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The Heat Transfer and Thermodynamics Laboratory

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PRESSURIZATION OF LIQUID OXYGEN CONTAINERS

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ABSTRACT

Work has continued on the assembly and initial operation of a test tank for the study of the effects of wall heat flux and various insulation configurations on the mass of residual gas. Experimental difficulties with alternating voltage pick-up in the thermocouple wires, structural weakness of Styrofoam, and problems with the level indicator have been encountered and resolved.

Analytical studies of the final mean gas density in an insulated pressurized-discharge container are made. Results are reported in several graphs and compared both with experimental measurements and calculations from an adiabatic pressurized-discharge process.

Studies on the influence of acceleration on boiling have been completed.

I. STUDY OF RESIDUAL GAS MASS

A. EXPERIMENTAL APPARATUS

Efforts have been directed toward the elimination of experimental problems in the operation of the new system. At present most of the difficulties have been eliminated except those associated with the new design of the level indicator. As soon as reliable operation is obtained, the collection of data on residual gas mass will be begun. The principal difficulties encountered have been a-c pickup in the thermocouple circuits from the a-c heaters, structural failure of Styrofoam under pressure, and spurious and uncertain electrical contact on the level indicator. The latter of these is the only one not satisfactorily corrected to date.

The induced alternating current in the thermocouple circuits was the result of the a-c heater circuits which were installed on the tank. The manifestation of this induced voltage is shown in Fig. 1-a which is an oscillographic recording of the thermocouple response when the heaters are on. Figure 1-b is a recording of the response of the same thermocouples after corrective measures had been taken. These corrective measures included installing grounded shielding on all thermocouple wires external to the tank and twisting together each pair of thermocouple wires in the tank, thus reversing the phase of the induced voltages and producing a cancellation of the effect of the induced voltage. It is noted from Fig. 1 that these measures have in all cases reduced the induced voltages to an insignificant amount and in most cases completely eliminated them.

The failure of the Styrofoam float was due to a combination of low temperature and hydrostatic pressure in excess of its compressive yield strength when pressurizing the system. Each time the system was pressurized, some of the closed cells in the Styrofoam would collapse. Upon repeated pressurization the Styrofoam lost its buoyancy to the point at which it became unsatisfactory as a float. This problem was circumvented for the present by abandoning Styrofoam as a buoyant material and adopting a balsa wood float with the grain sealed to prevent absorption of liquid nitrogen. For the future runs with the insulating piston, a different type of Styrofoam (HD-2) will be incorporated into the design. Type HD-2 Styrofoam has the same thermal properties but a higher density (4.5 compared with 1.7 lbm/ft^3) and higher compressive yield point (120-140 as compared with 16-35 psi).

The difficulties with the electrical contacts on the level indicator were caused by the formation of ice and solid CO_2 on the wires and contacts to form an electrical insulation and insufficient contact pressure. To eliminate the formation of ice and solid CO_2 at the contact points and on the wires, a program has been devised whereby the system is dried and purged of all foreign gas previous to a run. To increase the contact pressure, one method tried was to unbalance the float slightly. This gave it a cocking movement which enabled the contacts to exert more pressure on the wire and also to provide for an increasing scraping action. This measure proved to be inadequate. At the moment, a new design of contact is being tried which will increase reliability of electrical contact. Results from this design will be available in the next few days.

A design in which the pressurizing gas is completely surrounded on the inside of the tank with Styrofoam (HD-2) has been made. Inquiries concerning a supply of shaped Styrofoam indicate that it is available locally at a reasonable cost.

A Weston Wattmeter (model 310) has been obtained from University instrument supplies to measure input power to the heaters. This instrument has an accuracy of 0.25% full-scale reading. Another instrument which is being provided by the University for use in this work is a 36-channel, Model 1012 Visicorder, Minneapolis-Honeywell Direct Recording Oscillograph. This instrument, which was received on July 20, will considerably increase both the measuring sensitivity and the number of separate temperatures that can be simultaneously detected and recorded.

B. AN ANALYSIS OF THE THERMODYNAMIC BEHAVIOR OF A PRESSURIZING GAS IN A CLOSED CONTAINER HAVING ONE RECEDING BOUNDARY

Progress Report No. 16 (June, 1959) presented a preliminary analysis of the thermodynamic response of a pressurizing gas in a closed container having one receding boundary resulting from the transfer of heat with all the internal surfaces which it wets. The results of this preliminary study were based on the assumption that the heat-transfer surfaces behave as semi-infinite solids in contact and exchange heat with a gas through a coefficient of heat transfer in a transient process. The influence of the receding boundary which continuously exposed a fresh, cold surface to the pressurizing gas was included. It was also assumed that the initiating temperature difference in this transient process did not vary with time and was equal to the difference between the temperature of the inlet gas and that of the liquid. The consequence of this is estimated to be a somewhat greater cooling effect than would be found were a system similar to this model actually used. A greater cooling effect would then result in higher predicted final gas density and greater residual mass than anticipated in a physical prototype.

