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Progress Report No. 9

PRESSURIZATION OF LIQUID OXYGEN CONTAINERS

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ABSTRACT

During the period April 1, 1958, to May 15, 1958, approximately 300 payroll man-hours and 550 nonpayroll man-hours were spent on the project. During this period data were collected for the programmed heat-flux runs, all experimental data were reduced, and several theoretical analyses were conducted on the interaction of the pressurizing gas with the container walls, residual gas, and liquid interface.

The mass of the residual gas was computed for nitrogen pressurization and found to be a function of inlet gas temperature, decreasing approximately 20% for an increase in inlet gas temperature from -200°F to +110°F.

In addition, Ph.D. thesis work of Mr. Herman Merte, Jr., and Mr. Saul K. Fenster continued on a full-time basis.

NOMENCLATURE

A		annular area of container wall
a_1	=	$1/2\ (m + \sqrt{m^2 + 4\ p_o})$
a_2	=	$1/2\ (\sqrt{m^2 + 4\ p_o} - m)$
C		mean circumference of container
c_p		specific heat of container material
h_a, h_g, h_L		coefficients of heat transfer for air, gas, and liquid, respectively
k		thermal conductivity of container material
$\mathcal{L}$	=	$\dfrac{h_g\ C\ \Theta_g^*}{kA}$
m	=	$\dfrac{\rho\ VA\ c_p}{kA} = \dfrac{\rho\ V\ c_p}{k}$
p_o	=	$\dfrac{h_a C + h_g C}{kA}$
p_1	=	$\dfrac{h_a C + h_L C}{kA}$
q_o	=	$\dfrac{h_a C t_a + h_g C t_{g_\infty}}{kA}$
q_1	=	$\dfrac{h_a C t_a + h_L C t_L}{kA}$
t		temperature of container wall (fluid)
t^*		temperature of container wall at $x = 0$
t_∞		temperature of container wall at $x = \infty$:

$$t_\infty = \frac{h_a t_a + h_g t_{g_\infty}}{h_a + h_g}$$

NOMENCLATURE (Concluded)

$t_{-\infty}$ temperature of container wall at $x = -\infty$:

$$t_{-\infty} = \frac{h_a t_a + h_L t_L}{h_a + h_L}$$

t_g temperature of vapor

t_{go} temperature of vapor at $x = 0$ (equal to saturation temperature at 50 psia)

$t_{g\infty}$ temperature of vapor at $x = \infty$

t_L temperature of liquid

V velocity of container wall (equal to liquid-vapor interfacial velocity)

ρ density of container wall material

$\theta_g = t_{g\infty} - t_g$

$\theta_g{}^* = t_{g\infty} - t_{go}$

$\theta_1 = t - t_{-\infty}$

$\theta_1{}^* = t^* - t_{-\infty}$

$\theta_2 = t - t_\infty - \dfrac{\mathcal{L} e^{-\gamma x}}{(\gamma^2+m\gamma-p_o)}$

$\theta_2{}^* = t^* - t_\infty - \dfrac{\mathcal{L}}{(\gamma^2+m\gamma-p_o)}$

γ exponent describing vapor temperature:

$$\frac{\theta_g}{\theta_g{}^*} = e^{-\gamma x}$$

EXPERIMENTAL WORK

PROGRAMMED HEAT FLUX

To carry out runs with a programmed heat flux, electrical resistance wire has been wound around the tank. The assembly is made in the following way.

Mylar tape 0.001 in. thick was wrapped around the tank. This tape insulates only negligibly thermally but is an electrical insulator. Helical coils with a pitch of 3.5/in. of chromel A ribbon (1/8 in. by 0.0159 in.) were then placed over the mylar tape. A jacket of styrofoam 3 in. thick was placed over the coils.

For the programmed heat-flux data, two types of runs were employed.

A. The heat flux and boil-off rate were established prior to pressurization.
B. The heat flux was established subsequent to pressurization, simultaneously with beginning of discharge.

One run of type A and one run of type B, with an inlet gas temperature of 110°F, were performed at a power level of 4000 watts (heat flux = 1200 Btu/hr-ft^2) and are now in the process of reduction.

One run of type A, with an inlet gas temperature of 63°F at 1000 watts (heat flux = 300 Btu/hr-ft^2) has also been taken. Reduction of this run has been delayed owing to the low heat-flux level. The programmed heat-flux operation was discontinued when a short circuit caused a fire in the styrofoam insulation.

DATA REDUCTION

FLUID AND WALL TEMPERATURES

The temperatures of the wall, the liquid, and gas space have all been reduced as °F vs. time in seconds.

The wall-temperature distribution about the liquid-gas interface during discharge has been plotted for all runs. Representative data are shown in Figs. 1, 2, 3, and 4. These data suggested a theoretical model for the wall-temperature transient analysis which is included in this report.

Figures 1 to 4 show data for the wall temperatures about the liquid-vapor interface for both nitrogen and helium pressurization runs. These data are shown from the standpoint of an observer located at the interface. Two principal observations are to be noted: First, there is a significant difference in wall-temperature distribution about the interface when a condensible gas (nitrogen) rather than a noncondensible (helium) is used. The general flattening of the wall temperature in the regions just above the interface for the nitrogen runs appears to be a result of condensation of that gas on the cold wall. This effect is not found in the helium runs. The second observation is the relative constancy of the wall temperature during the time the wall is wetted with liquid followed by a rapid increase in temperature during or subsequent to the passage of the liquid-vapor interface. This behavior suggests the existence of a very steep temperature gradient in the liquid near the interface, something which is confirmed by the liquid temperature measurements in this region and which is discussed later in this report. It appears, furthermore, that axial heat-conduction effects in the wall are of the greatest importance only during the passage of the liquid-vapor interface and probably are of secondary importance before and after this time.

RESIDUAL GAS MASS

The mass of residual gas (Fig. 5) has been determined at the instant the tank is empty for all the nitrogen discharge runs, exclusive of the high heat-flux runs, as a function of the inlet gas gemperature.

Inlet Gas Temperature, °F	Residual Mass of Gas Nitrogen Discharge, lbm
111	1.077
63	1.126
-30	1.171
-299	1.352

These results are slightly changed from those previously reported owing to a misalignment discovered in the location of the fluid thermocouples.

ANALYTICAL WORK

The analytical prediction of the behavior of pressurizing-discharge systems requires technical data and procedures for calculating the interaction of the inlet (pressurizing) gas with the container walls, residual gas, and liquid-gas interface, and the conduction of heat along the container walls. With a view towards establishing such generalized calculation procedures, certain analyses have been made during this period in which the observed physical behavior of the experimental system has been used to indicate the nature of a theoretical model. Accordingly, the following interactions have been studied analytically and are described in the sections which follow:

(A) axial wall-temperature transient analysis;
(B) gas space and ambient heat-transfer coefficients; and
(C) liquid-gas interfacial condensation.

Because the original intent of the design of the experimental apparatus was directed principally towards the measurement of the mass of residual gas, the usefulness of these analyses for data presently collected is limited. In certain cases considerably more detailed information is required than has been measured by the present apparatus. This is especially true of analysis (B) where the measurement of more wall and fluid temperatures are necessary for a more satisfactory estimate of the local, transient coefficients of heat transfer on the gas side.

