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STUDY OF THERMAL CONDUCTIVITY REQUIREMENTS

MSFC 20-Inch and 105-Inch Cryogenic Tank Analyses

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FOREWORD

This report represents the results of work performed by the Thermal Environment Section of the Aero-Mechanics Department of the Lockheed Missiles & Space Company, Huntsville Research & Engineering Center, for the National Aeronautics and Space Administration, Marshall Space Flight Center, Huntsville, Alabama under contract NAS8-21347. The NASA contract monitor is John G. Austin, Jr., of the MSFC Astronautics Laboratory.

SUMMARY

Two cryogenic storage tanks 20 and 105 inches in diameter, respectively, were analyzed using a Lockheed-developed Thermal Analyzer Computer program. This program is suited to the thermal analysis of venting cryogenic tanks in that it incorporates not only the natural convection along the walls of the tanks but also the thermal effects of transfer of the venting gas. For the purpose of comparison, three methods were used in analyzing each of the tanks. These methods were:

- The venting gas was assumed to be stationary, and its temperature was constrained only at the liquid surface and at the top of the vent line.
- The venting gas was assumed to be stationary, and its temperature was fixed at the evaporation temperature at all points.
- The venting gas was assumed to be flowing out the vent line at the rate dictated by boiloff, and its temperature was constrained only at the liquid surface and the top of the vent line.

For the 20-inch tank the heat leak to the liquid for the three methods of analysis were 7.58, 4.15 and 5.79 Btu/hr, respectively. Test data for the 20-inch tank revealed a mass boiloff rate of 0.0508 lbm/hr which corresponds to a heat leak to the liquid of 4.38 Btu/hr.

For the 105-inch tank, the heat leak to the liquid for the three methods of analysis were 133.37, 79.52 and 85.99 Btu/hr, respectively. No test data are yet available.

These results indicate that the approximations made in the analysis concerning the venting gas has a significant effect on the predicted heat rate. The capability of treating the thermal effects of mass flow is a needed tool in analyzing venting cryogen tanks.

CONTENTS

NOMENCLATURE

A	area (ft^2)
C_p	specific heat (Btu/lbm-^oF)
Gr_x	Grashof number $= \dfrac{\rho^2 g \beta (T_w - T_g) x^3}{\mu^2}$
h	heat transfer coefficient (Btu/hr-ft^2-^oF)
h_{fg}	heat of vaporization of nitrogen (Btu/lbm)
k	thermal conductivity (Btu/hr-ft-^oF)
Nu_x	Nusselt number $= \dfrac{hx}{k}$
Pr	Prandtl number $= \dfrac{C_p \mu}{k}$
q	heat rate (Btu/hr)
Q	boiloff heat rate (Btu/hr)
R	thermal resistance between nodes (hr-^oF/Btu)
T	temperature (^oF)
$\dot{m}$	boiloff mass flow rate (lbm/hr)
x	vertical distance from liquid level (ft)
ΔT	temperature differential through insulation (^oF)
ΔX	insulation thickness (in.)
ρ	density (lbm/ft^3)
g	gravitational acceleration (ft/sec^2)
β	coefficient of thermal expansion ($^oF^{-1}$)

NOMENCLATURE (Continued)

μ	viscosity (lbm/ft-sec)

Subscripts

conv	convection
w	wall
g	gas
n	n^{th} node

Section 1
INTRODUCTION

The true effectiveness of high performance insulation (HPI) in actual applications cannot be determined simply by idealized calorimeter tests of the materials. In actual applications the thermal conductivity tends to be greater than the predicted value because of gaps, joints, penetrations, uncontrollable compression, offgassing and other effects. Therefore, full and subscale tests of insulated cryogenic tanks are being conducted. However, costs would be excessive to conduct tests for all future cryogenic tank applications. Therefore, it will be advantageous if adequate analytical, rather than experimental, prediction techniques could be used. Before an analytical prediction could be considered adequate, however, it is necessary to conduct comparisons between analytical and experimental predictions of the same tanks. If the analytical predictions do indeed match test data for a variety of tank designs, then, in the future, confidence could be placed in the predictions alone, making them a more valuable tool for use in design work.

Section 2
TECHNICAL DISCUSSION

2.1 THE NASA/MSFC 20-INCH CRYOGENIC TANK

2.1.1 Experimental Results

The NASA/MSFC 20-in. cryogenic tank, shown schematically, in Fig. 1, was tested in a vacuum facility at MSFC. The test was documented in Ref. 1. and the following information is found therein. The tank was insulated with an average of 1.37 in. of 1/4-mil double-aluminized Mylar and red polyurethane foam insulation. After 96 hours of testing, the nitrogen gas boiloff was steady at 0.0508 lbm/hr. The chamber pressure was estimated to be between 1×10^{-7} and 1×10^{-5} torr Hg. The insulation backside pressure was estimated to be between 1×10^{-3} and 50×10^{-3} torr Hg. The boiloff gas was maintained at slightly above atmospheric pressure ($\approx$ 15.25 psia) by a throttle valve and was measured to be room temperature at the vent exit.