The consequence of a model consisting of a semi-infinite solid is ignoring any influence of the ambient. This results in a compensated effect. At high inlet gas temperatures, the ambient has a reduced influence on the final density of the pressurizing gas owing to the lower temperature difference for heat flow to the gas. On the other hand, previous experimental results obtained with an imposed heat flux at the wall indicated the influence of local boiling in the liquid and the pumping of saturated vapor into the gas space which tended to increase the mass of residual gas. It appears of considerable importance to reduce this last effect but to promote the former, i.e., heat transfer between gas and ambient (providing, of course, the inlet gas temperature is at or below that of the ambient). At low inlet gas temperatures, heat transfer with the ambient is quite significant in reducing the quantity of residual mass. These effects are discussed below in relation to the results of the theoretical study.

In Progress Report No. 16* the spacially mean temperature of the pressurizing gas $\bar{t}(\theta)$ was shown to be a function of time by the following equation:

$$\frac{\bar{t}(\theta) - t_i}{t_o - t_i} = \frac{\bar{h}_g}{(w_i/A)c_p}\left\{\frac{e^{a_p\theta}}{a_p\theta}\,\mathrm{erfc}\sqrt{a_p\theta} - \frac{1}{a_p\theta} + \frac{2}{\sqrt{a_p\theta\pi}}\right.$$

$$+ \frac{e^{a_T\theta}}{a_T\theta}\,\mathrm{erfc}\sqrt{a_T\theta} - \frac{1}{a_T\theta} + \frac{2}{\sqrt{a_T\theta\pi}}$$

$$+ \frac{4(Y_p/D)}{(a_w\theta)^2}\left[e^{a_w\theta}\,\mathrm{erfc}\sqrt{a_w\theta} + 2\sqrt{a_w\theta/\pi}\right. \qquad (1)$$

$$\left.\left. + 4/3\sqrt{(a_w\theta)^3/\pi} - a_w\theta - 1\right]\right\}$$

This result indicates the cumulative effect on $\bar{t}(\theta)$ of heat trasfer with the container top, side walls, and receding interface. Of particular interest is the partial of fractional influence of heat transfer with each of these surfaces on the mean gas temperature, $\bar{t}(\theta)$. This has been computed from the ratio of the terms in the braces in Eq. (1) relating separately to the top, side, and receding surface to the sum of all the terms in the braces. The results are given in Fig. 2 for a container all of whose surfaces are made of "Styrofoam," $\rho = 1.768$ lbm/ft^3, $c_p = 0.27$ Btu/lbm-°F, $k = 0.0233$ Btu/hr-ft-°F. This selection of a material was made since it represents a commercially available substance having the physical properties of a thermal insulator. Two values of the heat-transfer coefficient, $\bar{h}_g$, between the gas and the solid have been taken, namely, 2 and 10 Btu/hr-ft^2-°F. It is not known exactly what the numerical value of this coefficient should be or even if it is constant, but it seems probable from free convection calculations that an effective value should lie between 2 and 10.

Of significance in the results in Fig. 2 is the greatly predominant influence on $\bar{t}(\theta)$ of heat transfer with the walls of the container in comparison

*See Eq. (35), p. 9, Progress Report No. 16, June, 1959.

with that of both the top and the receding boundary. At the end of a discharge time of 150 seconds, 80% of the heat transfer has been with the walls and only a total of 20% has been with the top and the receding boundary together. While it is important to remember that the model for this analysis is idealized to permit its mathematical treatment, the results shown in Fig. 2 are probably realistic. These show the importance of the wall heat transfer on final gas temperature, mean density, and residual mass, and suggest that it will be most fruitful to devote design refinements to considerations of either increasing or decreasing the heat-flow interaction between the walls and the gas. In general terms, if the pressurizing gas is at a temperature below that of the ambient, heat flow with the ambient (through the walls) ought to be encouraged, while if the pressurizing gas is at a temperature above that of the ambient, the opposite condition tends toward optiminzation.

Also of interest in Fig. 2 is the relatively minor influence produced in the fractional heat transfer for systems with different gas-solid heat-transfer coefficients in the range 2 to 10 Btu/hr-ft^2-°F.

In Figs. 3 and 4 the transient response of the "Styrofoam" system is shown for a period of time of from 0 to 150 seconds. For these results a pressurizing gas flow rate of 40 lbm/hr is assumed. This represents an average flow rate for the experimental runs previously completed and reported. The results shown in Figs. 3 and 4 also have been computed for the two values of gas-solid heat-transfer coefficient, 2 and 10 Btu/hr-ft^2-°F.

The results in Fig. 3 have been computed using Eq. (1). The curves of Fig. 4 are taken from the data of Fig. 3 for T_o = 140°R (-320°F) and T_i = 460°R (0°F). These indicate a rapidly rising initial gas temperature during the first 30 seconds of discharge followed by a slowly decreasing temperature during the remainder of the discharge period. As would be anticipated for these conditions, the higher heat-transfer coefficient produces a lower gas temperature. The gas temperature corresponding to zero time ($\theta = 0$) in Fig. 4 cannot be computed from the theory, Eq. (1), owing to a mathematical singularity at that point in time, but from physical reasoning it seems reasonable that $\bar{t}(\theta)$ should be equal to t_o(140°R). The curves of Fig. 4 also appear to converge to this value at zero time.

