

THE UNIVERSITY OF MICHIGAN
COLLEGE OF ENGINEERING
Department of Mechanical Engineering
The Heat Transfer and Thermodynamics Laboratory

Progress Report No. 15

PRESSURIZATION OF LIQUID OXYGEN CONTAINERS

J. A. Clark

W. A. Warren

H. Merte, Jr.

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ABSTRACT

A new tank design which permits a more accurate determination of residual mass is described. This design also allows for measurement of liquid-vapor hold-up in a boiling system using a floating piston.

A theoretical analysis is given of the heat-mass transfer between a pressurizing gas and an interface, including the transient build-up and decay of a condensate film. It is shown that the significant thermal property of the interface which governs the residence time of a condensate film is $k\rho c_p$. Several different types of interfaces are compared and it is found that during a 2-minute pressurized-discharge process a condensate film may be expected to build up on a liquid nitrogen interface but to exist for approximately 0.65 seconds only on a styrofoam interface.

Further results are given on the study of pool boiling in an accelerating system.

NOMENCLATURE

A	- Area ft ²
c_p	- Specific heat, Btu/lbm-F
e	- Energy, Btu/lbm
$f'(\theta)$	- See Eq. (14), °F/ft
$f'_o(\theta)$	- See Eq. (15), °F/ft
h	- Enthalpy, Btu/lbm
\bar{h}	- Heat-transfer coefficient, Btu/hr-ft ² -F
h_{fg}	- Latent heat of vaporization, Btu/lbm
m	- Mass, lbm
P_x	- Rate of work transfer, Btu/hr
p	- Pressure, psia
q	- Rate of heat transfer, Btu/hr
t	- Temperature, F
t^*	- Saturation temperature after pressurization, F
μ	- Internal energy, Btu/lbm
V	- Flow velocity, ft/hr
v	- Specific volume, ft ³ /lbm
w	- Mass flow rate, lbm/hr
x	- Coordinate, ft
α	- Angle between V and outward drawn normal (1)
α	- Thermal diffusivity, ft ² /hr
σ	- Condensate film thickness, ft
ρ	- Density, lbm/ft ³
θ	- Time, hr
θ^*	- Residence time of condensate film, hr

SUBSCRIPTS

g - gas
 l - liquid
 p - piston
 o - outgoing stream
 i - incoming stream

I. STUDY OF RESIDUAL GAS MASS

A. EXPERIMENTAL APPARATUS

During the last period, a new tank was designed to extend the present program of residual gas mass optimization. The new tank will also enable a more accurate evaluation of the volume of vapor and the quantity of liquid nitrogen present in a closed tank undergoing bulk boiling.

The new tank has been made from a 12-in.-ID, 6061-T6 aluminum pipe, machined to a wall thickness of 1/8 in. This design is shown in Fig. 3. The stainless-steel guide rods have been moved from their previous position to the center of the tank. This offsets the effect of contraction of the styrofoam owing to changes in temperature from room temperature to -320°F . Such contraction caused binding between the piston and its guides. To represent the two-phase system accurately, careful consideration will be given to the determination of the optimum piston design. This new system has been constructed to accommodate the several piston designs contemplated. The previous system could not attain this versatility because of the permanent installation of the thermocouple and slide wires.

To study the effect of a variation in heat flux on the above programs, the tank will be wound with a nickel-copper alloy resistance heating wire called Midohm. A thermally insulating jacket will then cover the tank and heating wire to minimize the heat transfer with the ambient while operating. This wire will be wound in sections so that specified portions of the tank may be heated independently. Such a design will enable vapor-liquid holdup to be measured as a function of liquid fill.

B. ANALYSIS OF INTERFACIAL INTERACTION

1. Introduction.—Previous experimental runs have been made to determine the influence of the inlet temperature of the pressurizing gas on the quantity of that gas remaining in the gas space when tank discharge is complete. Two types of pressurize-discharge systems have been used: (1) direct contact between pressurizing gas (N_2) and liquid (N_2) and (2) a system utilizing an insulating floating piston separating the gas and liquid phase. Experimental results on both systems have been reported in Progress Report No. 9, Fig. 5, Progress Report No. 10, Fig. 3, and Progress Report No. 14, Fig. 1. In general, the results both with and without an insulating interface have been equivalent.

The interaction between the pressurizing gas and the liquid interface has been analyzed and reported in Progress Report No. 9. Experimental measurements are reported in Progress Report No. 8. These results show that, upon pressurization for the direct contact gas-liquid system, the interfacial temperature increases immediately to the saturation temperature corresponding to the new (pressurization) pressure, and that such a condition exists throughout the 2-min discharge period. This indicates that condensation of the pressurizing gas is occurring at the gas-liquid interface. The quantity of gas condensed in the 2-min interval was computed and appears to be of the order of 0.1 lbm/ft² of interfacial area. This computation was made from a theoretical analysis of the transient temperature distribution in the liquid in the region of the interface.