(A) WALL-TEMPERATURE TRANSIENT ANALYSIS

In the work described here, analytical descriptions and solutions to the problem of wall-temperature transients during liquid emptying have been developed.

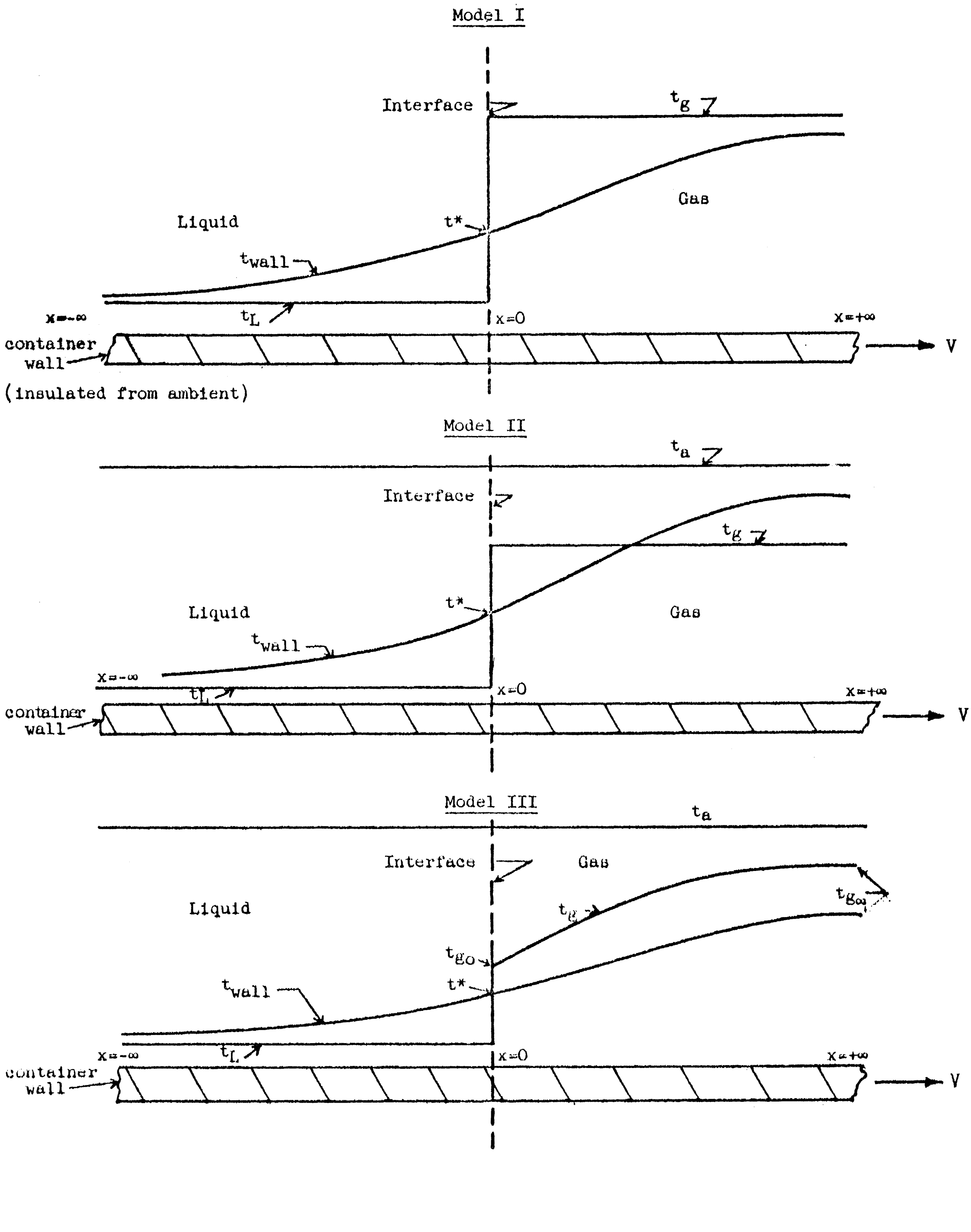
Three idealized models of the physical system have been used and are shown in the following sketches. In the resulting analyses, the system is viewed from the vantage point of an observer located at the liquid-vapor interface. From this point of view, it is the container wall which moves, and with a velocity equal in magnitude but opposite in direction to that of the liquid-vapor interface, when the liquid is being discharged from the container. In this way, the metal wall of the container may be likened to a fluid flowing steadily through a heat exchanger, and the equations for temperature developed represent the temperature history of the fluid (container wall) from the time it enters the system to the time it leaves. In this way the transient conditions in the container wall have been reduced to those corresponding to a steady-state process.

Analytical solutions for the first two models have been completely developed and solved and are available. In the interest of brevity, only the solutions to the third have been included in this report.

Model III.—The model assumed for this analysis is shown on the sketches as Model III. The coefficients of heat transfer are assumed to remain constant. The temperature of the vapor is assumed to be described by an "e-power" equation as follows:*

*All symbols are defined in the Nomenclature.

IDEALIZED MODELS I, II, and III



$$\frac{\theta_g}{\theta_g^*} = \frac{t_{g\infty} - t_g}{t_{g\infty} - t_{go}} = e^{-\gamma x} \qquad (0 \leqq x \leqq \infty) \quad . \tag{1}$$

Applying the First Law of Thermodynamics to the region $(-\infty \leqq x \leqq 0)$, the following differential equation is obtained:

$$\rho\, VA\, c_p \frac{\partial t}{\partial x} = kA \frac{\partial^2 t}{\partial x^2} + h_a C(t_a - t) - h_L C(t - t_L) \quad . \tag{2}$$

This reduces to:

$$\frac{\partial^2 t}{\partial x^2} - m \frac{\partial t}{\partial x} - p_1 t = -q_1 \quad . \tag{3}$$

In deriving this equation, an energy balance is made for an element of length dx in space, whose mean circumference and annular area are C and A, respectively. The container wall flows through this element. Heat exchange occurs between the wall and ambient, and between the wall and the liquid. In addition, axial heat conduction takes place along the wall.

Applying the boundary conditions,

(a) At $x = 0$, $t = t^*$

(b) At $x = -\infty$, t is finite, (4)

the following solution for t is obtained (for $-\infty \leqq x \leqq 0$):

$$t = \left\{ t^* - \frac{h_a t_a + h_L t_L}{h_a + h_L} \right\} e^{1/2(m+\sqrt{m^2+4p_1})x} + \frac{h_a t_a + h_L t_L}{h_a + h_L} \quad . \tag{5}$$

For the region $(0 \leqq x \leqq \infty)$ the following differential equation is written:

$$\rho\, VA\, c_p \frac{\partial t}{\partial x} = kA \frac{\partial^2 t}{\partial x^2} + h_a C(t_a - t) + h_g C(t_g - t) \tag{6}$$

$$= kA \frac{\partial^2 t}{\partial x^2} + h_a C(t_a - t) + h_g C(t_{g\infty} - \theta_g^* e^{-\gamma x} - t) \quad . \tag{7}$$

This reduces to:

$$\frac{\partial^2 t}{\partial x^2} - m \frac{\partial t}{\partial x} - p_o t = \mathscr{L} e^{-\gamma x} - q_o \quad . \tag{8}$$

Applying the boundary conditions,

(a) At $x = 0$, $t = t^*$

(b) At $x = \infty$, t is finite, (9)

the following solution is obtained:

$$t = \left\{ t^* - \frac{h_a t_a + h_g t_{g_\infty}}{h_a + h_g} \right\} e^{-1/2(\sqrt{m^2+4p_o}-m)x} + \frac{h_a t_a + h_g t_{g_\infty}}{h_a + h_g} + \frac{\mathcal{L}[e^{-\gamma x} - e^{-1/2(\sqrt{m^2+4p_o}-m)x}]}{(\gamma^2 + m\gamma - p_o)} \quad . \tag{10}$$

To evaluate t^* (the wall temperature at the liquid-vapor interface), the gradients of the wall temperature in the regions $(0 \leqq x \leqq \infty)$ and $(-\infty \leqq x \leqq 0)$ are equated.