As a means toward computing the final mean gas density, $\bar{\rho}$ (at 120 seconds) and consequently the mass of residual gas as a function of pressurizing gas temperature T_i, the curves of Fig. 5 were computed from the results of Fig. 3 for a time θ equal to 120 seconds, t_o = -320°F (140°R) and t_i = 0°F (460°R). The influence of the same two values of the gas-solid heat-transfer coefficient $\bar{h}_g$ is shown and is not great.

The mean gas density $\bar{\rho}$ at the end of a 120-second discharge period as computed from the theory is given in Fig. 6 as a function of inlet gas temperature T_i, for values of the gas-solid heat-transfer coefficient of 2 and 10 Btu/hr-ft^2-°F. As found in the results of Figs. 4 and 5, the influence of h_g on $\bar{h}_g$ in the range shown is not especially great. The results of the computation of $\bar{\rho}$ indicate that, as the inlet gas temperature is increased, the final mean gas density is decreased, as might be expected. The mass of residual gas also would decrease.

The results of the theory are compared with the experimental results from a thin-walled aluminum tank with direct contact between liquid and gas and with the calculated results, assuming adiabatic conditions between the gas and all surfaces it wets. The results of the theory shown in Fig. 6 compare very well with the calculations made assuming perfectly adiabatic conditions between the pressurizing gas and the walls. Ignoring any influence of the ambient (which is inherent in the theory), it is the adiabatic case which produces the minimum possible final mean gas density and consequently the minimum residual mass of gas. Any departure from the adiabatic condition, that is, a heat exchange from the gas to the container walls, should result in a lowered gas temperature, larger mean gas density, and of course an increased residual mass of gas. Such a result is predicted by the theoretical calculations shown in Fig. 6. Furthermore, as would be expected, these results fit in well with those of the adiabatic case which were obtained from an independent calculation and are shown as the dotted curve in Fig. 6. At low inlet gas temperatures (about 150°R), there is less agreement between the theory and the adiabatic case. This results from the fact that the gas densities for the adiabatic case were taken from nitrogen gas property tables corresponding to 50 psia, the pressure of pressurization. In this case it is impossible actually to realize an inlet gas temperature as low as t_0, or -320°F, as the saturation temperature at 50 psia is about -300°F. The theory has been computed to inlet temperatures as low as t_0, however. In this region of temperature, the theory is less accurate as condensation from the gas doubtless would occur at the wetted surfaces. The residence time for such a condensate film could easily exceed the total discharge time of 120 seconds. Above inlet gas temperatures of about 185°R (-275°F), it is estimated that a condensate film will have a residence time less than 6 seconds for a Styrofoam surface,* and the theory ought to predict reasonable results.

Experimental data from runs with direct contact, pressurized discharge in a thin-walled aluminum tank exchanging heat with the ambient are also shown in Fig. 6. The theoretical results fall above and below these data. Owing to the influence of the ambient and the heat capacity of the aluminum walls of the tank, close agreement between the experimental and theoretical results cannot be expected. However, it is interesting that as reasonable a comparison is obtained as is found in Fig. 6. At temperatures above about 300°R (-160°F), the experimental results give higher mean gas densities than the theory. It is possible that this is the result of the heat capacity of the walls of the tank, heat transfer with the liquid interface (at -300°F), and the pumping of saturated vapor into the gas space owing to surface boiling in the liquid, all of which will tend to increase the gas density. Heat transfer with the ambient compensates these effects, but apparently it is not of a significant magnitude

*See Eq. (17), p. 6, Progress Report 15, March, 1959.

until the inlet temperatures are approximately 300°R (-160°F). Below this temperature the experimental data shows a lower mean gas density than is predicted by theory. It is expected that this is a result of heat exchange with the ambient.

From the experimental and theoretical results obtained to date, the following appear to be possible methods of reducing the residual gas mass:

(a) High inlet temperature of pressurizing gas;

(b) Reduce heat transfer between the pressurizing gas and all wetted surfaces, especially the container walls;*

(c) Reduce ambient heat transfer with liquid to prevent surface boiling and the pumping of saturated vapor into gas space;

(d) Use pressurizing gas of lowest possible molecular weight; and

(e) Level of pressure of pressurizing gas as low as possible.

C. WORK DURING THE NEXT PERIOD

It is anticipated that data will be collected and reduced during the next period for programmed runs with and without the insulating piston for various heat flux inputs to the system. In addition, present plans include the incorporation of a design into the system in which the pressurizing gas is completely surrounded by a Styrofoam insulation. Data from this system will be compared with those from the uninsulated partially insulated systems.