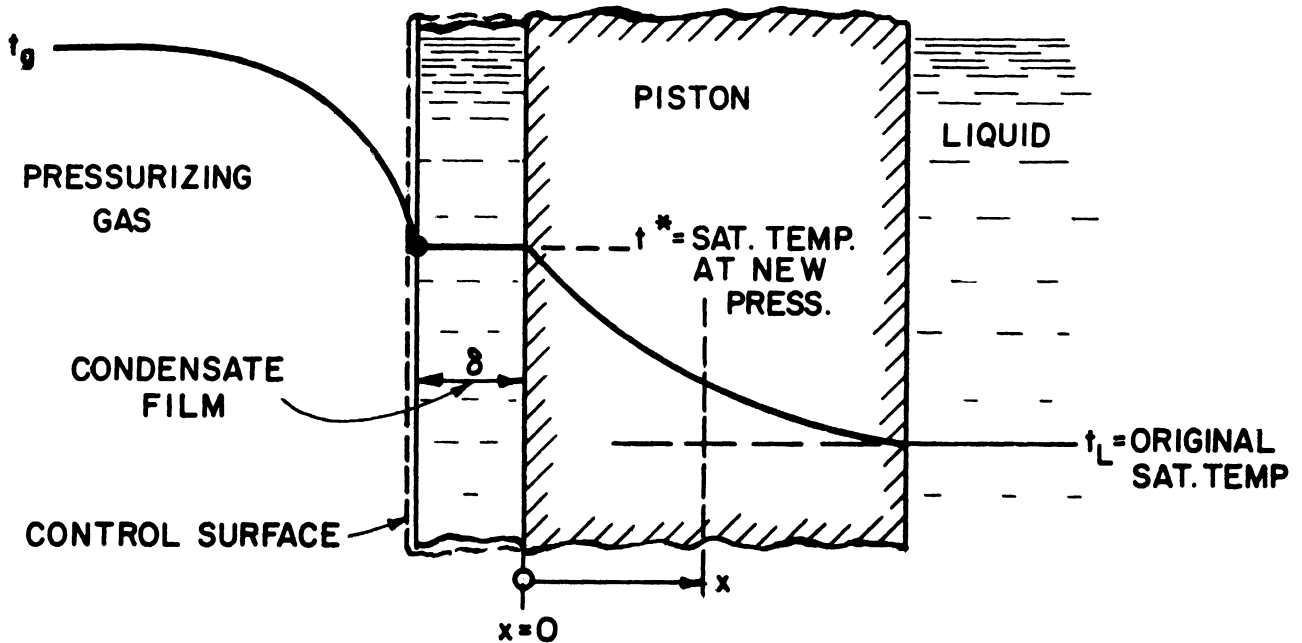
For the system in which an insulating (styrofoam) piston is placed on top of the liquid, separating it from the pressurizing gas, a somewhat different condition prevails. Owing to the different physical properties of the styrofoam interface (k , ρ and c_p), a film of condensate builds up immediately following pressurization but subsequently re-evaporates because of heat transfer with the pressurizing gas. During the time the condensate film is on the interface, the interfacial temperature is approximately the saturation temperature corresponding to the existing pressure. However, upon removal of this film by re-evaporation, the interfacial temperature undergoes a transient increase resulting from continued heat-flow interaction with the gas.

Hence, to investigate the nature of the heat-flow interaction between this type of interface and the gas so that a comparison may be made with that of the direct contact system, it is necessary first to estimate the residence time of the condensate film. Such an analysis has been completed and is outlined below. These results are, of course, general and apply to any type of an interface, including that of the direct-contact liquid-vapor system. If the residence time of the condensate film is less than the discharge time, the analysis of the total heat-flow interaction must also include an examination of the transient heat flow between the gas and the solid following the re-evaporation of the initial condensate film. In the general case this would require an analysis between a gas and a composite solid (piston and supporting liquid) starting with a nonuniform initial condition. Such an analysis is probably most conveniently done by numerical procedures for each special case. For the case of styrofoam, however, it turns out that the residence time of the condensate film is a small fraction of the discharge time, and the nonuniformity of temperature produced in the piston is not significant; therefore fairly exact calculations from continuous theory can be made of its thermal response. This has been done here both for numerical and analytical methods.

2. Analysis of Build-up and Re-evaporation of Condensate Layer During Transient Condensation.—Prior to pressurization, the interface and all matter below it is at a uniform temperature, corresponding to the saturation temperature at the system pressure just before pressurization. Upon pressurization, some of the pressurizing gas condenses at the interface. In the case of the direct gas-liquid system, it has been observed experimentally that this condensation process

causes the interfacial temperature to increase immediately to the saturation temperature corresponding to the new (pressurization) pressure. Hence, in the case of an interface consisting of another material (such as a piston), it will be assumed that the interfacial temperature will do likewise and will remain at the saturation temperature of the new pressure just as long as a condensate film remains on the interface.

The system analyzed is shown in the following sketch.



A "control surface"¹ is placed so as to completely enclose the condensate film which is experiencing a transient growth owing to heat, work, and mass transfer with the pressurizing gas at one of its interfaces and a heat transfer with the piston at its other interface.

For transient conditions, energy and mass relationships may be expressed from the First Law of Thermodynamics and the Law of Conservation of Mass as the following two rate equations:

1st Law

$$\frac{\partial}{\partial \theta} \int_V e dm + \int_A \rho e V \cos \alpha dA = q - P_x \quad (1)$$

Conservation of Mass

$$\frac{\partial}{\partial \theta} \int_V dm + \int_A \rho V \cos \alpha dA = 0 \quad (2)$$