For the region $(-\infty \leqq x \leqq 0)$ we have:

$$t = (t^* - t_{-\infty})e^{a_1 x} + t_{-\infty} \quad . \tag{5a}$$

Differentiating with respect to x yields:

$$\frac{\partial t}{\partial x} = a_1(t^* - t_{-\infty})e^{a_1 x} \quad . \tag{11}$$

Evalusting the gradient at the interface (x=0) gives:

$$\left(\frac{\partial t}{\partial x}\right)_o = a_1(t^* - t_{-\infty}) \quad . \tag{12}$$

For the region $(0 \leqq x \leqq \infty)$:

$$t = \left\{ t^* - t_\infty - \frac{\mathcal{L}}{(\gamma^2 + m\gamma - p_o)} \right\} e^{-a_2 x} + t_\infty + \frac{\mathcal{L} e^{-\gamma x}}{(\gamma^2 + m\gamma - p_o)} \quad . \tag{10a}$$

The gradient is therefore:

$$\frac{\partial t}{\partial x} = -a_2 \left\{ t^* - t_\infty - \frac{\mathcal{L}}{(\gamma^2 + m\gamma - p_o)} \right\} e^{-a_2 x} - \frac{\gamma \mathcal{L} e^{-\gamma x}}{(\gamma^2 + m\gamma - p_o)} \quad . \tag{13}$$

At the interface, the gradient becomes

$$\left(\frac{\partial t}{\partial x}\right)_o = - a_2 \left\{ t^* - t_\infty - \frac{\mathcal{L}}{(\gamma^2+m\gamma-p_o)} \right\} - \frac{\gamma\mathcal{L}}{(\gamma^2+m\gamma-p_o)} \quad . \tag{14}$$

Equating the gradients at the interface, Eqs. (12) and (14), and solving for t^*, the following result is obtained:

$$t^* = \frac{1}{a_1+a_2} \left\{ \frac{\mathcal{L}(a_2-\gamma)}{(\gamma^2+m\gamma-p_o)} + a_1 t_{-\infty} + a_2 t_\infty \right\} . \tag{15}$$

A substitution of the system constants into this expression for t^* reveals the most influential term to be:

$$\frac{a_1 t_{-\infty}}{a_1+a_2} \quad .$$

Note further that $a_1 \gg a_2 (a_1+a_2 \approx a_1)$ gives the result:

$$t^* \approx t_{-\infty} \quad .$$

This is well supported by experimental data as is anticipated on physical grounds.

Substituting the value of t^* as obtained in Eq. (15) into Eq. (12) gives:

$$\left(\frac{\partial t}{\partial x}\right)_o = \frac{a_1}{a_1+a_2} \left\{ \frac{\mathcal{L}(a_2-\gamma)}{(\gamma^2+m\gamma-p_o)} + a_2(t_\infty - t_{-\infty}) \right\} \quad . \tag{16}$$

Again making the approximation that $a_1/a_1+a_2 \approx 1$, the expression for the wall-temperature gradient at the interface becomes:

$$\left(\frac{\partial t}{\partial x}\right)_o \cong \left\{ \frac{\mathcal{L}(a_2-\gamma)}{(\gamma^2+m\gamma-p_o)} + a_2(t_\infty - t_{-\infty}) \right\} . \tag{17}$$

As shown by Eqs. (11) and (13), the temperature gradient is greatest at the interface. This is to be expected since the most severe liquid-gas temperature changes are experienced in the region of the interface.

For infinite interfacial velocity, the gradient becomes zero for all values of x, indicating that no heat transfer is taking place.

The table below shows a comparison of t* computed from Eq. (15) for two cases with that obtained experimentally for the 111°F inlet temperature runs.

	Case I (a)	Case II (b)
t*, Eq. (15)	-305.27°F	-306.62°F
t*, Experimental (mean of 3 wall thermocouples)	-306.67°F	-306.67°F

Case I (a)	Case II (b)
h_a = 2 Btu/hr-ft²-°F	h_a = 2 Btu/hr-ft²-°F
h_L = 60 Btu/hr-ft²-°F	h_L = 60 Btu/hr-ft²-°F
h_g = 24 Btu/hr-ft²-°F	h_g = 12 Btu/hr-ft²-°F
t_L = -320°F	t_L = -320°F

These results indicate the insensitivity of t* to h_g, the gas-space heat-transfer coefficient. The temperature gradient and the wall temperature above the interface may be expected to be more sensitive to this quantity, however.

It is of interest to examine the behavior of the equations derived for wall temperature when the extreme conditions of infinite and zero container wall velocities are imposed.

For the region $(-\infty \leqq x \leqq 0)$, an infinite container-wall velocity, when substituted into Eq. (5) for the wall temperature, yields:

$$t = t_{-\infty} = t^* \quad . \tag{17a}$$

Since the container-wall velocity is infinite, there is no time during which heat transfer may occur. Hence the wall temperature remains at the same level as at $(-\infty)$ throughout the entire region.

For the region $(0 \leqq x \leqq \infty)$, an infinite velocity yields from Eq. (10):

$$t = t^* = t_{\infty} \quad . \tag{17b}$$

Once again, the constancy of the wall temperature is observed. Further, it is seen that:

$$t_{-\infty} = t^* = t_{\infty} \quad . \tag{17c}$$

When the velocity of the container wall is zero, the wall-temperature distribution in the region $(-\infty \leqq x \leqq 0)$ becomes:

$$t = \{t^* - t_{-\infty}\} e^{\sqrt{p_1}\, x} + t_{-\infty} \quad , \tag{18}$$

and for the region ($0 \leq x \leq \infty$) for the same condition, the wall temperature becomes:

$$t = \left\{ t^* - t_{\infty} - \frac{\mathcal{L}}{(\gamma^2 - p_o)} \right\} e^{-\sqrt{p_o}\, x} + t_{\infty} + \frac{\mathcal{L} e^{-\gamma x}}{(\gamma^2 - p_o)} \tag{19}$$

These results are those one would obtain for a "static" system, where there is no motion, in which the wall itself is essentially a "fin" connecting two semi-infinite regions, one having liquid-ambient heat interaction, the other having gas-ambient heat interaction.