Additional theoretical studies on residual gas mass will be undertaken to show the influence of the wall heat capacity for various materials and the effect of the ambient. A study on bubble rise velocity and lifetimes in liquid nitrogen will be started.

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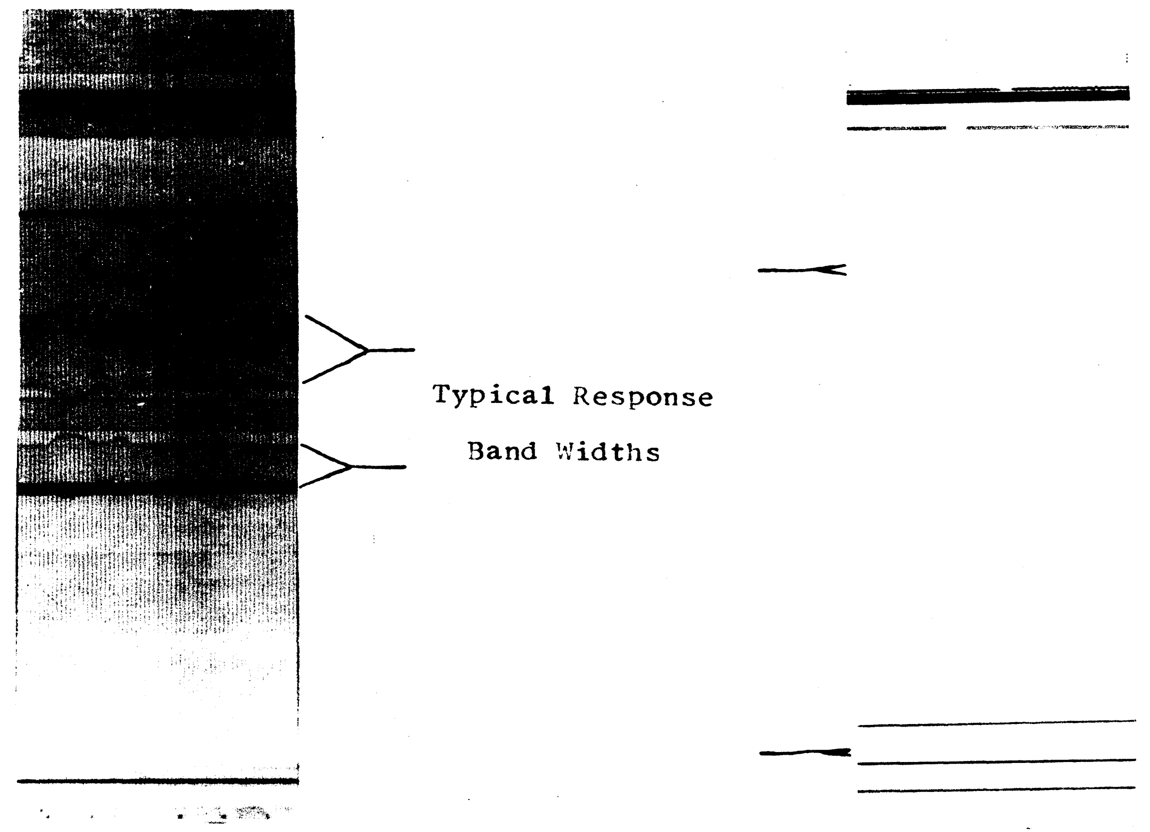
*If ambient heating of the container is sufficient to heat the walls to a temperature above that of the pressurizing gas or if very low pressurizing gas inlet temperatures are used, restrictions on method (b) can be relaxed.

II. A STUDY OF POOL BOILING IN AN ACCELERATING SYSTEM

CURRENT STATUS OF THE WORK

The initial phase of this study is now completed. The ranges of variables covered were listed on p. 11 of Progress Report 16, June, 1959. Correlations of the data and the writing of the final report are in progress.

It is expected that this report will be issued as Technical Report No. 3 some time during August, 1959.



A. Before noninductive twisting of thermocouple wires in tank

B. After noninductive twisting of thermocouple wires to reduce AC pickup

Figure 1. Thermocouple response with AC heaters on

FRACTIONAL CONTRIBUTION OF THE TOTAL HEAT TRANSFERRED FROM THE PRESSURIZING GAS TO CONTROL SURFACE AS A FUNCTION OF TIME FOR AN ALL STYROFOAM CONTAINER