The integrals in Eqs. (1) and (2) are over the volume and area, respectively, of the control surface. In Eq. (1), the term q is the net heat-transfer rate to and from the control surface and P_x is the net work-flow rate between the control surface and the gas. This latter term arises because vapor is flowing into or out of the control surface and also because the condensate layer is changing its volume, both of which give rise to instantaneous work effects. The various terms of Eqs. (1) and (2) will be evaluated for an area A of the interface. In Eq. (1),

$$\begin{aligned} P_x &= +p_L A \left(\frac{\partial \delta}{\partial \theta} \right) + \int_{\text{vapor}} p v \rho V \cos \alpha dA \\ &= +p_L v_L \rho_L A \frac{\partial \delta}{\partial \theta} + (p v)_g (w_o - w_i)_g \end{aligned} \quad (3)$$

and

$$q = \bar{h}_g A (t_g - t^*) + k_p A \left(\frac{\partial t}{\partial x} \right)_{x=0} \quad (4)$$

Now,

$$\begin{aligned} \int_A \rho e V \cos \alpha dA &= (w_o - w_i)_g e_g \\ &= (w_o - w_i)_g u_g \end{aligned} \quad (5)$$

and

$$\frac{\partial}{\partial \theta} \int_V e dm = \frac{\partial}{\partial \theta} (u_L \rho_L A \delta) = A u_L \rho_L \left(\frac{\partial \delta}{\partial \theta} \right) \quad (6)$$

Hence, Eq. (1) may be written

$$\begin{aligned} A u_L \rho_L \left(\frac{\partial \delta}{\partial \theta} \right) + (w_o - w_i)_g u &= \bar{h}_g A (t_g - t^*) + k_p A \left(\frac{\partial t}{\partial x} \right)_o - p_L u_L \rho_L A \left(\frac{\partial \delta}{\partial \theta} \right) \\ &\quad - (w_o - w_i)_g (p v)_g \end{aligned} \quad (7)$$

or

$$A \left(\frac{\partial \delta}{\partial \theta} \right) \rho_L h_L + (w_o - w_i)_g h_g = \bar{h}_g A (t_g - t^*) + k_p A \left(\frac{\partial t}{\partial x} \right)_o \quad (8)$$

Now, from the conservation of mass, Eq. (2),

$$w_o - w_i = -\rho_L A \left(\frac{\partial \delta}{\partial \theta} \right) \quad (9)$$

Thus, Eq. (8) is written

$$-\rho_L \left(\frac{\partial \delta}{\partial \theta} \right) (h_g - h_L) = \bar{h}_g (t_g - t^*) + k_p \left(\frac{\partial t}{\partial x} \right)_0 \quad (10)$$

or, rewriting in terms of total differentials and making the following definitions

$$h_g - h_L = h_{fg} \quad (11)$$

$$\left(\frac{\partial t}{\partial x} \right)_0 = f'_0(\theta), \quad (12)$$

Eq. (10) may be written in integral form as

$$\delta = - \frac{\bar{h}_g (t_g - t^*)}{\rho_L h_{fg}} \theta - \frac{k_p}{\rho_L h_{fg}} \int_0^\theta f'_0(\theta) d\theta \quad (13)$$

The problem which remains is to find an expression for $f'_0(\theta)$. In the case of this type of a transient, which is assumed to start with a sudden increase in surface temperature from t_L to t^* the function of $f'_0(\theta)$ may be taken as that transient temperature gradient in a homogeneous semi-infinite solid having a sudden increase in surface temperature. This is given by Carslaw and Jaeger² as follows, and will be a reasonable description of this event providing that, during the total discharge time θ , the temperature at the piston-liquid interface remains relatively unchanged. Hence

$$\left(\frac{\partial t}{\partial x} \right) = f'_0(\theta) = \frac{t_L - t^*}{\sqrt{\pi \alpha_p \theta}} e^{-\frac{x^2}{4\alpha_p \theta}} \quad (14)$$

where

$$\alpha_p = \frac{k}{\rho c_p}$$

Thus at $X = 0$

$$f'_0(\theta) = \frac{t_L - t^*}{\sqrt{\pi \alpha_p \theta}}$$

and

$$\int_0^\theta f'_0(\theta) d\theta = \frac{t_L - t^*}{\sqrt{\pi \alpha_p}} \int_0^\theta \frac{d\theta}{\theta^{1/2}} = - \frac{2(t^* - t_L)}{\sqrt{\pi \alpha_p}} \theta^{1/2} \quad (15)$$

With this result Eq. (13) becomes

$$\delta = \frac{2(t^*-t_L)}{\sqrt{\pi} \rho_L h_{fg}} \sqrt{k \rho c_p \theta} - \frac{\bar{h}_g(t_g-t^*)}{\rho_L h_{fg}} \theta \quad (16)$$

It will be observed in Eq. (16) that both $(t^*-t_L) \geq 0$ and $(t_g-t^*) \geq 0$. Thus Eq. (16) describes the build-up or transient growth of a condensate film which either grows continuously from zero to finite thickness within finite time ($t_g-t^* = 0$), or does not exist at all ($t^*-t_L = 0$ and $t_g-t^* = 0$), or which grows during some initial time to a maximum thickness and then through re-evaporation diminishes to zero again ($t_g-t^* > 0$ and $t^*-t_L > 0$). The magnitude of the maximum thickness and the time at which it occurs depends on the magnitude of t^*-t_L , t_g-t^* and the other parameters in Eq. (16).

