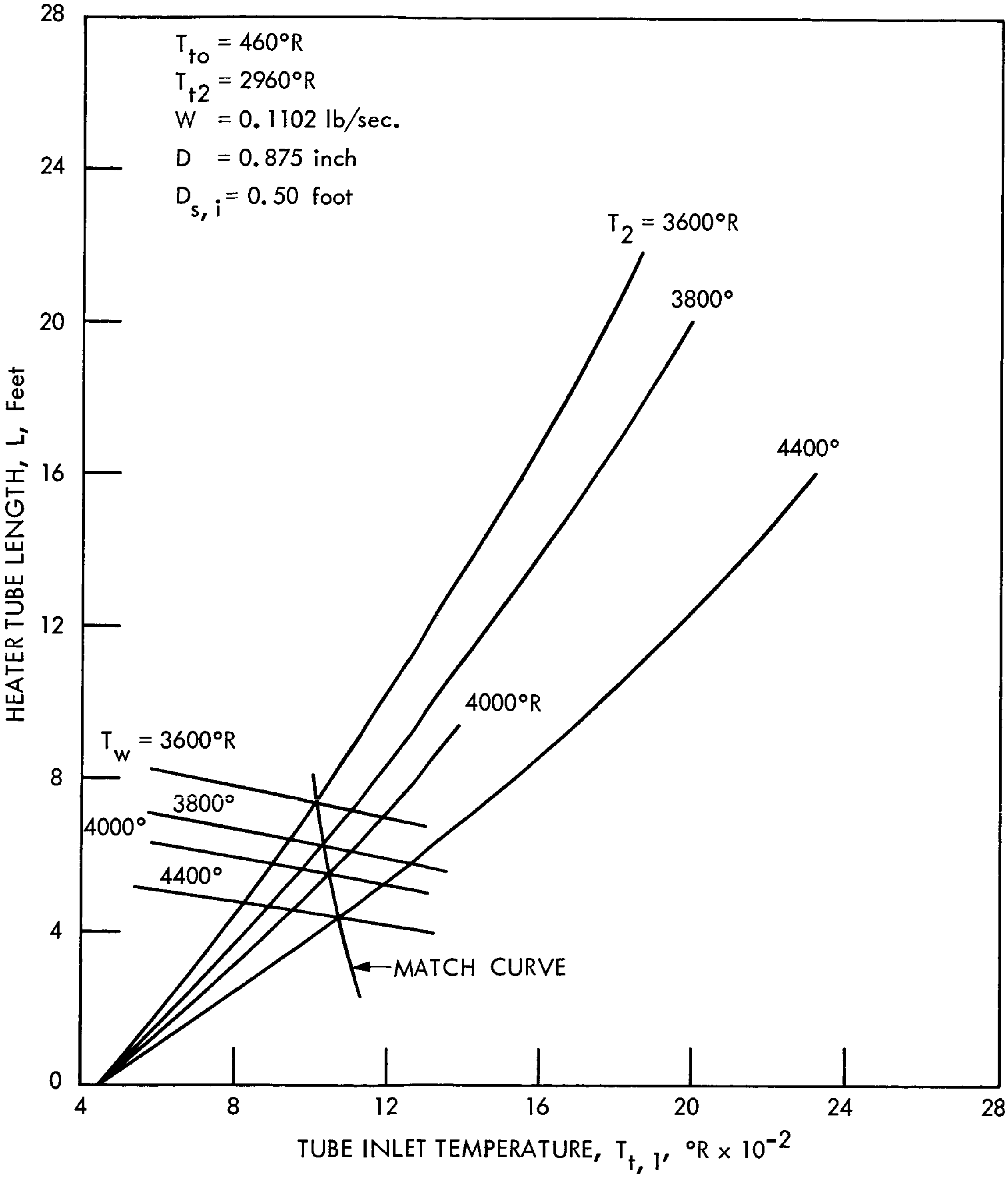

Figure 14

HEAT - LENGTH BALANCES

TABLE I

THICKNESS VERSUS DISTANCE

T_w = 4000 °R, T_s = 2830°R, I = 15,000 amps

$\frac{x}{L}$	t, inches	Δt
0	0.388	
0.1	0.369	0.019
0.2	0.350	0.019
0.3	0.331	0.019
0.4	0.313	0.018
0.5	0.294	0.019
0.6	0.276	0.018
0.7	0.259	0.017
0.8	0.242	0.017
0.9	0.225	0.017
1.0	0.210	0.015

TABLE II

I = 15,000 amperes

T_w, °R	t_{inlet}	t_{exit}
3600	0.217	0.475
3800	0.212	0.423
4000	0.210	0.388
4200	0.205	0.358
4400	0.203	0.338

VIII. SUMMARY OF DESIGN DIMENSIONS

Length of tube	73 inches
Inside diameter	0.875 inches
Entrance tube wall thickness	0.210 inches
Exit tube wall thickness	0.388 inches
Inner shield diameter	6 inches
Outer shield diameter	7 inches
Number of radiation shields	4
Casing inner diameter	7.625 inches
Casing outer diameter	8.625 inches

IX. OFF-DESIGN OPERATION

It is desired to determine whether the tube as designed will be able to yield helium gas at 2260°R exit temperature, at a 460°R inlet temperature, a flow rate of 0.071 pounds per second and a pressure level of 1500 psia. It is also desired to estimate the time required to heat the tube to operating temperature with stagnant helium gas in containment.

a) Time to heat tube:

The rate of heat addition to the heater tube is given by two expressions, viz,

$$\frac{dQ}{d\tau} = \frac{RI^2}{1054} \text{ and } \frac{dQ}{d\tau} = MC_m \frac{dT_w}{d\tau}$$

where τ = time in seconds

M = mass of tube, lb

C_m = thermal capacity of tube metal

R = total resistance of tube, ohms.