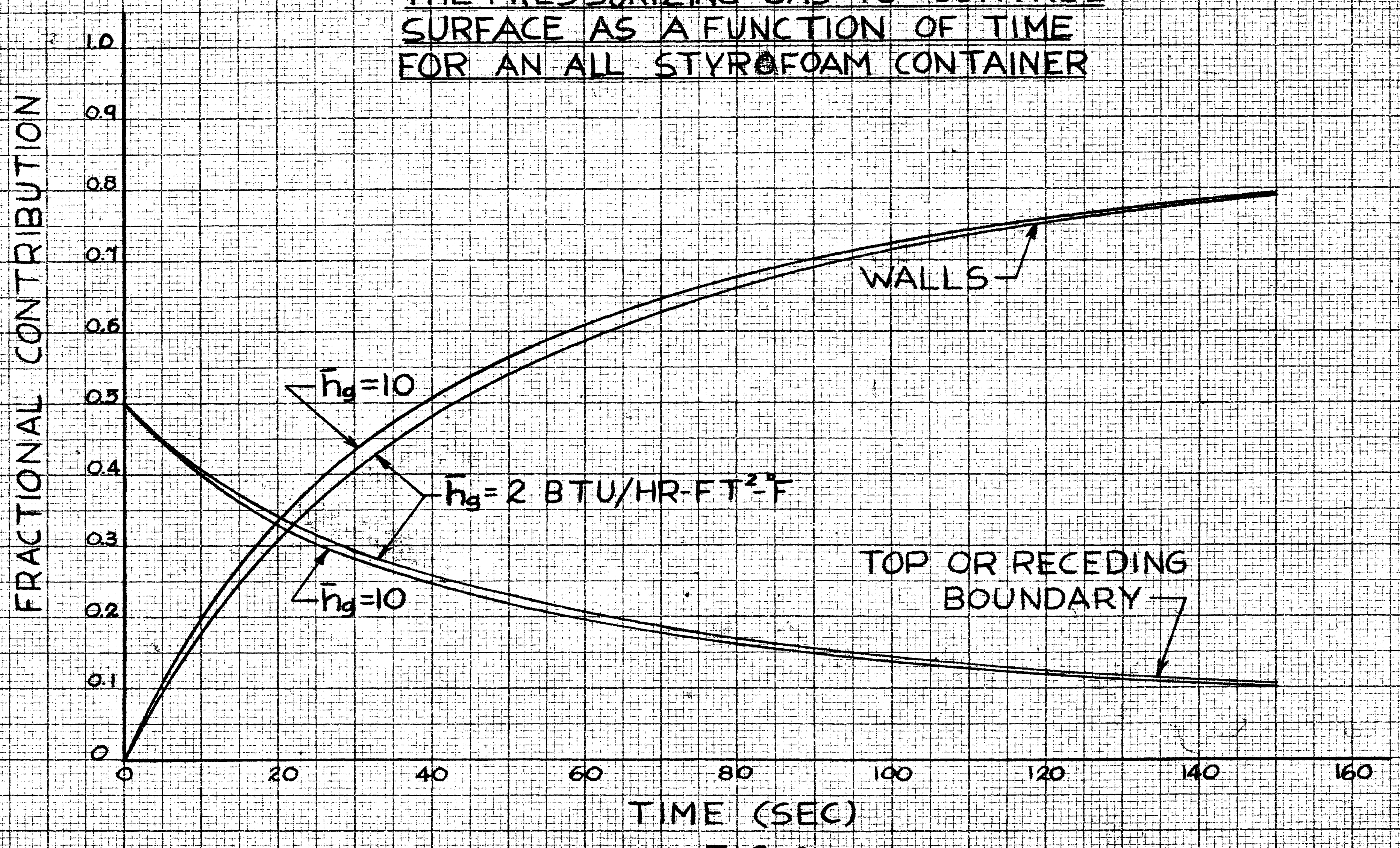


FIG. 2

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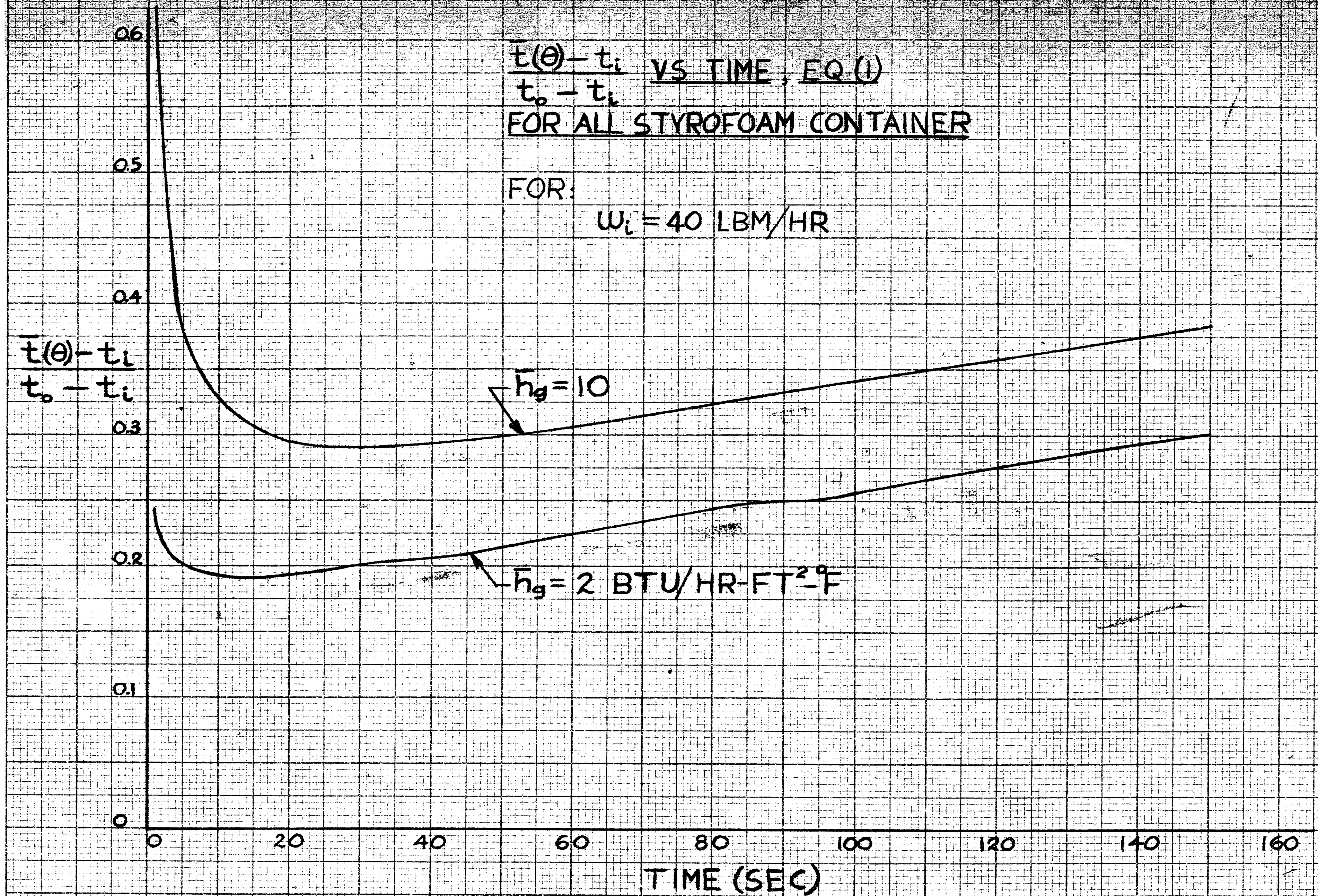