For the purposes of this study, it is of primary interest to determine the time θ^* at which the film has completely re-evaporated. Subsequent to this time this piston and the pressurizing gas interact in a transient heat-flow process without a liquid film of condensate between them. This time θ^* may be found as the solution of Eq. (16) for $\delta = 0$.

$$\theta^* = 0$$

and

$$\theta^* = \frac{4}{\pi} \frac{(k \rho c_p)_p}{\bar{h}_g^2} \left(\frac{t^*-t_L}{t_g-t^*} \right)^2 \quad (17)$$

Equation (17) indicates the significant parameters influencing the duration of the residence of a condensate film on an interface. For the purpose of optimization of pressurizing-discharge process, it is desirable that θ^* be as small as possible. It is of interest to note that Eq. (17) discloses a new thermal property of materials which is useful in discriminating between various materials to be selected for separating the liquid from the pressurizing gas during transient processes. This property is $k \rho c_p$. It is new in that it is a different property from those frequently employed to select materials for insulating purposes. In steady-state processes the property k is used to select insulating substances; in transient distributed systems the governing thermal property is $k/\rho c_p$; in transient lumped systems it is ρc_p . For processes such as those being investigated here, a short-duration residence time to a transient condensate film, the important thermal property of the interface is $k \rho c_p$. Following the disappearance of the condensate film the significant thermal property of the interfacial material will be $k/\rho c_p$, as will be shown.

For purposes of comparison the values of $k \rho c_p$ of several substances are given below.

<u>Substance</u>	<u>(kρc_p)10³, (Btu/hr-F-ft²)²hr</u>
Aluminum (70°F)	3,000,000
Cork (100°F)	56
Fiberglas (70°F)	29
Nitrogen (liquid - 320°F)	2,310
Styrofoam (40°F)	9
Water (Liquid, 70°F)	24,920

As can be observed from this tabulation, styrofoam, the material selected for the floating piston, is far superior to the others listed insofar as minimum residence time of the condensate film is concerned. The residence time of the condensate layer on a liquid nitrogen surface is approximately 260 times longer than that on a styrofoam surface. For conditions corresponding to $t_g = -220^\circ\text{F}$, $t_L = -320^\circ\text{F}$, $t^* = -300^\circ\text{F}$, $h = 2 \text{ Btu/hr-ft}^2\text{-F}$, the residence time θ^* is found to be as follows for a styrofoam and liquid nitrogen surface:

	<u>θ^*, seconds</u>	<u>Discharge time, seconds</u>
Styrofoam	0.65	124.2
Liquid Nitrogen	169	124.2

Thus it may be estimated that the residence time θ^* is negligibly small for a styrofoam surface compared with the 125-sec discharge time.

Subsequent to θ^* the pressurizing gas and the interface continue to interact in a transient heat-transfer process, with the interfacial temperature increasing and approaching t_g . The interior of the piston also experiences transient changes in temperature.

Computations of these transient temperatures have been made and the results are shown in Figs. 1 and 2 for gas temperatures of -120 and -220°F . In view of the negligibly small time θ^* , the results have been made assuming the gas to increase suddenly in temperature from a uniform initial condition and to exchange heat with the piston interface through a coefficient of heat transfer, \bar{h}_g . Two types of analysis have been made: (1) a numerical finite difference calculation of the transient temperature distribution in the piston,³ assuming the liquid interface to be adiabatic, and (2) a mathematical calculation, assuming the piston to be a semi-infinite solid, exchanging heat with a gas through a coefficient of heat transfer, \bar{h}_g .⁴ The results from both types of calculations are essentially the same, as shown in Fig. 1. Both Figs. 1 and 2 show the influence of various values of the heat-transfer coefficient \bar{h}_g , which between the anticipated values of 2.2 and 4 Btu/hr-ft²-F is not particularly significant.

Of importance in these results is the relatively large increase in surface temperature of the piston during the 124.2-sec discharge time. This is indicative of a greatly reduced heat transfer between the gas and the piston as com-

pared with that of a direct contact gas-liquid system having a liquid nitrogen interface, as in this latter system the interface remains at -300°F during the discharge period. This reduced heat transfer then acts to preserve lower gas density in the gas space and consequently a reduced residual mass in proportion to the importance of gas-piston heat flow to the total heat flow from the gas.

C. BUBBLE DYNAMICS IN A PRESSURIZED LIQUID

Initial studies into this problem have been started. A literature search has been begun and is continuing. Included in contemplated analyses is the bubble collapse or growth times and the bubble velocity and acceleration in a pressurized, sub-cooled liquid as a function of bubble size.

D. TRANSIENT FREE CONVECTION IN A LIQUID

Work has been initiated to continue and extend the investigation of Dr. S. K. Fenster, which was reported in Technical Report No. 1. This will involve a theoretical and experimental investigation of transient free convection processes in a closed container, including any special effects of a bottom corner.

E. WORK DURING THE NEXT PERIOD

The above analysis will be extended to include a calculation of the total quantity of heat transferred to the piston face from the gas. The results will then be compared with the heat quantity transferred to the liquid nitrogen interface and their relative influence on residual gas mass obtained.