(B) GAS SPACE AND AMBIENT HEAT-TRANSFER COEFFICIENTS

During the experimental operation certain wall and fluid temperatures were measured as functions of time. The transient in the wall temperature in regions well removed from and above the liquid interface resulted from convective heat-transfer interaction with the pressurizing gas and the ambient and heat conduction along the wall. This can be described analytically at any point in the wall by the First Law of Thermodynamics as:

$$\frac{\partial t}{\partial \tau} = \frac{1}{\rho\ c_p\ \Delta x} [h_g(t_g - t) + h_a(t_a - t)] + \frac{k}{\rho\ c_p} \left(\frac{\partial^2 t}{\partial x^2} \right) . \tag{20}$$

Except in regions near the liquid-vapor interface, the axial heat-conduction term $(k/\rho c_p)(\partial^2 t/\partial x^2)$ was found to be negligible. Hence, for locations removed from this interface:

$$\frac{\partial t}{\partial \tau} = \frac{1}{\rho\ c_p\ \Delta x} [h_g(t_g - t) + h_a(t_a - t)] \quad , \tag{21}$$

where:

- ρ = density of wall material
- c_p = specific heat of wall material
- Δx = thickness of wall
- h_g = heat-transfer coefficient on pressurizing gas side
- h_a = heat-transfer coefficient on ambient side
- t_g = mean temperature of pressurizing gas in same plane as t
- t = wall temperature
- t_a = ambient temperature
- τ = time

Equation (21) suggests the possible use of the measured transients in wall and fluid temperatures in the determination of either h_g or h_a or both. This technique is essentially based on recognizing the wall as a calorimeter. Experimentally, both t and t_g are measured as functions of time. This means that at an instant of time $\partial t/\partial \tau$, t, t_a, and t_g may, in principle, be known if a sufficient number of thermocouples have been put into the system. Then if an estimate can be made of, say, h_a, the only unknown is h_g which may be obtained from Eq. (21). On the other hand, if h_a cannot be estimated but if it may be assumed that both h_g and h_a remain constant over an interval of time, then Eq. (21) may be used at both ends of this interval to give two equations containing the two unknowns h_g and h_a.

This technique has been used on some of the data but as yet without a great deal of success. The basic problem is the difficulty of obtaining a reliable measurement of t_g. The apparatus originally was designed with other purposes in mind, and hence only a limited number of gas temperature measurements are made. This is further complicated by the slow thermal response of the thermocouples as they emerge from the liquid and penetrate the gas space. For a considerable length of time following their emergence from the liquid, the thermocouples are apparently surrounded with a slowly evaporating film of liquid which causes them to register the saturation temperature rather than gas temperature. This phenomenon was mentioned in the last progress report (page 2). However, further attempts will be made with this technique to evaluate its use with the present data.

For the programmed heat-flux runs, the influence of the ambient is replaced with a measured heat flux, $(q/A)_a$. This provides another possibility for obtaining h_g. This case may be described by the following equation for negligible axial heat conduction:

$$\frac{\partial t}{\partial \tau} = \frac{1}{\rho \; c_p \; \Delta x} \; [h_g(t_g - t) + (q/A)_a] \quad . \tag{22}$$

The problem of t_g remains in the use of this result but some advantage is obtained from eliminating h_a as a variable. During the next period an attempt will be made to use this equation on the programmed heat-flux data.

(C) LIQUID-GAS INTERFACIAL CONDENSATION

A problem associated with the discharge process is the heat-mass interaction of the pressurizing gas with the liquid interface. Pressurization consists of the sudden exposure of the subcooled liquid to a condensible gas at much higher temperature. This results in an immediate condensation of pressurizing gas at the interface, causing the interfacial temperature to increase suddenly to the saturation temperature corresponding to the pressurizing pressure. Two effects result: (1) a transient conduction of heat from the hotter interface to the cooler liquid, and (2) condensation of pressurizing gas on the interface.

Assuming this interaction to be similar to that of transient heat conduction into a semi-infinite solid, estimates have been made of the temperature distributions in the liquid and the amount of pressurizing gas condensed.

Figure 6 shows the temperature of the liquid in regions below the interface at times of 1/2, 1, and 2 minutes following pressurization. The significant result of this is the predicted steep temperature gradients which exist in the liquid below the interface. That is, at the end of discharge (θ = 2 minutes), only that region of the liquid less than 1 in. below the interface has increased in temperature by more than about 4°F. At θ = 1/2 minute, only that region within 1/2 in. of the interface has increased in temperature by more than 4°F. This suggests that the bulk of the liquid is essentially uninfluenced by heat transfer with the pressurizing gas during discharge. These steep temperature gradients in the liquid have been observed (see Figs. 3 and 4, Progress Report No. 8), and have the character described in Fig. 6. Furthermore, it might be expected that inlet gas temperature would have no significant effect on this progress, which appears to be the case from the measurements reported in Progress Report No. 8. The presence of these steep gradients in temperature in the liquid assist in the indication of the passage of the liquid-vapor interface from measurements of the response of the liquid thermcouples.

While the liquid is essentially uninfluenced by the gas, the liquid affects the gas by providing a cold surface on which the gas may condense. From the above results it is possible to estimate the amount of gas which condenses on the interface during discharge. This estimation assumes that the effect of the condensation does not alter the movement of the interface owing to a build-up of condensate. This will be reasonable if the amount of condensed gas is small, which it appears to be.

From the First Law of Thermodynamics it is seen that the heat effect of condensation, Q, at any time θ, results in an increase of enthalpy H of the liquid. Hence,

$$Q = mh_{fg} = \int_0^\infty (h-h_o)\, dm$$

$$= \int_0^\infty c_p\rho(t-t_o)\, Adx$$

$$= \rho\, c_p\, A(t_s-t_o) \int_0^\infty \left(\frac{t-t_o}{t_s-t_o}\right) dx \quad . \tag{23}$$

Now, by heat-conduction theory:

$$\frac{t - t_o}{t_s - t_o} = 1 - \text{erf} \frac{x}{2\sqrt{\alpha\theta}} \quad . \tag{24}$$

Thus,

$$m h_{fg} = \rho \, c_p \, A \, 2\sqrt{\alpha\theta} \, (t_s - t_o) \int_0^\infty \left(1 - \text{erf} \frac{x}{2\sqrt{\alpha\theta}}\right) d\left(\frac{x}{2\sqrt{\alpha\theta}}\right) , \tag{25}$$

or,

$$m = \frac{2 \rho \, c_p \, A}{h_{fg}} \sqrt{\alpha\theta} \, (t_s - t_o) \int_0^\infty \text{erf } c\phi \, d\phi \quad . \tag{26}$$

Now, Carslaw and Jaeger, in Conduction of Heat in Solids,* give

$$\int_0^\infty \text{erf } c\phi \, d\phi = 0.564 \quad . \tag{27}$$

Hence,

$$m = 1.128 \frac{\rho \, c_p \, A}{h_{fg}} (t_s - t_o) \sqrt{\alpha\theta} \tag{28}$$

For liquid nitrogen with the following properties:

ρ = 47 lbm/ft^3
c_p = 0.49 Btu/lbm-°F
k = 0.123 Btu/hr-ft-°F
α = 0.0535 ft^2/hr

$t_s - t_o$ = 21.6°F
θ = 2 minutes
h_{fg} = 77 Btu/lbm at 50 psia
subscript o = initial condition
subscript s = saturation condition

the amount of condensed gas is 0.0764 lbm. This is negligibly small.

*Clarendon Press, Oxford, 1950, pp. 371, 373.