FIG.
FIG. 3

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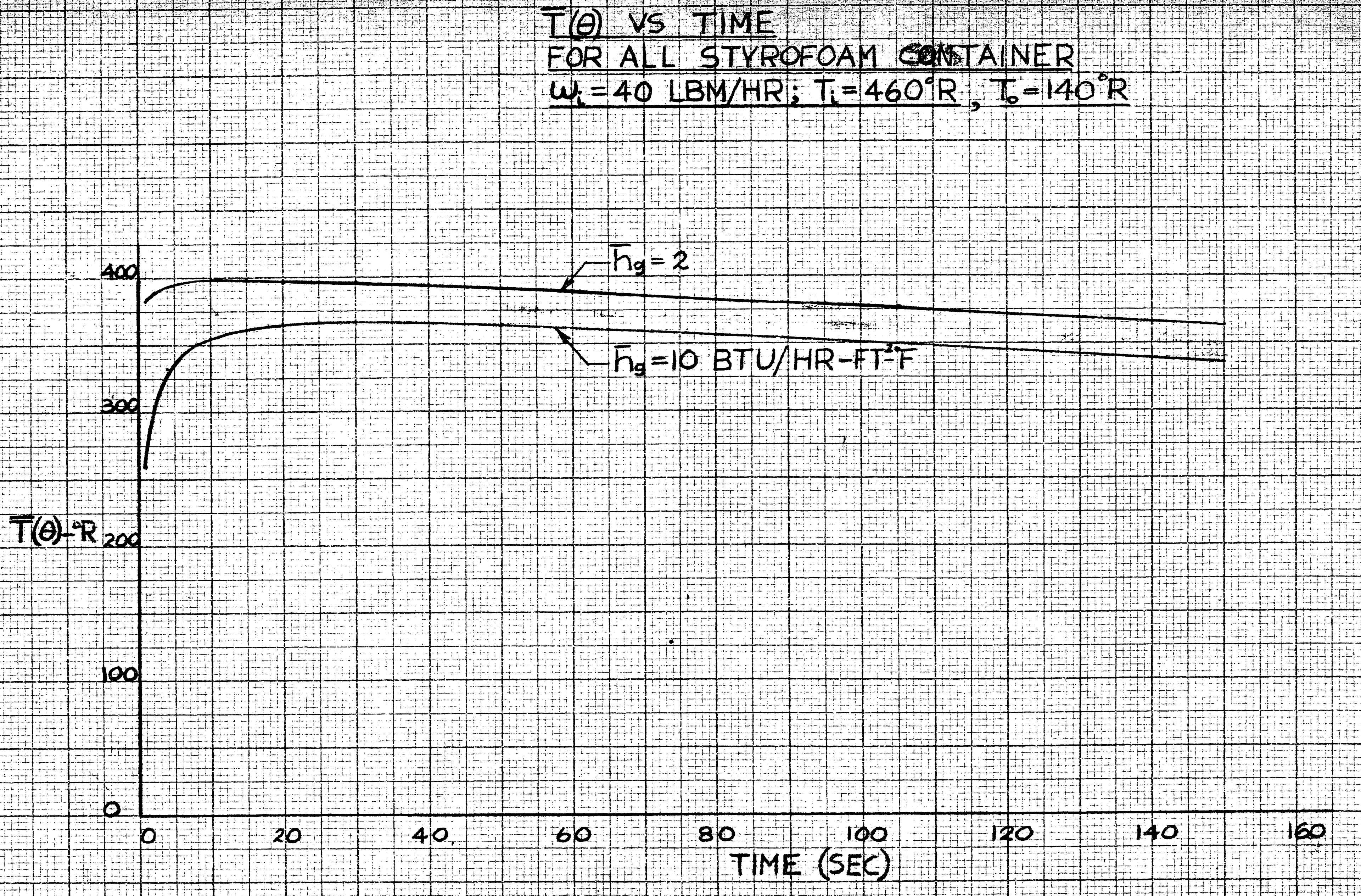