In addition, an analysis will be made of the total heat transferred between the pressurizing gas and all the inside surfaces of the container it wets, including the face of the receding piston. This analysis will include the effect of the continuous exposing of new cold surface as the piston (or interface) retreats from its initial position at the start of discharge. The consequence of this heat transfer on the final mass of residual gas will, of course, be the object of this study. Several configurations will be studied. An interesting configuration, which is suggested by this study, is one in which all exposed surfaces are made of styrofoam or similar insulating type of material. Such a system should result in a minimum heat transfer between the pressurizing gas and the container walls and should, therefore, have also the minimum residual gas mass. This final mass in such a system would include, of course, the mass of the insulation in addition to that of the residual gas. However, it seems possible that even the total final mass could be of a magnitude less than the residual gas mass in an uninsulated container. An approximate representation of this is shown by the following equation:

$$\frac{m_2}{m_1} = \frac{\rho_{g2}}{\rho_{g1}} \left(1 - \frac{4\delta}{D_0} \right) + \frac{\rho_s}{\rho_{g1}} \left(\frac{4\delta}{D_0} \right) \quad (18)$$

where

- m_2 = residual mass including insulation and gas
- m_1 = residual mass of gas in uninsulated container
- δ = thickness of insulation
- D_0 = outside diameter of container
- ρ_s = density of insulation
- ρ_{g1} = density of gas in uninsulated container
- ρ_{g2} = density of gas in insulated container

For conditions corresponding to an inlet temperature of 460°R(0°F), the residual gas mass density from experimental results on an uninsulated container (Progress Report No. 9, Fig. 5), and using styrofoam as the insulating material of thickness σ of 1 in. in a container having D_0 of 72 in., the ratio m_2/m_1 is computed to be 0.629, or a reduction in residual mass (total) of about 37%. This estimate is doubtless high, but a 15-20% reduction may be possible with this type of system.

The design and construction of the experimental apparatus for the study of liquid-vapor hold-up will continue. This equipment will consist of an electrically heated insulated tank capable of pressurization and instrumented to measure liquid, gas, and wall temperatures.

II. STUDY OF POOL BOILING IN AN ACCELERATING SYSTEM

A. CURRENT STATUS OF THE WORK

During the past reporting period, after the heater had been repaired, a total of 10 test runs were made. Of these tests, 5 were discounted because one or more of the following conditions existed:

(1) Boiling was not taking place to a saturated liquid (i.e., the liquid became subcooled). Since the degree of subcooling was not controlled, a variable is introduced which is beyond the scope of the present study.

(2) The resistivity of the water in the test vessel dropped below 900,000 Ω -cm after the test, indicating contamination.

(3) Discolorations or spots existed on the heating surface after the test, where contaminants in the water had been deposited at the active boiling sites.

(4) The value of $T_{\text{wall}} - T_{\text{sat}}$, with the system subjected to gravitational acceleration only, did not remain stable during the test period, from 10 to 12 hr.

In general, conditions (2), (3), and (4) existed simultaneously in any one test.

In addition, two tests run under presumably identical conditions were discarded because of nonreproducibility. A flow guide (MK-1) was placed over the heater with the intention of changing the configuration from a circular heated section surrounded by an unheated section to a circular heated section isolated from the unheated section. The flow guide (MK-1) consists of a stainless-steel cylindrical shell with a diameter slightly larger than the diameter of the heater, with 18-1/4-in.-diameter holes in the shell. The purpose of the holes was to equalize the water level inside and outside the flow guide. However, at this time they are not believed necessary. In the above two tests, care was not taken in placing the holes in the same position relative to the rotational direction and it is believed that the nonreproducibility resulted from different cross currents produced by the Coriolis effect on the flow pattern. A new flow guide (MK-2) was made which contained no holes.

Of the three remaining tests, two were run at the same value of heat flux, one with the flow guide and one without. The third test was run at a lower value of heat flux with the flow guide. The results are plotted in Fig. 4 as the wall temperature T_w minus the saturation temperature T_{sat} , at the heating surface versus total acceleration at the heating surface. Included also are the results for $q/A = 100,000$ Btu/hr - ft² reported in Progress Report No. 13, except that the data are plotted as $T_w - T_{\text{sat}}$ instead of the change in $T_w - T_{\text{sat}}$ between the accelerating and nonaccelerating conditions.

The trend with increasing flux at the higher accelerations should be noted. It is postulated to be due to the decreasing contribution of convection at the higher fluxes, since the nucleating sites cover a greater portion of the heating surface. This aspect will be discussed in greater detail at a future date when more data are available.

Subsequent to the above series of tests, the heater failed again, due again to a breakdown of the mica insulation at a point where the heater ribbon and mica emerge from a slot and make a 90° bend. The heater block was redesigned (designated MK-2) with the following significant changes:

(1) The slots are wider and deeper, enabling the use of thicker mica insulation.

(2) The number of slots was increased 50% and the diameter of that portion of the block containing the slots was increased, which results in a lower value of heat flux from the heater ribbon to the copper block for a given heat flux at the boiling surface. This change should also result in a higher possible heat flux at the boiling surface.

(3) The width of the shoulder on the copper block at the sides where the heater ribbons emerge from the slot was increased, eliminating the necessity of making a 90° bend in the mica insulation. This appears to have been the major weakness in the previous design.

While the redesigned heater is being fabricated, the old heater was repaired and two convection tests were made at $q/A = 5,000$ Btu/hr - ft² and $q/A = 10,000$ Btu/hr - ft² with the flow guide MK-2 installed. The data have not yet been completely reduced and will be reported during the next period.

B. WORK DURING THE NEXT PERIOD

Repeats of tests with saturated boiling will be made at heat fluxes of 10,000, 25,000, and 100,000 Btu/hr - ft² both with and without the flow guide. Also, tests will be made at $q/A = 50,000$ or 75,000 Btu/hr - ft². An attempt will then be made to reach a $q/A = 150,000$ or 200,000 Btu/hr-ft² to make this particular phase of the investigation as complete as possible.