Now from

$$R = \int_o^L \frac{\rho^* dx}{\pi t(D + t)} = \frac{\rho^*}{\pi D} \int_o^L \left(\frac{1}{t} - \frac{1}{D + t}\right) dx \quad , \qquad (31)$$

and the relation

$$t = t_o + (t_1 - t_o)\xi \tag{32}$$

one can determine the total tube resistance by integration to be

$$R = \frac{L\rho^*}{\pi D(t_1 - t_o)} \ln\left[\frac{t_1(D + t_o)}{t_o(D + t_1)}\right] \quad . \tag{33}$$

The mass of the tube is given by

Metal density × tube volume = M =

$$\gamma \frac{\pi L}{12}\left[(D+2t_o)^2 + (D+2t_o)(D+2t_1) + (D+2t_1)^2 - 3D^2\right] \tag{34}$$

with γ = metal density, lb/ft^3.

The time to heat from $T_{w,o}$ to $T_{w,1}$ is given by, ignoring the variability of C_m with metal temperature,

$$\tau = \frac{1054MC_m}{I^2(R/\rho^*)} \int_{T_{w,o}}^{T_{w,1}} \frac{dT_w}{\rho^*} \quad . \tag{35}$$

A linear relationship for the electrical resistivity is assumed as

$$\rho^* \times 10^6 = 0.17 + 0.50\left(\frac{T_w - 460}{1000}\right)$$

and the indicated integration of (35) can then be performed as

$$\tau = \frac{2108MC_m \times 10^3}{\left(\frac{I}{1000}\right)^2 (R/\rho^*)} \ln\left[\frac{\rho^*(T_{w,1})}{\rho^*(T_{w,o})}\right], \text{ sec.} \tag{37}$$

Figure 15 is a plot of (37). This time to heat ignores the three-dimensional effects and the cooling effect of the helium. It should provide a lower bound to the time necessary to reach any given temperature starting from room temperature of 530°R. As can be seen, the time is sufficiently