T(θ) VS TIME
FOR ALL STYROFOAM CONTAINER
w_i = 40 LBM/HR; T_i = 460°R, T_o = 140°R
$\bar{h}_g$ = 2
$\bar{h}_g$ = 10 BTU/HR-FT²-F
$\bar{T}(\theta)$-°R
400
300
200
100
0
0
20
40
60
80
100
120
140
160
TIME (SEC)

FIG. 4

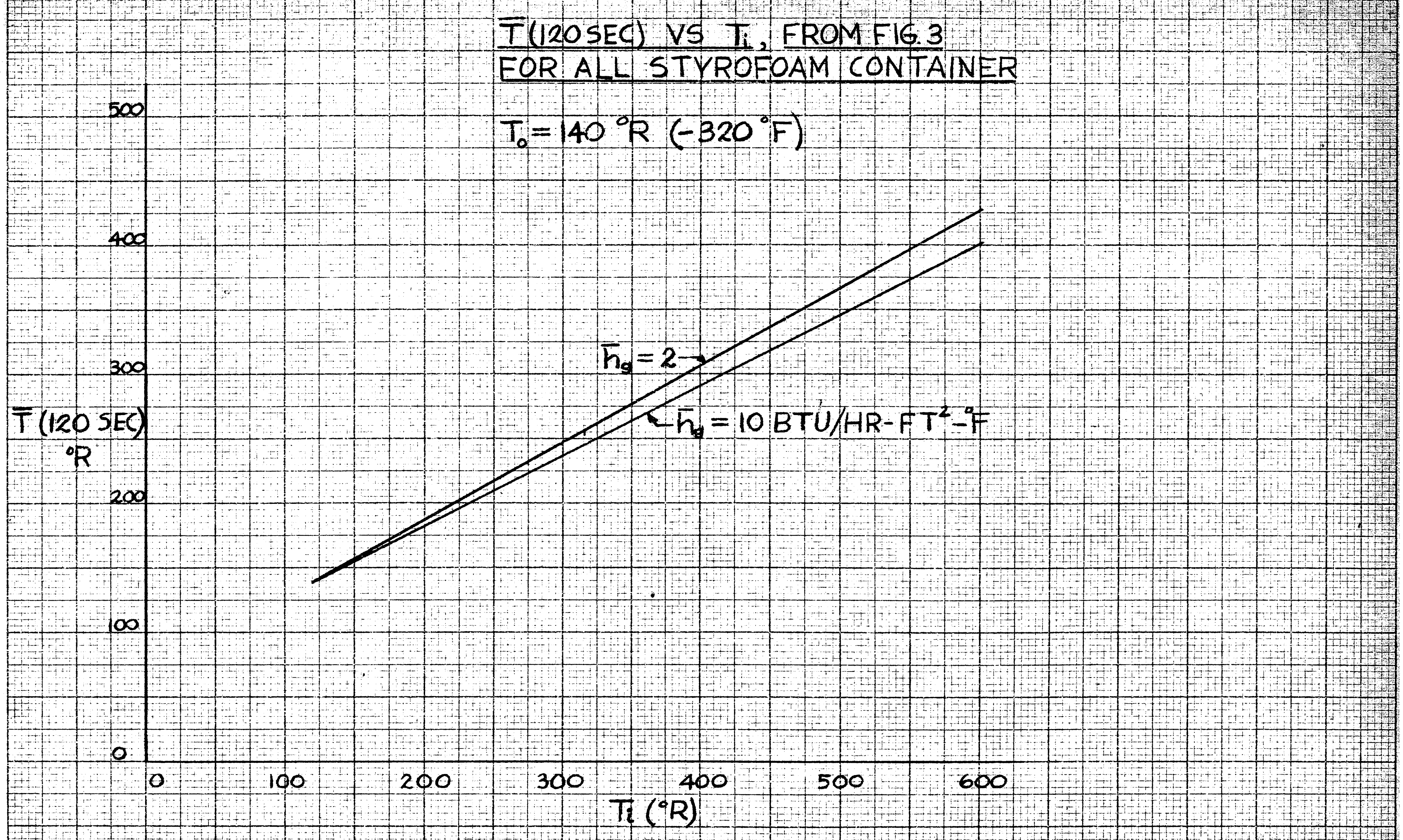
$\overline{T}$ (120 SEC) VS T_i, FROM FIG. 3
FOR ALL STYROFOAM CONTAINER
T_o = 140 °R (-320 °F)
$\overline{h}_g$ = 2
$\overline{h}_g$ = 10 BTU/HR-FT^2-°F
$\overline{T}$ (120 SEC) °R
0
100
200
300
400
500
600
T_i (°R)