REFERENCES

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2. Carslaw, H. S., and Jaeger, J. C., Conduction of Heat in Solids (Oxford University Press, Fair Lawn, New Jersey, 1950), p. 41, Eq. (7).
3. McAdams, W. H., Heat Transmission, 3rd ed. (McGraw-Hill Book Co., New York, 1954), Chap. 3.
4. Carslaw and Jaeger, op. cit., p. 53, Eq. (1).

FIG. 1

PISTON RESPONSE TO A STEP CHANGE IN TEMPERATURE
(ANALYSIS OF RESIDUAL GAS MASS)

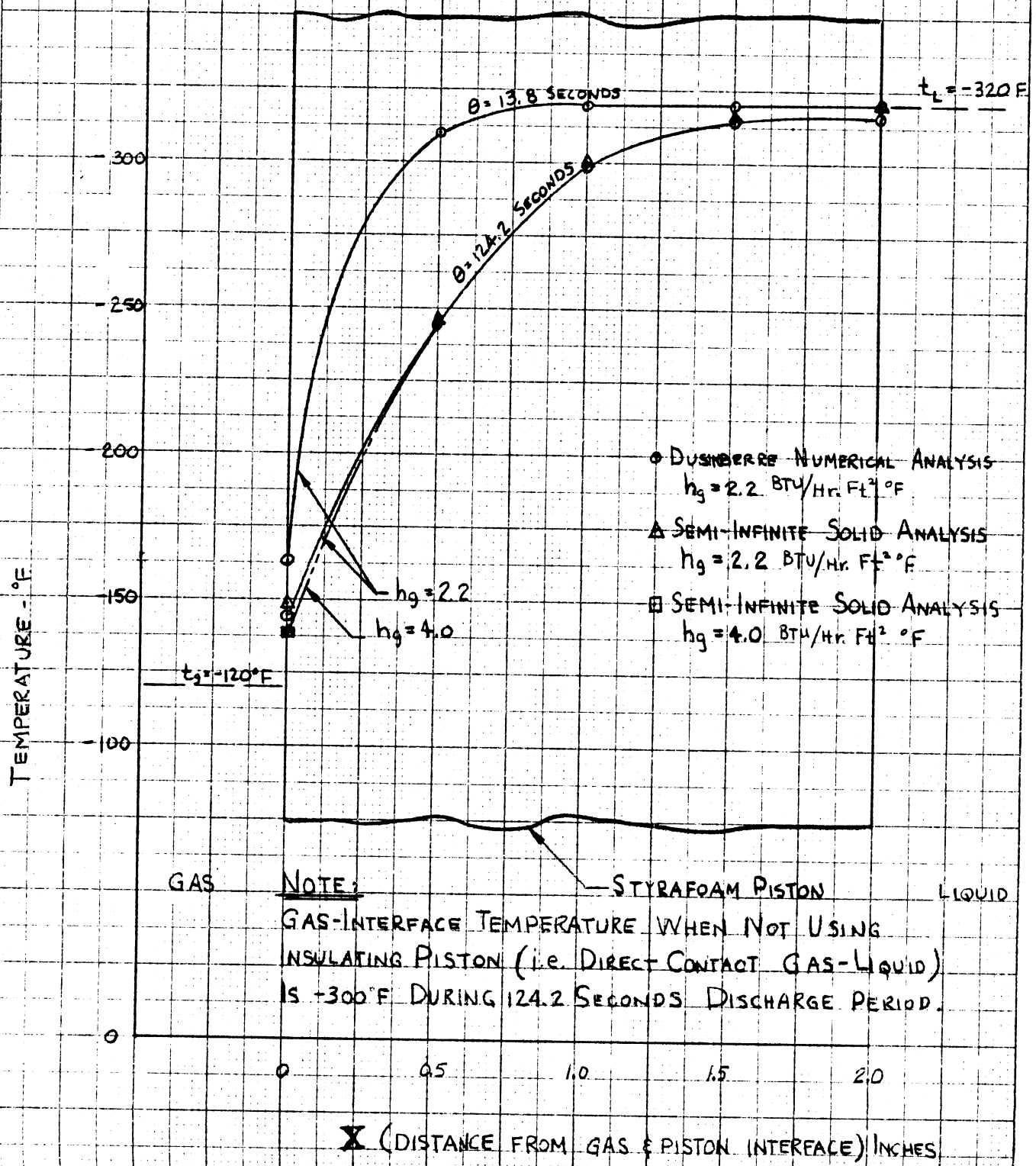


Fig. 2

PISTON RESPONSE TO A STEP CHANGE IN TEMPERATURE
(ANALYSIS OF RESIDUAL GAS MASS)