MERTE'S THESIS:
A STUDY OF POOL BOILING IN AN ACCELERATING SYSTEM

A. CURRENT STATUS OF WORK

1. Experimental Apparatus

The construction of the basic experimental apparatus by the sub-contractor is approximately 75% complete. Difficulty has been encountered in milling the 32 slots (0.008 in. wide by 5/16 in. deep) in the copper surface for the heater ribbons. About 75% of the associated instrumentation furnished by the University has been received.

2. Theoretical Considerations

The theoretical aspects of the effect of acceleration on boiling heat transfer is proceeding along two directions:

(a) Existing correlations have been modified to take into account the effect of an acceleration. The correlation due to Rohsenow[1] for nucleate boiling is as follows (modified):

$$\frac{c_L T_x}{h_{fg}} = c_{sf}\left[\frac{q/A}{\mu_L\, h_{fg}}\sqrt{\frac{g_o \sigma}{(\rho_L-\rho_V)\; g(1+a/g)}}\right]^{0.33}\left(\frac{c_L \mu_L}{k_L}\right)^{1.7} , \qquad (29)$$

where a is the difference between the absolute acceleration on the system and that due to gravity. The insertion of the acceleration term $(1+a/g)$ is justifiable since the expression in which it appears has been shown by Fritz[2] to be related to the diameter of a bubble at departure from a heating surface. Fritz' derivation is based on the solution of Bashforth and Adams[3] of the equilibrium of bubbles in a force field. The magnitude of the force enters the solution in combination with other properties as a parameter. However, in deriving the above expression, the assumption is made that the product $f \times D_b$ = constant (where f = frequency of bubble formation, D_b = diameter of bubble at moment of departure from the surface). This has been shown to be approximately true in a gravitational field by Jacob,[4] but may no longer be valid with an accelerating system. The experimental data should clarify this. A correlation for peak heat fluxes has been developed by Zuber[5] and with the modification is:

$$\frac{(q/A)_{max}}{h_{fg}\,\rho_V} = \frac{\pi}{24}\left[\frac{\sigma(\rho_L-\rho_V)g}{\rho_V^2\, g_o}\;(1+a/g)\right]^{1/4}\left(\frac{\rho_L+\rho_V}{\rho_L}\right)^{-1/2} . \qquad (30)$$

This expression indicates that, with an acceleration of 10 g, for example, the peak heat flux would increase by a factor of about 1.8.

(b) Attempts are being made to establish a new relationship which, in conjunction with the experimental data, may reveal more information on the phenomenon of boiling, particularly as regards the departure of the bubble from the heating surface.

B. EXPECTED STATUS AS OF JUNE 30, 1958

(a) Thermocouple assemblies to be constructed and calibrated.

(b) The manufacture of the flat plate heater to be completed and assembled. The assembly will involve cementing the heater ribbon-mica laminations into the 0.008-in. slots.

(c) A small stainless-steel drum has been procured and stainless-steel immersion heaters will be obtained to make a de-aerating unit for distilled water.

(d) Partial assembly of the basic experimental apparatus and associated instrumentation.

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3. Bashforth, F., and Adams, J., Capillary Action, Cambridge, 1883.

4. Jacob, M., Heat Transfer, Vol. I, John Wiley and Sons, Inc., New York, 1949.

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FENSTER'S THESIS:
TRANSIENT THERMAL RESPONSE OF A STEP-PRESSURIZED
BOILING LIQUID NITROGEN SYSTEM

A. CURRENT STATUS OF WORK

The calorimeter tanks have been received from the fabricator and the installation of the instrumentation has been initiated. The primary electrical control equipment and switching, guard-heater-control equipment and switching, and power-measuring equipment and switching have been mounted and wired. The installation of thermocouples and heating ribbon has been begun.

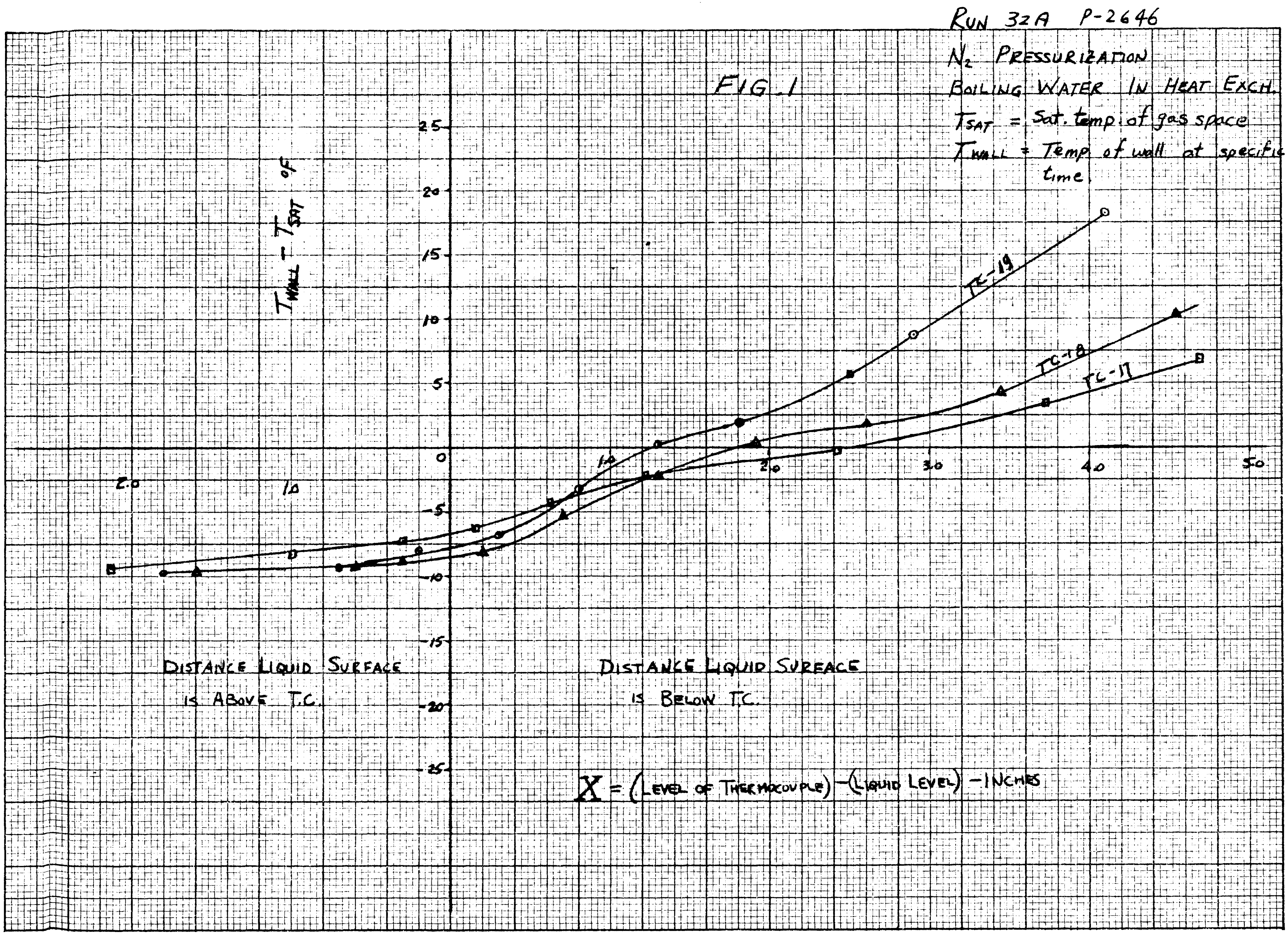
The heater-control system and design will permit variable heat fluxes up to 12,500 Btu/hr-ft^2 to be attained through the walls of the primary boil-off and pressurization cylinder. The response of the boiling liquid nitrogen system to a rapid pressurization will be investigated as a function of magnitude of pressurization and heat flux.