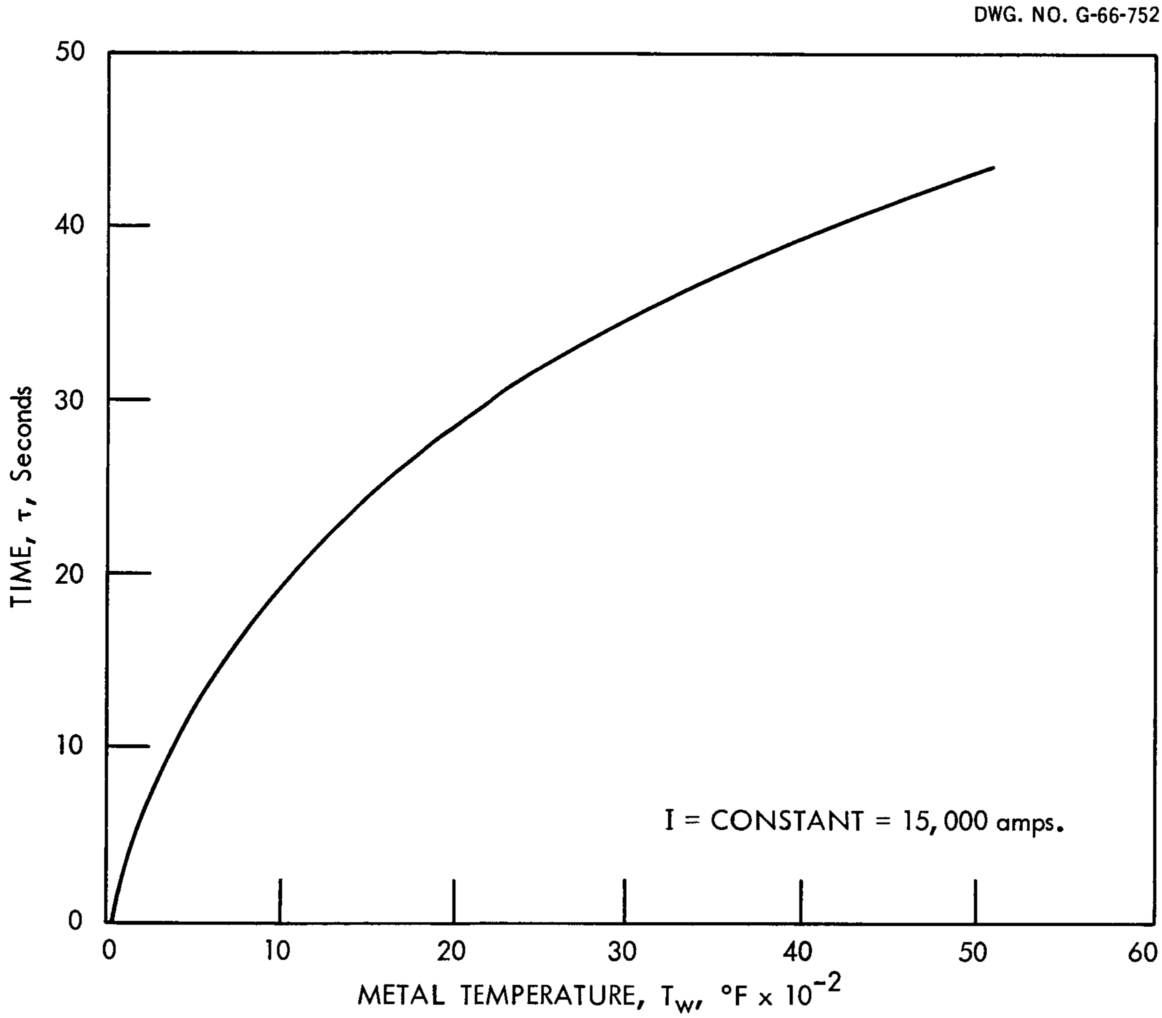

Figure 15

TIME REQUIRED TO PREHEAT HEATER TUBE

long so that one can pre-heat the tube before opening the helium system to the heater exchanger. This will allow the heat exchanger to be tested under thermal shock conditions.

One can, of course, lengthen the time to heat by running the electrical system so that it does not produce maximum amperage during the time of heating.

During the initial phase of heat exchanger testing the heater tube will operate at a maximum amperage of 13,500. It must deliver helium at a flow rate of 0.071 pounds per second with an exit temperature of 2260°R. The power supplied to the gas at these conditions is 167.2 kw. An upper estimate to the tube wall temperature can be calculated by assuming that all the heat is transferred to the gas inside the tube. Using the given dimensions of the tube (neglecting the 10% increase in length) and considering that the tube wall temperature will be practically constant, one determines, by iteration and rearrangement of (7), that the tube wall temperature will be not greater than 3560°R. The total resistance of the tube at this temperature is 1.244×10^{-3} ohms. The power that can be supplied with a current of 13,500 amperes is 227 kw. This is ample and the actual operating amperage will be smaller than 13,500.

X. References

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Report Number: NASA CR-72045
K-D-1870

Title: SUMMARY REPORT - HEAT EXCHANGER TEST EXPERIMENTAL APPARATUS
SECTION I - HELIUM HEATER
SECTION II - HELIUM CIRCULATOR

AUTHORS: SECTION II - HELIUM CIRCULATOR
D. W. Burton
J. O. McCullough

ABSTRACT

The circulator described in this section of the report is intended to provide the necessary helium flow through the experimental test loop when the facility is not being supplied from a helium trailer. This circulator is of a type that has seen extensive service in nuclear reactor testing using helium and is called a peripheral compressor

The report deals with the performance, construction materials and design revisions necessary to utilize this circulator in the heat exchanger test facility. It has been determined that this machine is an inexpensive, reliable circulator and will provide the necessary helium flow for the test facility.

HEAT EXCHANGER TEST EXPERIMENTAL APPARATUS

HELIUM CIRCULATOR

I. OBJECTIVES

The helium circulator is intended to provide the necessary test loop helium circulation when the facility is not being operated in a trailer-mode. This circulator is to be an integral part of a test facility in which various helium-to-air heat exchangers can be evaluated.

This report deals with the aerodynamic performance, materials of construction, and design revisions to an already existing circulator. Since the operating characteristics of the loop are yet to be determined, a short discussion of ways in which the aerodynamic performance of the circulator can be changed is also included.

II. CONCLUSIONS

The helium circulator proposed, a modification of an existing type compressor, will provide the necessary gas circulation in the test facility at a reasonable initial cost.

III. RECOMMENDATIONS

It is recommended that the circulator as proposed in this report be used in the loop facility for use as stated in the OBJECTIVES.

IV. DATA GIVEN AND NOMENCLATURE

A. The purpose of the circulator is to provide the mass flow of helium required through the test heat exchanger when the test facility is not operating in the trailer-mode.[5][6]*

B. The proposed location is shown in Figure 1, a schematic of the test loop facility, and it is seen that a cooler is provided before the circulator to ensure that the entrance gas temperature does not exceed 700°F under maximum test conditions, with a nominal 600°F operating temperature.

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* See References, page 57.

C. The maximum pressure level is to be 1500 psig.

D. For design purposes the total loop pressure drop was assumed to be a maximum of 25 psi at a weight flow of 0.25 pounds per second.

E. Maintenance-free life of three to five thousand hours is required.

V. SELECTION OF CIRCULATOR

The initial capital cost of the circulator was of prime importance, so a review of existing machines was made. It was determined that a peripheral compressor, designated HECT-2, had the required aerodynamic performance, but did not meet the pressure containment requirements. The aerodynamic test data for this circulator, developed by the Oak Ridge National Laboratory, are available in references (1) and (2) and are reproduced here as Figure 2. Since the original mechanical design was for lower pressure level operation (400 psig), a redesign was carried through to enable the machine to operate safely at the 1500 psig level.