FIG. 5

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FINAL MEAN DENSITY $\bar{\rho}$ OF PRESSURIZING GAS AS A FUNCTION OF INLET GAS TEMPERATURE T_i

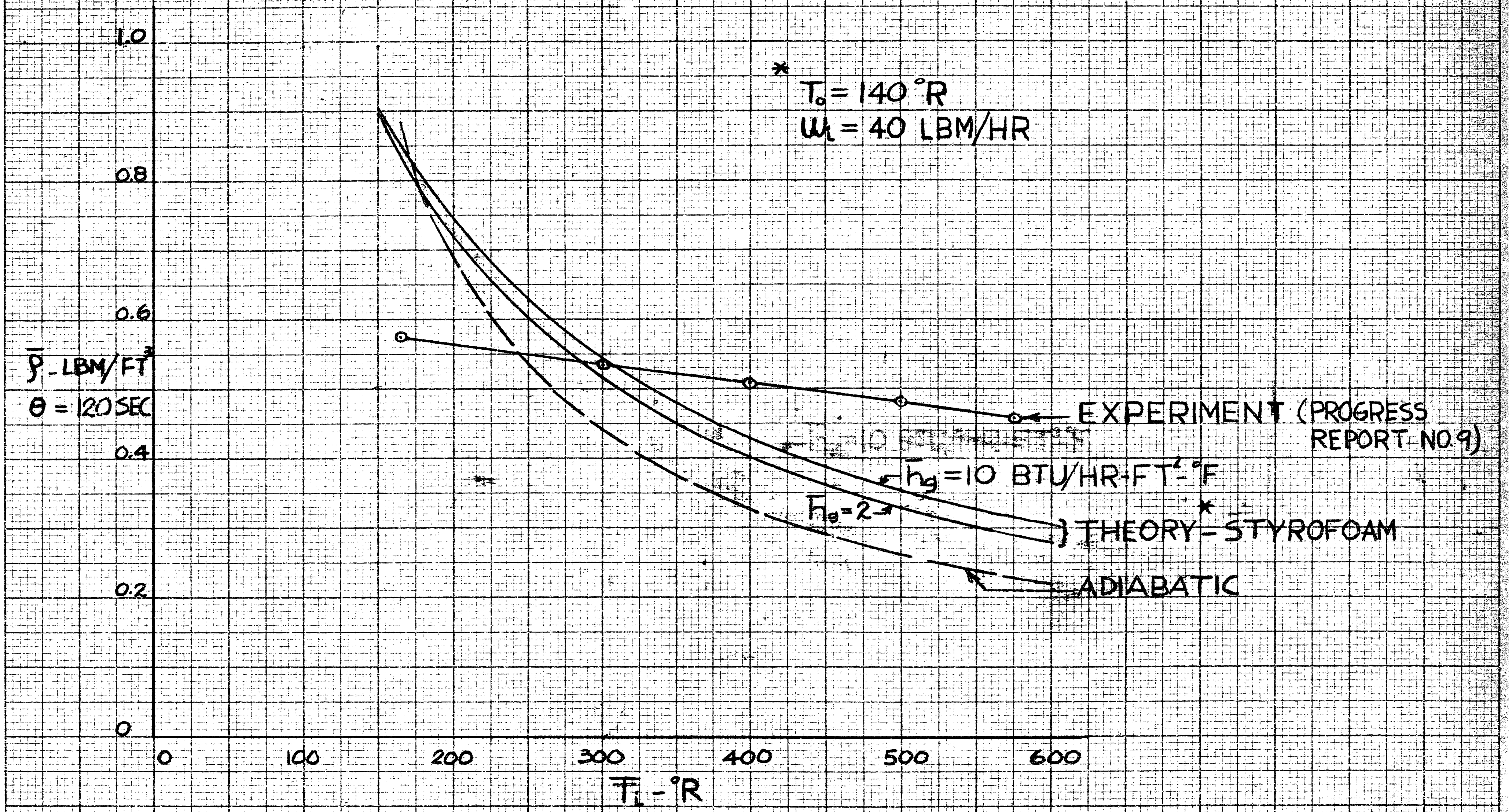

FIG. 6

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M. N.