During the course of preparation for the experimental work, certain technical problems related to low-temperature research were encountered. A good electrical insulator was required to isolate the electric heating ribbon from the cylinder walls, and at the same time withstand the extensive low temperatures of the system. Such a substance was found to be mylar tape.

To be sure of accurate location of thermocouples located in the liquid nitrogen, a stainless-steel cage was designed and fabricated for position and support. It is expected that any uncertainty as to location will be eliminated.

B. EXPECTED STATUS AS OF JUNE 30, 1958

It is expected that experimental operation will be begun with some data collected by the end of the next period.


RUN 32A P-2646
FIG. 1
N_2 PRESSURIZATION
BOILING WATER IN HEAT EXCH.
T_{SAT} = Sat. temp. of gas space
T_{WALL} = Temp of wall at specific time
$T_{WALL} - T_{SAT}$ °F
25
20
15
10
5
0
-5
-10
-15
-20
-25
2.0
1.0
1.0
2.0
3.0
4.0
5.0
TC-19
TC-18
TC-17
DISTANCE LIQUID SURFACE IS ABOVE T.C.
DISTANCE LIQUID SURFACE IS BELOW T.C.
X = (LEVEL OF THERMOCOUPLE) - (LIQUID LEVEL) - INCHES

33E P-2646

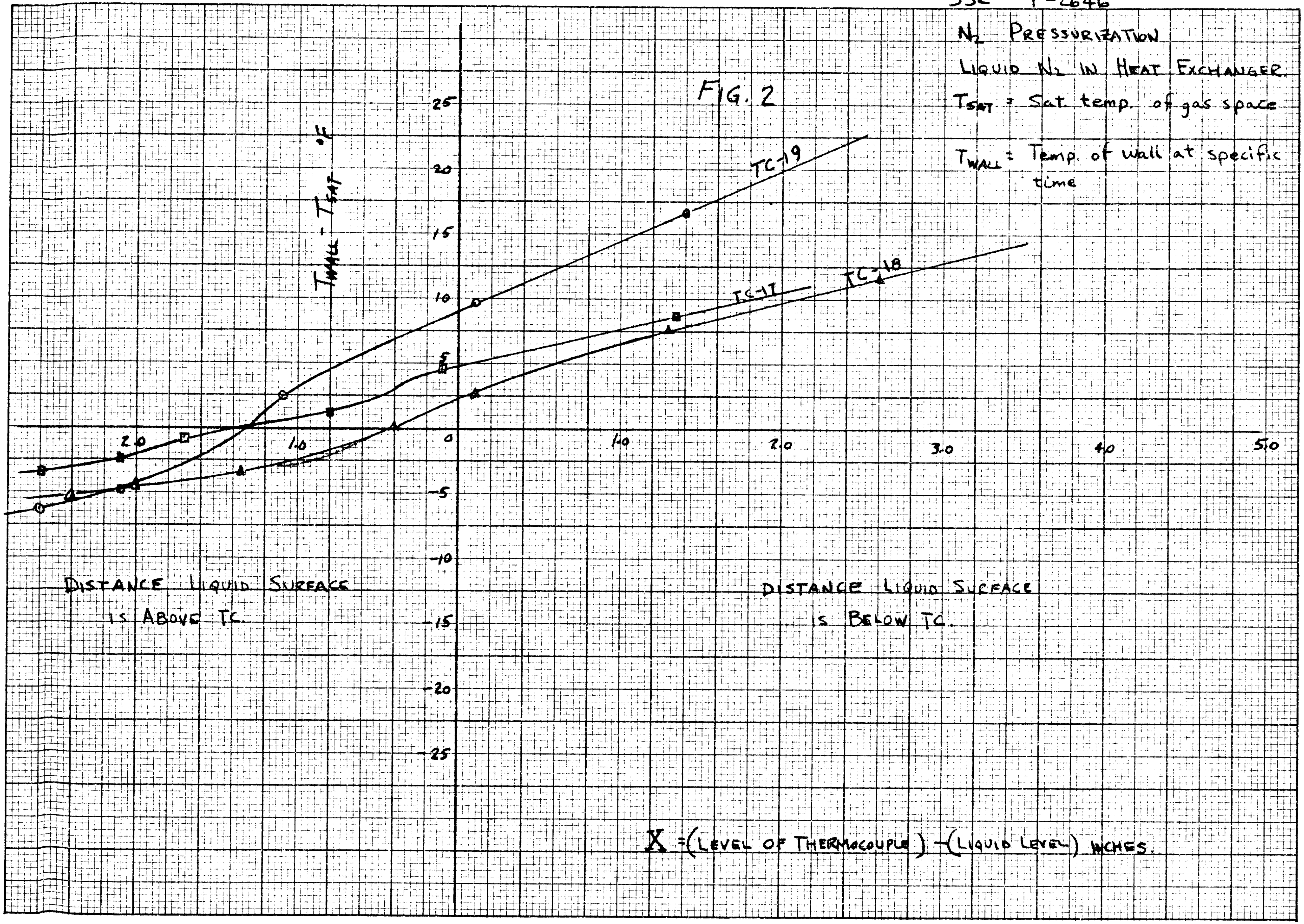
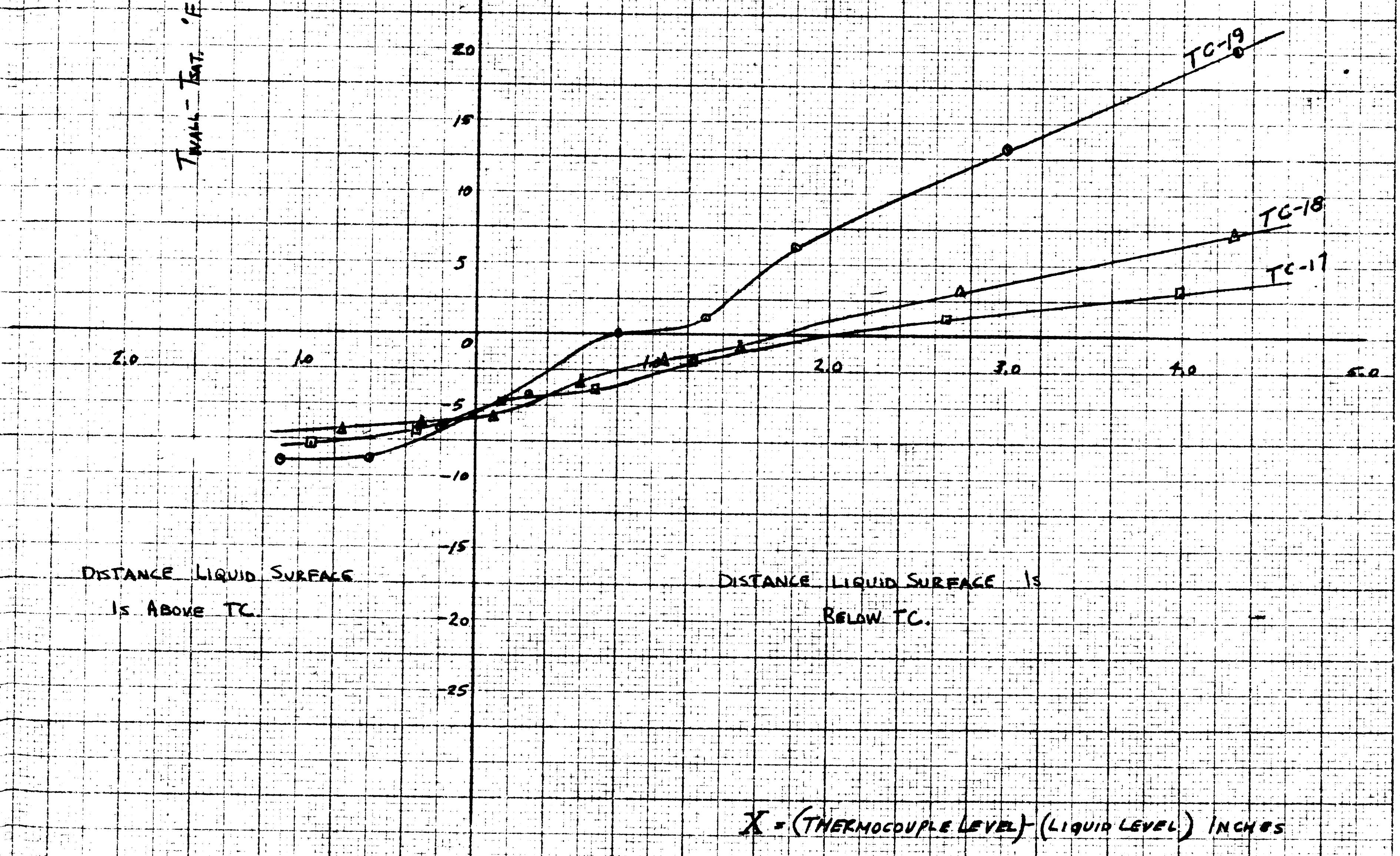