VI. MECHANICAL REDESIGN OF HECT-2 CIRCULATOR

The Helium Circulator is a redesigned model of the original HECT-2 gas compressor (see Figure 5) which was intended for applications in nuclear reactor experimental in-pile loops using high purity gas. The HECT-2 was designed to meet some very stringent operating requirements, including high rotational speed (12,000 rpm), high temperature operation (600° - 1000°F), high degree of operational cleanliness, and a long maintenance-free life (three to five thousand hours of continuous operation). A comparable life can be expected for the Helium Circulator. This type of compressor is completely enclosed by a pressure shell which makes it difficult to detect any trouble which might be developing, so this maintenance-free life is of great operational importance. The Heat Exchanger Test Helium Circulator will have the same operating conditions as the HECT-2, with the exceptions of the absence of a radioactive atmosphere and an increase in maximum operating pressure. This pressure has been increased from 400 psig to 1500 psig. This increase dictated several changes in the existing design of the compressor and pressure vessel. The compressor is mounted within the pressure vessel, and all internal and external compressor components, with the exception of the coolant cavities, are at the same pressure. Therefore these components will not be subjected to pressure differential stresses. Design changes to the compressor and pressure vessel are listed as follows:

(1) The pressure vessel was completely redesigned in accordance with Section VIII of the ASME Unfired Pressure Vessel Code. Design conditions were 1500 psig at 300°F using type 304 stainless steel. An allowable stress of 15,000 psi was obtained from table UHA-23 in the ASME Pressure Vessel Code at these conditions.

(2) The original HECT-2 compressor was constructed almost entirely of inconel due to the fact that it was to be employed in a nuclear application with high operating temperatures. With the exception of the impeller which will remain inconel, type 304 stainless steel was substituted in the Helium Circulator due to cost, weldability and absence of any radioactive atmosphere.

(3) The shell thickness was increased from 1/8 to 7/16 due to the increased pressure differential at the coolant cavities.

(4) The coolant cavity and spirals were machined from inconel in the HECT-2 which proved to be a costly operation. By using stainless steel these spirals can be formed from strip material and tack welded to the end plate. This may permit a small amount of cross flow of coolant but this will not be enough to affect the total flow pattern.

(5) Experience with the HECT-2 has shown that the "cold finger" coolant passage within the hollow shaft can be eliminated. Its purpose was to cool the bearings and seals but it was

discovered that sufficient cooling to these areas could be provided by the end spirals. The shaft will remain hollow to restrict heat transfer from the impeller base to the bearings.

(6) The bearing grease seals were redesigned so as to eliminate the deep groove. This groove was to provide a grease reservoir which proved to be unnecessary due to the lack of elevated temperatures in this area.

(7) Thermocouples were used in the HECT-2 compressor to permit examination of temperatures at the bearings, rotor and shell while the compressor was in operation. These thermocouples may not be necessary in the Helium Circulator. If not, it will be an easy matter to plug the ports in the compressor shell and pressure vessel.

(8) Other features such as the electrical connectors and internal venting ports near the impeller were not modified. If it is necessary to eliminate or revise some of these features, the drawings will be revised later, prior to procurement of the compressor.[7]

VII. AERODYNAMIC PERFORMANCE

At a system pressure level of 1500 psia and a circulator inlet temperature of 600°F (normal operating value), the loop pressure drop of 25 psi and flow rate of 0.25 lb/sec convert into the following requirements for the circulator:

$$\text{Head} = 6820 \text{ ft.}$$

$$\text{Volume Flow} = 28.4 \text{ cfm.}$$

If a square law is assumed for the loop resistance, (i.e. $\Delta P \sim \rho Q^2$, where Q is the volume flow at inlet conditions) the curve labeled normal operation in Figure 2 can be obtained from the above data. While the performance data for the HECT-2 were obtained at 400 psia, this performance (with the exception of power input) is quite insensitive to pressure level. The power requirement, however, is directly proportional to the density, so that the Helium Circulator power requirement can be obtained by scaling the proper test power input curve with the density ratio (or in this case the ratio of the operating pressure to the test pressure, 1500/400, since the temperature of operation will be identical to the test temperature of 600°F).

From Figure 2 the compressor will operate at approximately 8300 rpm to achieve the desired head and flow requirements that are listed above. The power requirement at 400 psia is 3.65 Kw for a speed of 8300 rpm. The Helium Circulator power requirement will then be

$$\text{power (Kw)} = 3.65 \; \frac{1500}{400} = 13.7 \text{ Kw}$$

$$= 18.3 \text{ horsepower,}$$

which is within the capability of the present motor (a nominal 20 hp.).

The test facility flow resistance line varies little with pressure level, but does vary with temperature. Before a final decision is made to construct the circulator, a system analysis must be made to determine more carefully the conditions under which the circulator will be expected to operate.

The power requirement of the compressor motor, as the compressor is operated along the system line, can be obtained from the experimental power input data given on Figure 2. The expression

$$\text{Power input (Kw)} = 4.38 \times 10^{-4} \; (Q)^{2.7}$$

fits the data at a pressure level of 400 psia, when the compressor follows the system line shown (the compressor speed varies along the system curve). At pressure levels other than 400 psia the motor power is obtained by ratio as

$$hp = \text{horsepower} = \frac{1.341(4.38\times10^{-4})}{400} \; P\,Q^{2.7}$$

$$= 1.468 \times 10^{-6} \; P\,Q^{2.7}$$

where P is the pressure level, psia.