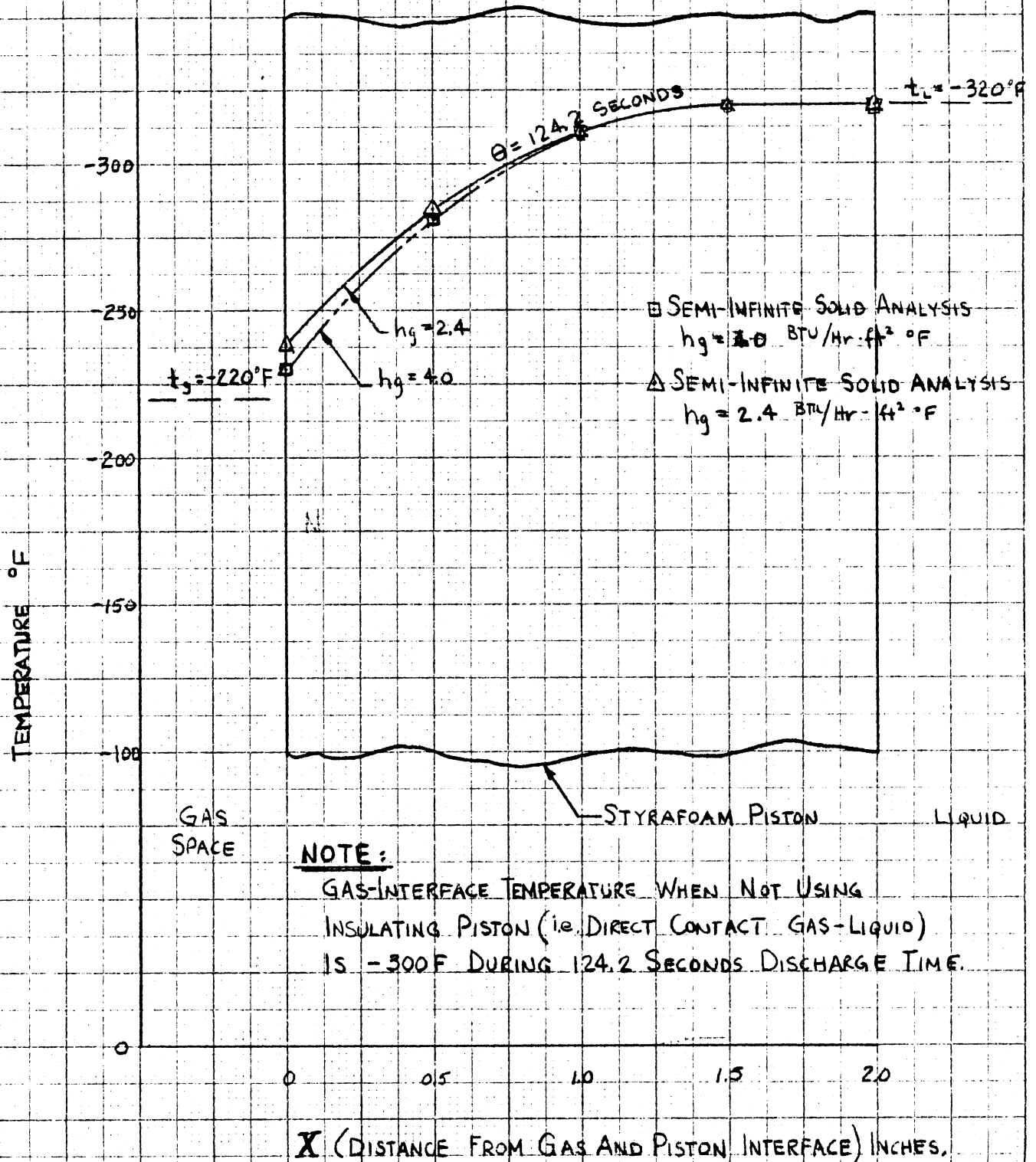


Fig. 3. New Design of Apparatus to Measure Mass of Residual Gas and Liquid-Vapor Fractions