FIG. 2

FIG. 3

RUN 33B P-2646
He PRESSURIZATION
BOILING WATER IN HEAT EXCHANGER



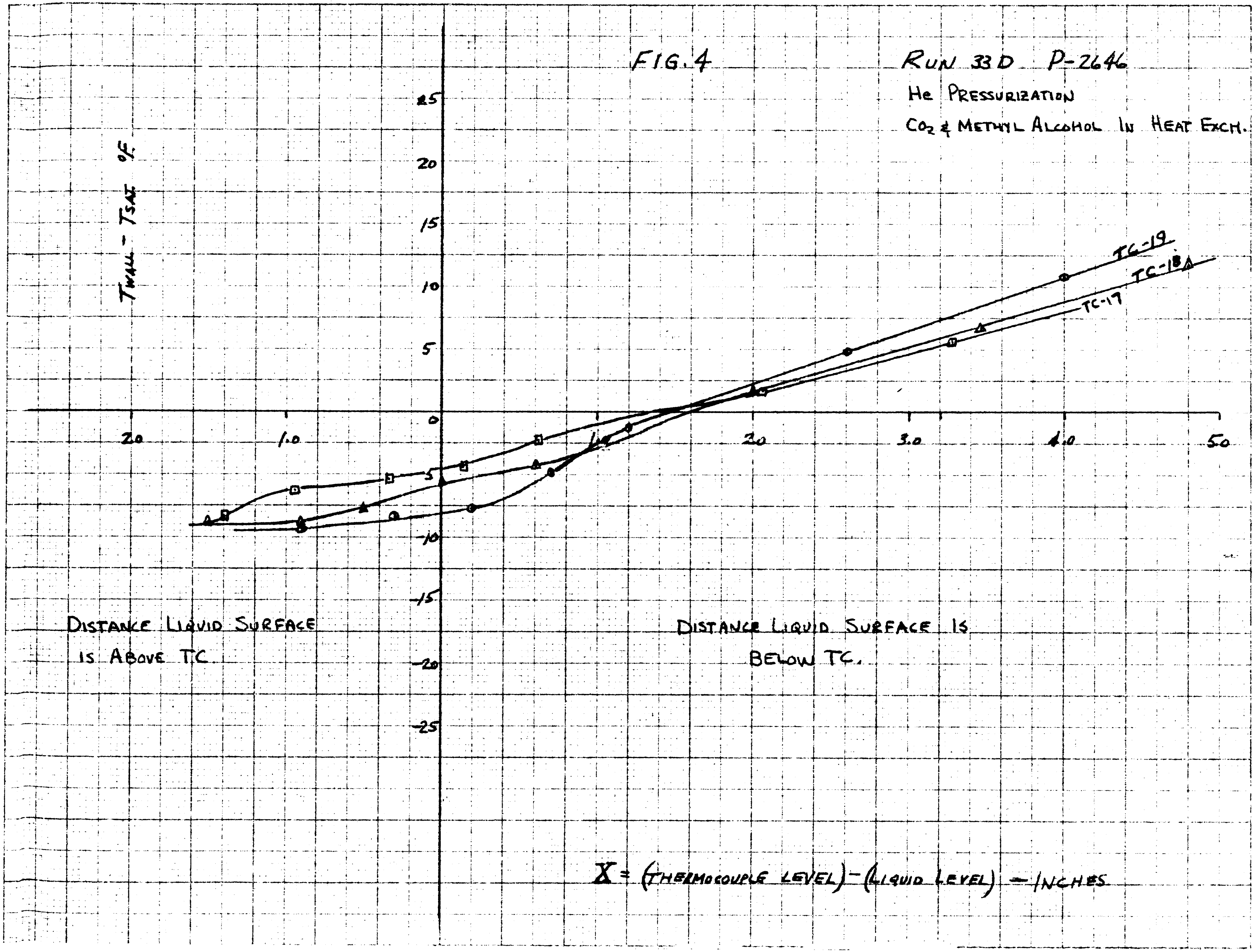
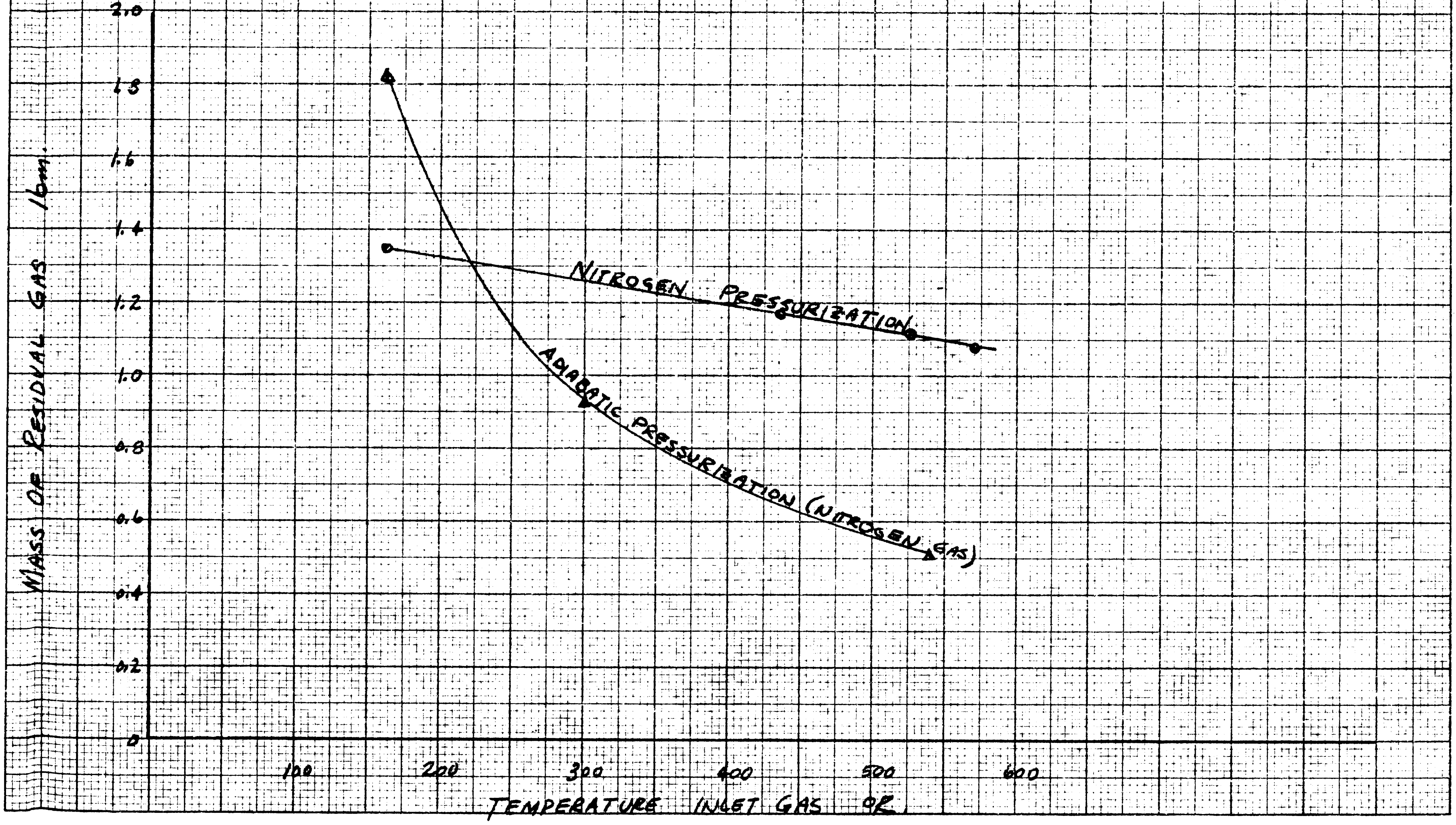