Since there is a nominal rating of 20 hp for the motor as installed in the HECT-2 compressor, the maximum loop pressure for safe motor operation along the system resistance curve can be obtained from the relation

$$P_{max} = \frac{13.6\times10^{6}}{Q^{2.7}}, \text{ psia.}$$

where the motor horsepower has been set at the nominal rating of 20 hp.

Figure 3 shows a graph of this maximum pressure as a function of the mass flow along the system line. It is seen that the loop pressure must be restricted for flows below the value of 0.25 pounds per second, or else the motor will be seriously overloaded.

If this performance capability is not sufficient to satisfy the testing requirements, the impeller and casing can be modified, relatively inexpensively, to give better performance. Figure 4 (Figure 3 of reference (3)) is a comparison of several peripheral compressors operating on helium. It is seen that the HECT-2 performance lies below that of the others, especially the ORGDP-1 (a machine with the same diameter impeller as the HECT-2). The blading parameters and the channel parameters can be changed quite inexpensively if further system analysis shows that the HECT-2 compressor cannot meet the aerodynamic performance. From the figure one can see that, at the same capacity coefficient λ, one can obtain at least twice the head coefficient ψ as the result of minor modifications. This means that the machine could be operated at a lower rotational speed to achieve that same desired head-flow characteristic. Since power is approximately proportional to the rpm raised to the third power, it is probable that the present motor would suffice for all desired operating test points.

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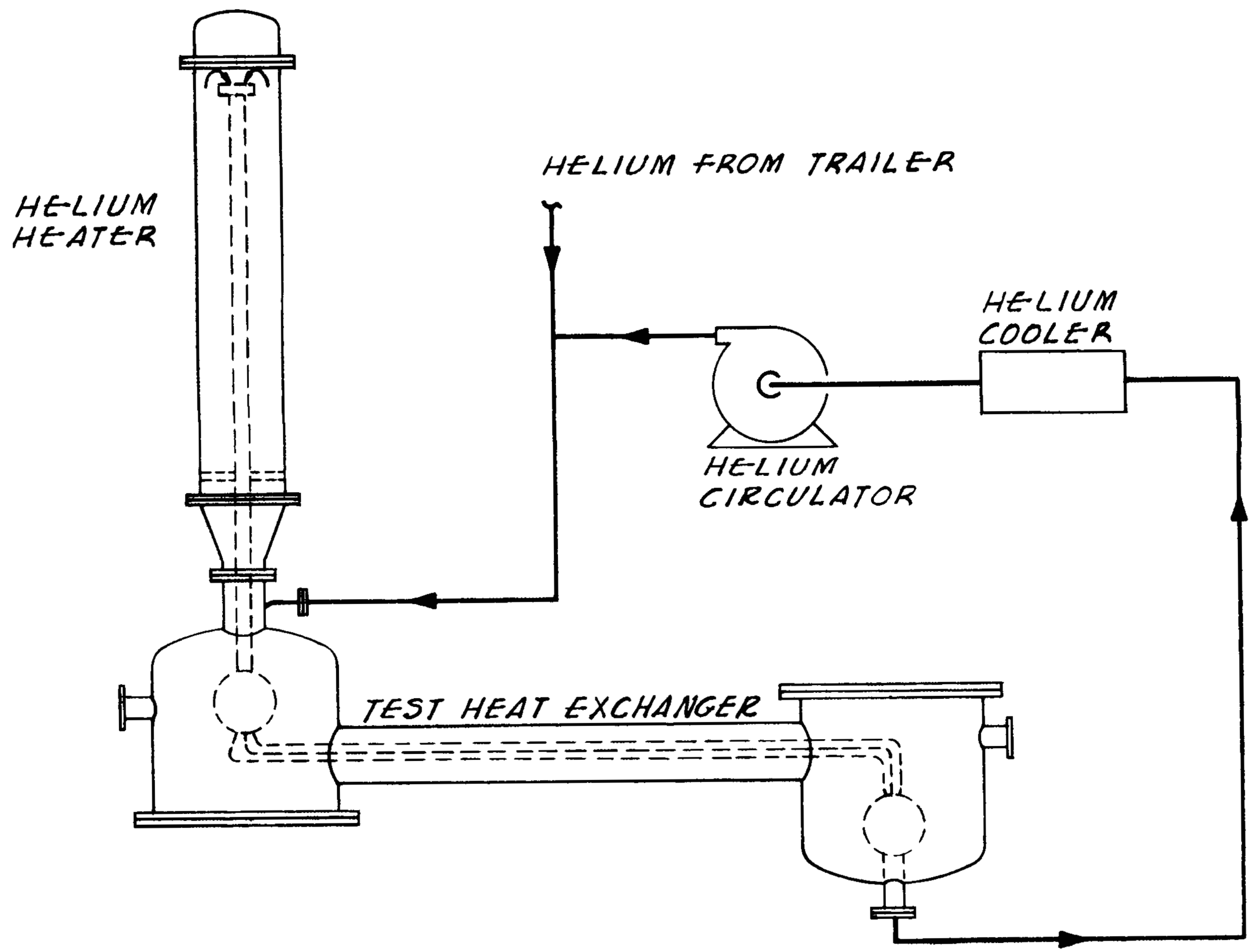