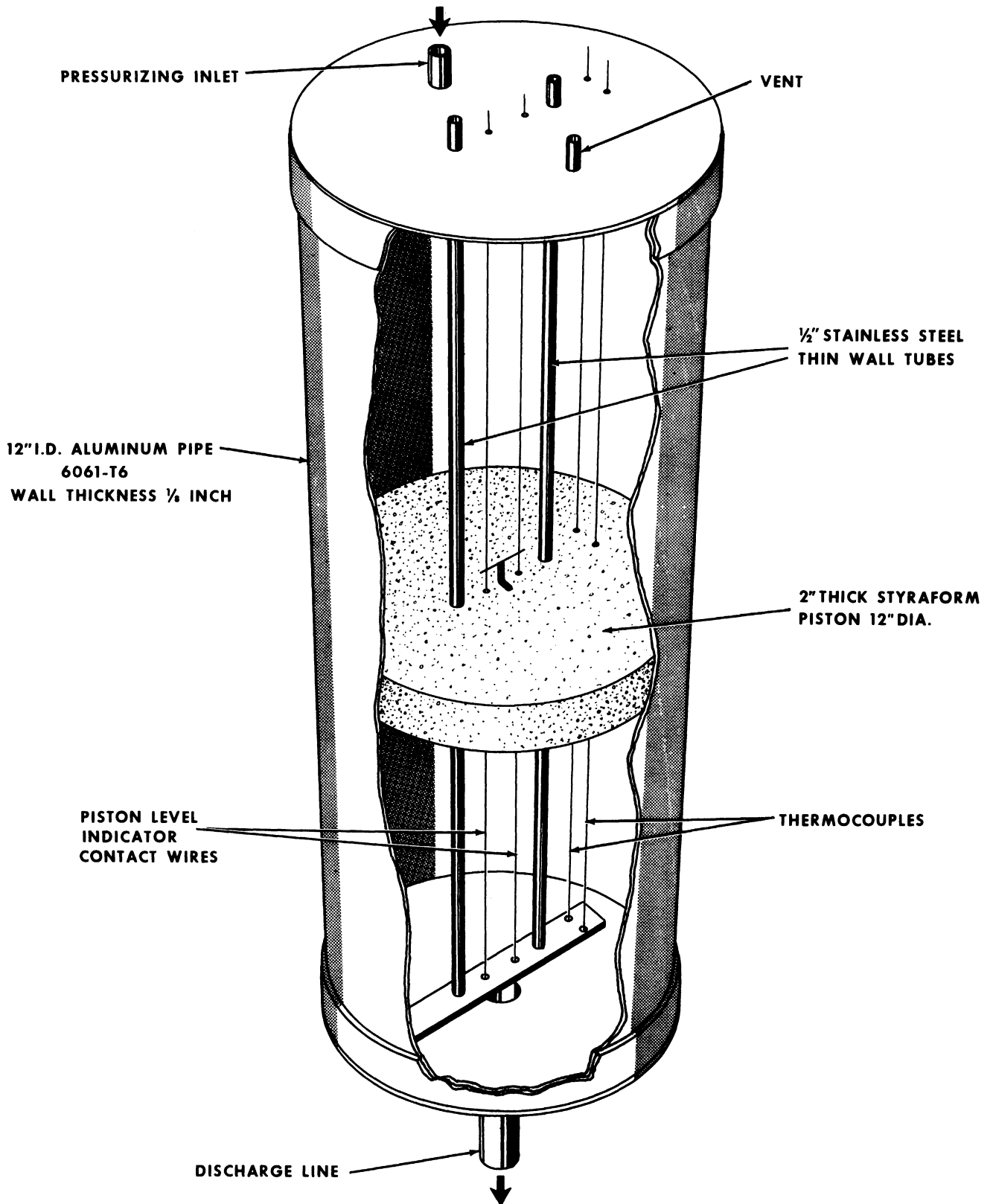
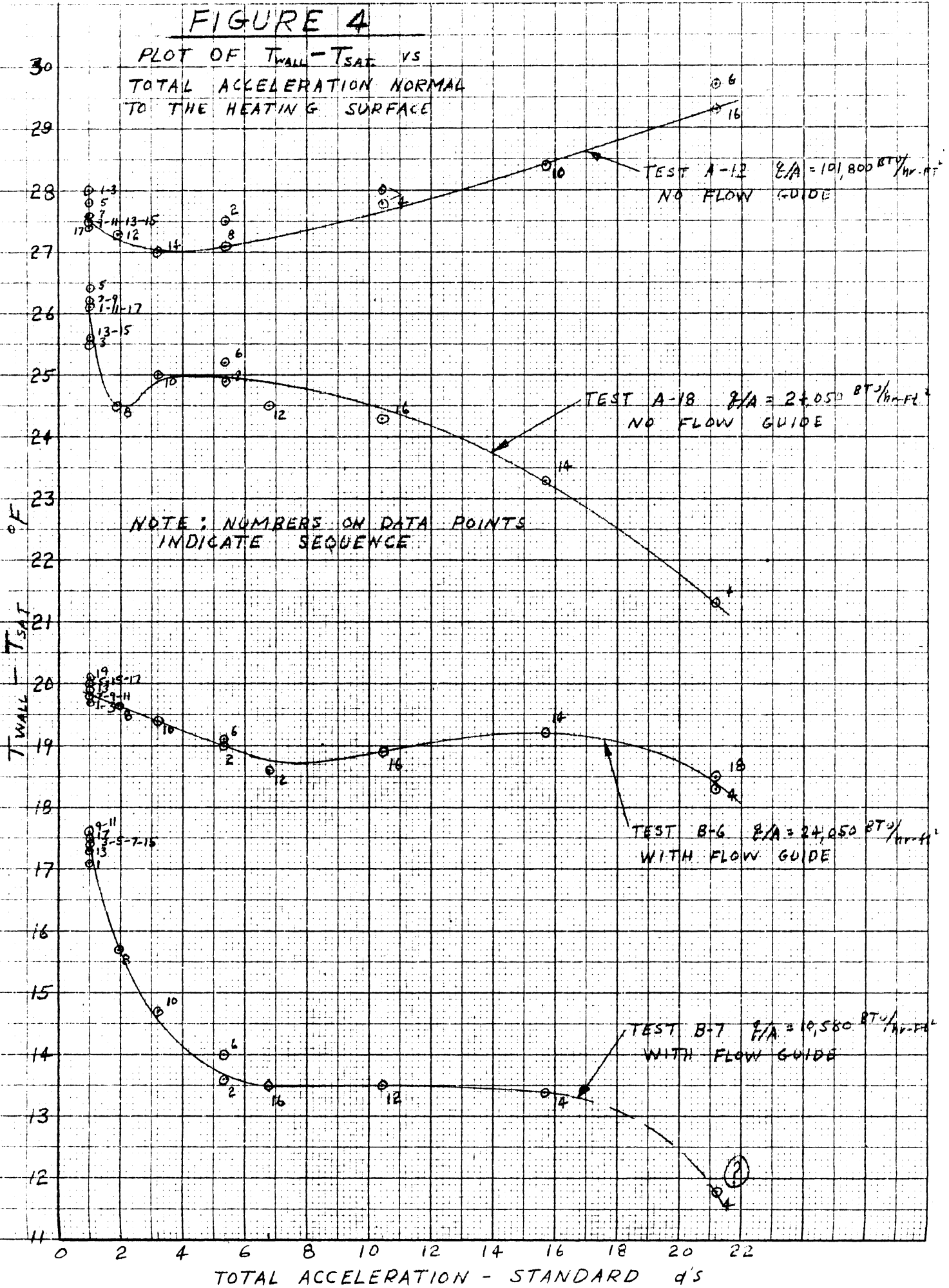


FIGURE 4

PLOT OF $T_{WALL} - T_{SAT}$ VS
TOTAL ACCELERATION NORMAL
TO THE HEATING SURFACE



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