FIG. 4
RUN 33D P-2646
He PRESSURIZATION
CO_2 & METHYL ALCOHOL IN HEAT EXCH.
TWALL - TSAT °F
25
20
15
10
5
0
-5
-10
-15
-20
-25
2.0
1.0
1.0
2.0
3.0
4.0
5.0
TC-19
TC-18A
TC-17
DISTANCE LIQUID SURFACE IS ABOVE TC
DISTANCE LIQUID SURFACE IS BELOW TC.
X = (THERMOCOUPLE LEVEL) - (LIQUID LEVEL) - INCHES

FIG. 5

PROJECT 2646

EFFECT OF INLET GAS TEMPERATURE ON RESIDUAL GAS MASS IN TANK.



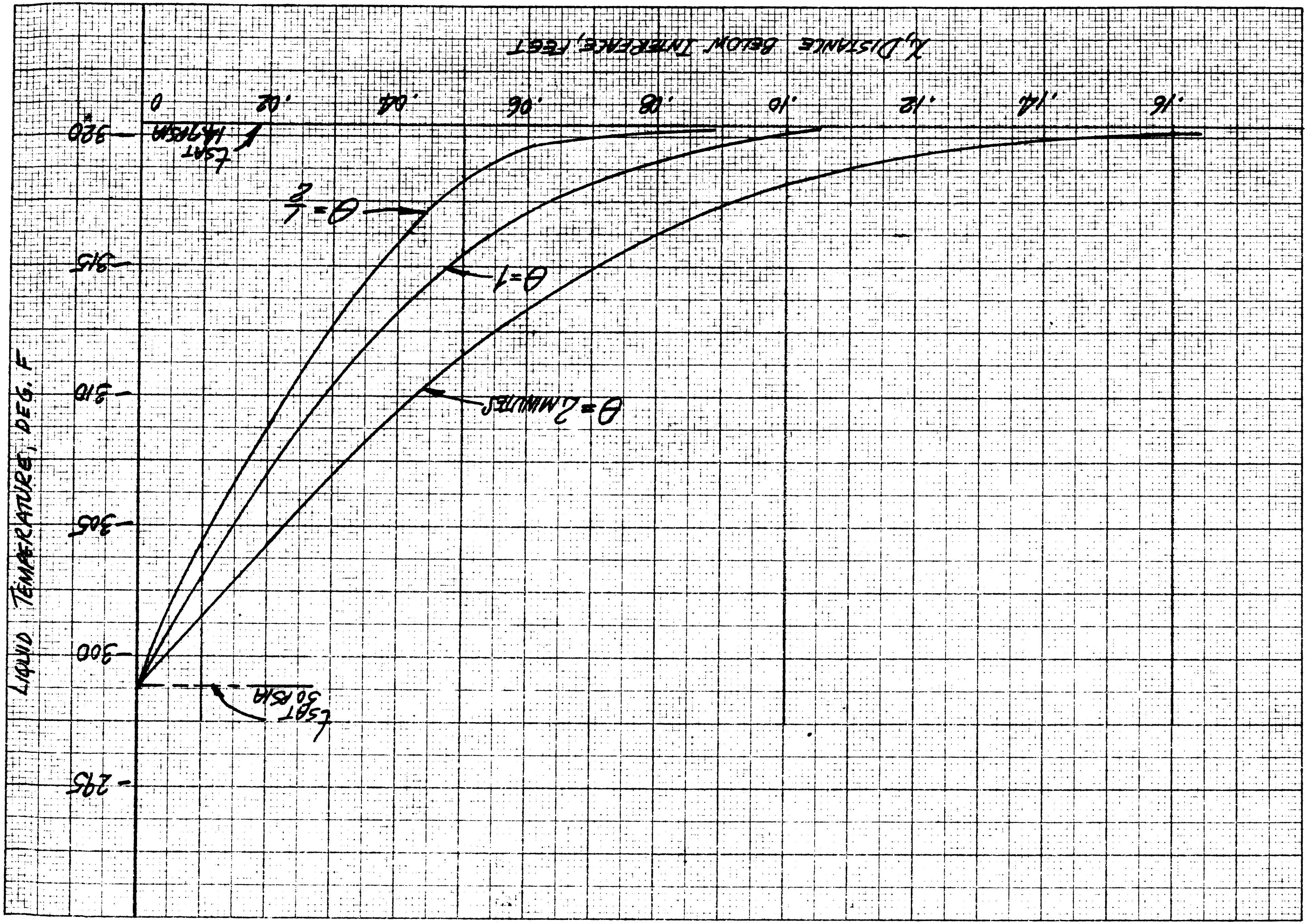


FIG. 6. PROJECT 2646