Figure 1
HELIUM LOOP SCHEMATIC

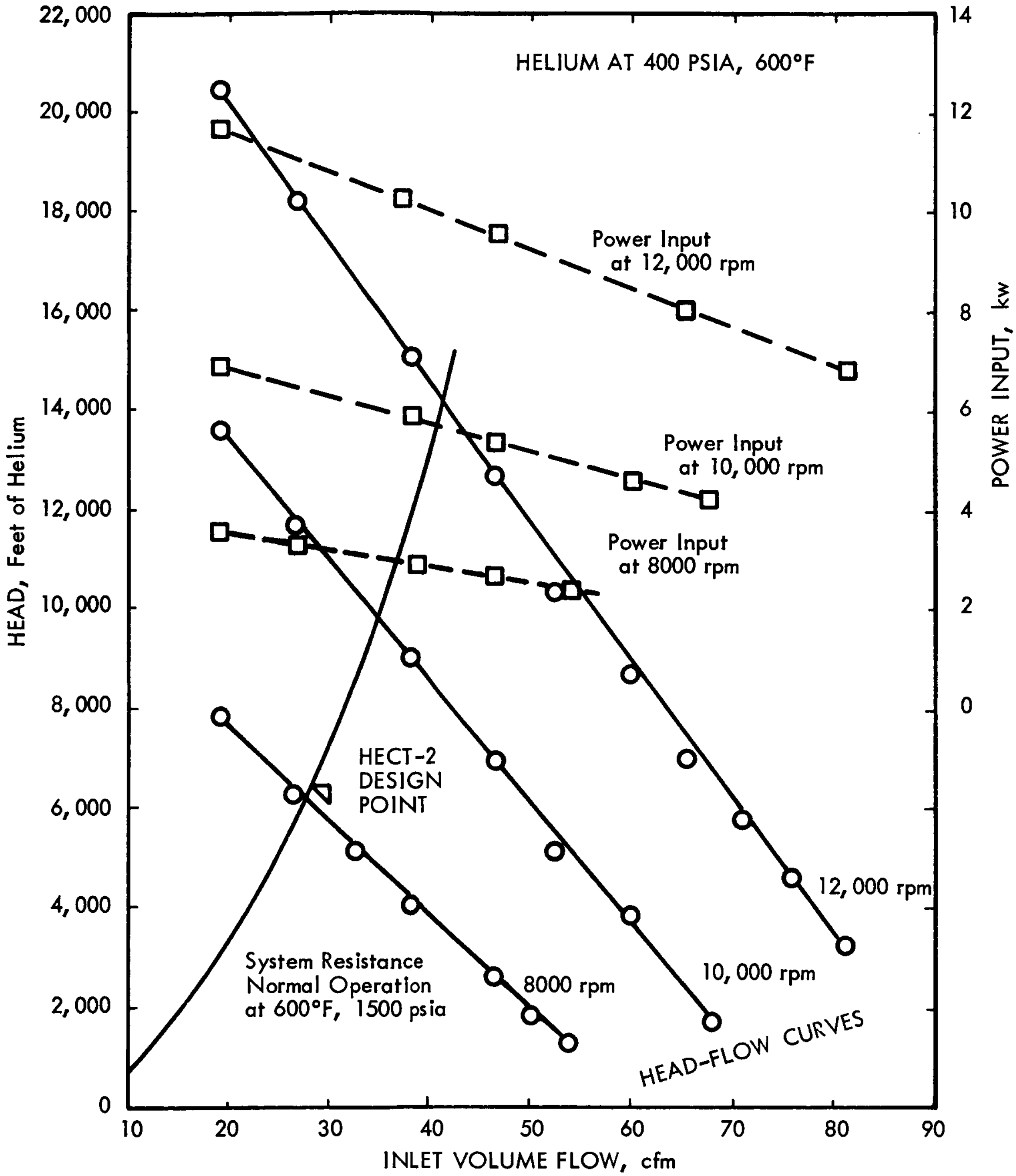

Figure 2

AERODYNAMIC PERFORMANCE OF HECT-2 CIRCULATOR

DWG. NO. G-66-746

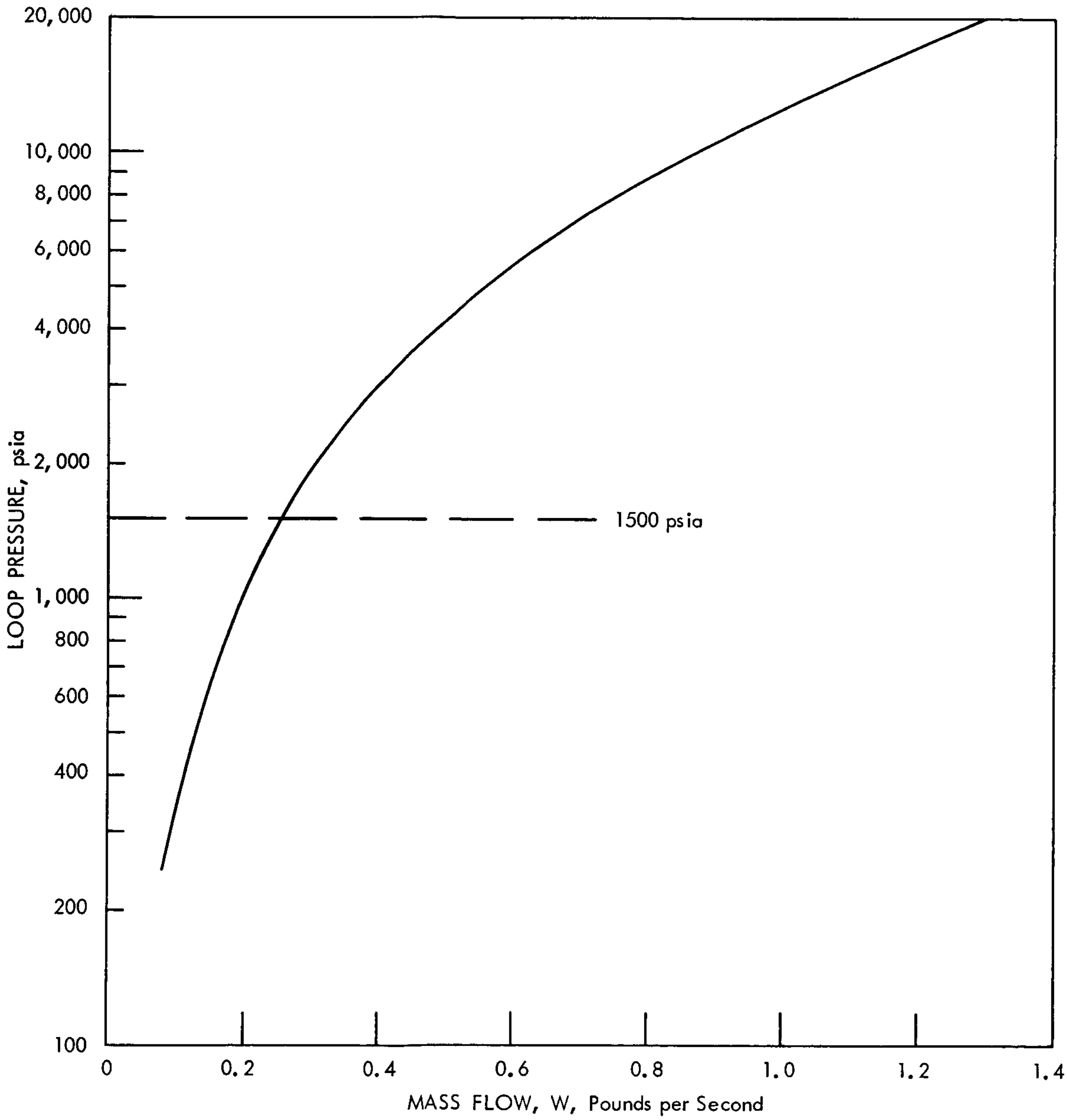

Figure 3

LOOP PRESSURE REQUIRED FOR 20 HP OPERATION
ALONG SYSTEM LINE OF FIGURE 2

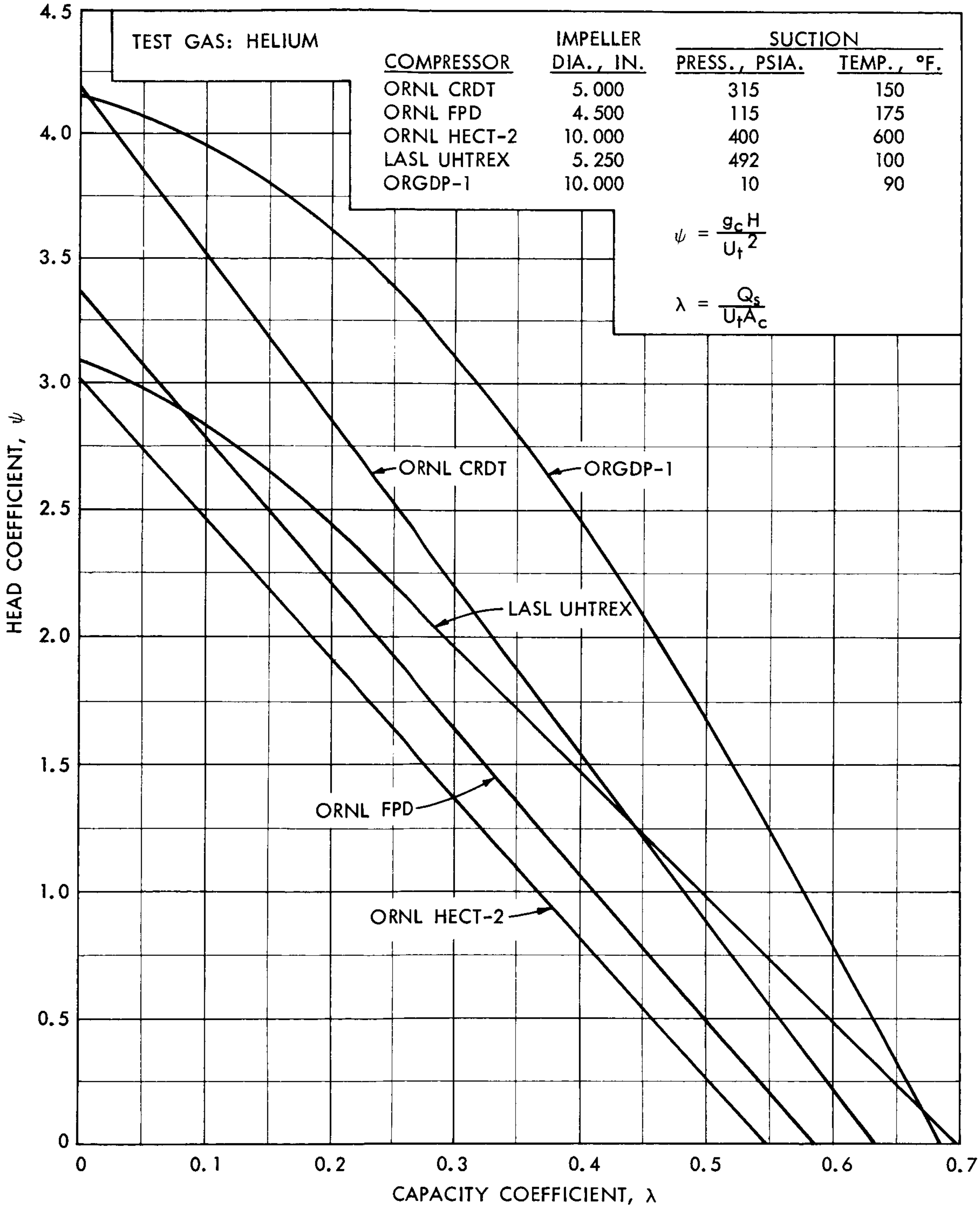

Figure 4

COMPARISON OF SEVERAL PERIPHERAL COMPRESSORS

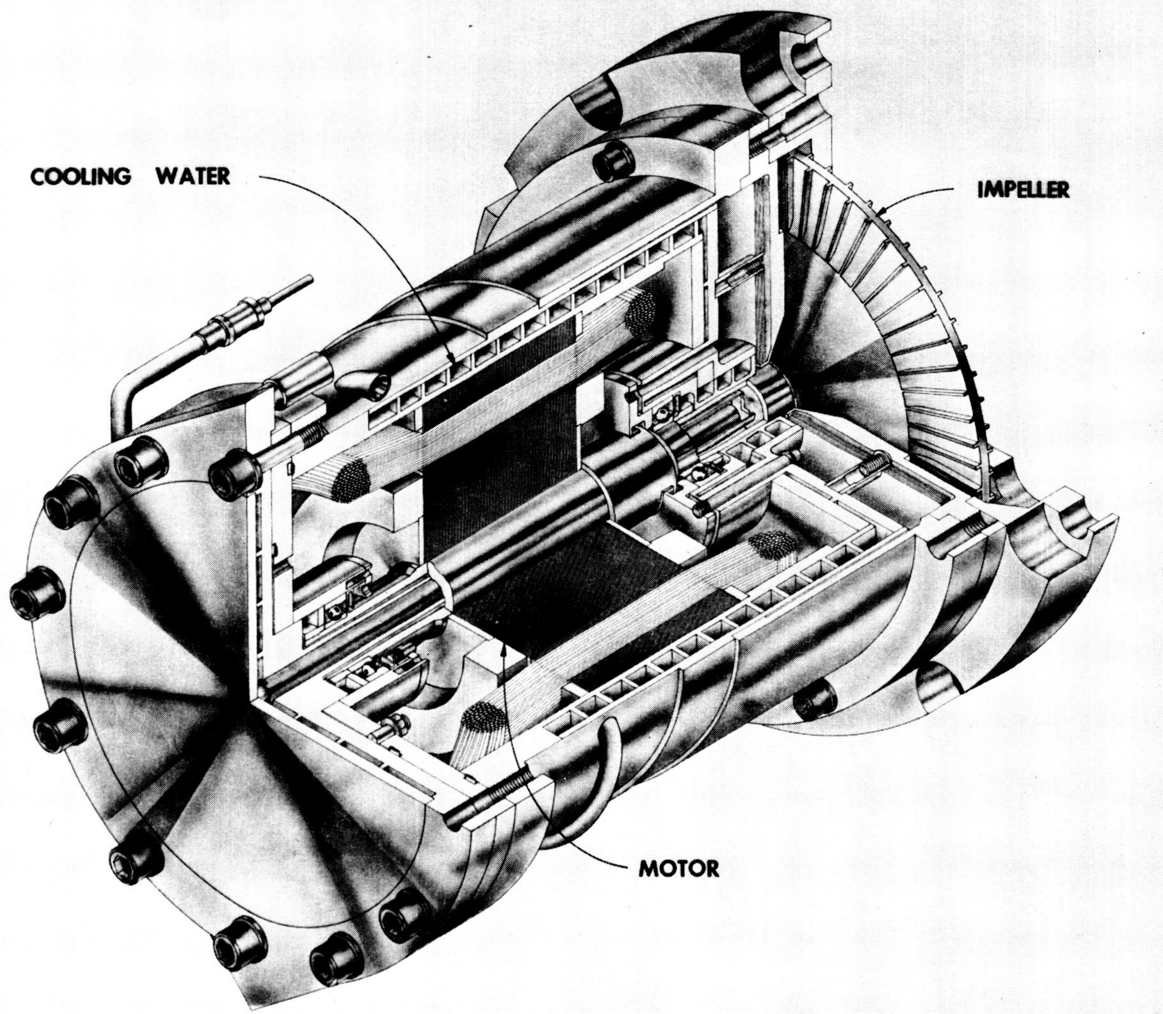

Figure 5
HECT-2 CIRCULATOR

Title: SUMMARY REPORT - HEAT EXCHANGER
EXPERIMENTAL APPARATUS
SECTION I - HELIUM HEATER
SECTION II - HELIUM CIRCULATOR

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