Based on the steady state boiloff rate the heat rate to the nitrogen can be determined.

$$Q = \dot{m} h_{fg} = 0.0508 \times 86.2 = 4.38 \text{ Btu/hr.}$$

2.1.2 Analytical Results

The NASA/MSFC 20-inch cryogenic tank was analyzed using a Lockheed-developed Thermal Analyzer computer program. Figure 1 shows a schematic of the tank. The liquid level was chosen by setting the gas fraction in the top of the tank at approximately 5% which is consistent with the experimental data. Only the portions of the tank above the adiabatic separator, including

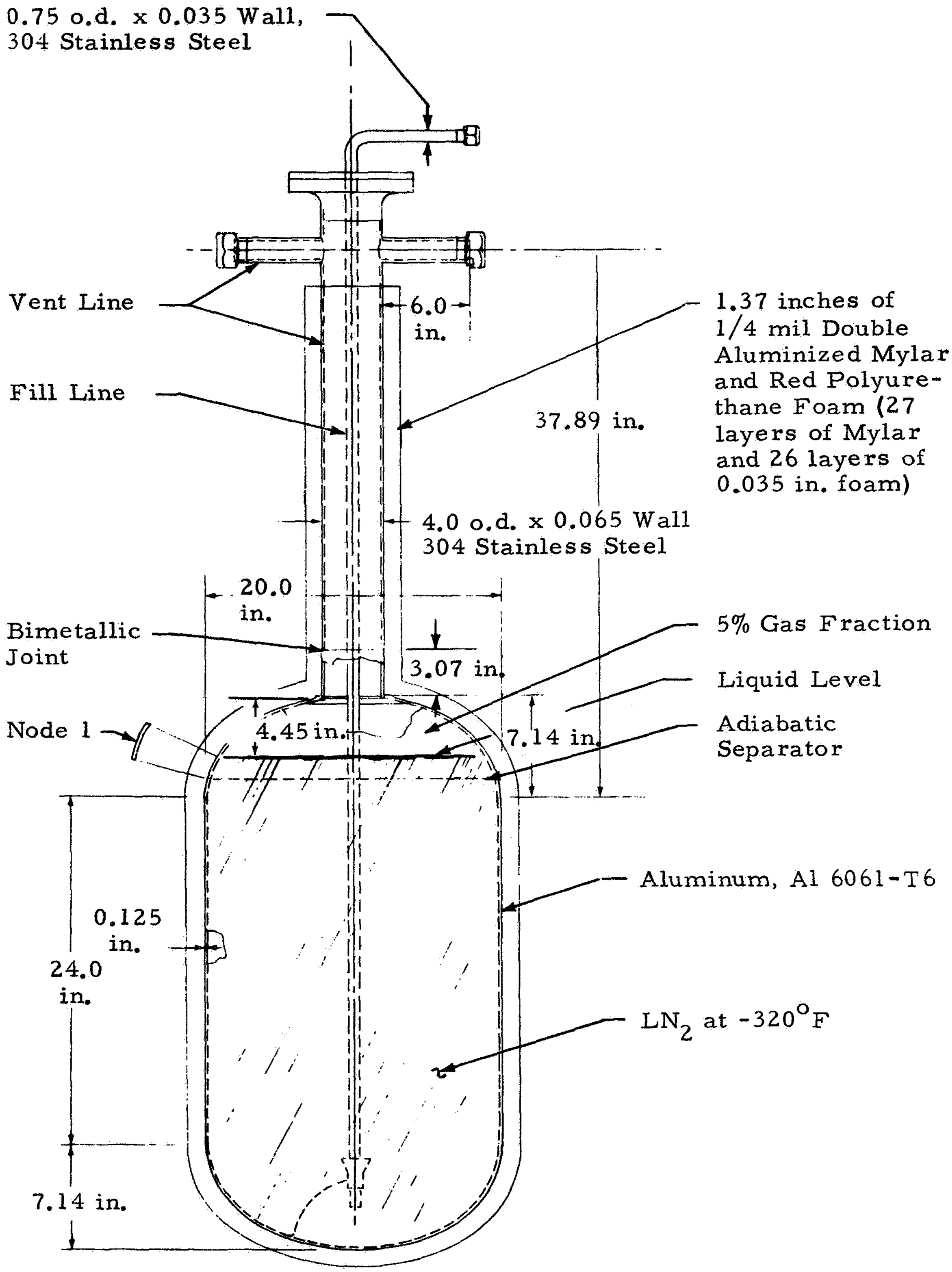

Fig. 1 - Schematic of MSFC 20-Inch Cryogenic Tank

the neck, were analyzed. The heat flow in regions of the tank below the adiabatic separator was assumed to be one-dimensional and, therefore, not in need of computer analysis. Since there are no penetrations to the tank other than the neck, heat flow in the portions of the tank below the separator are assumed to be perpendicular to the insulation layers and the tank wall itself. No heat is expected to flow down the tank wall since the uniform liquid temperature holds the thin-walled tank at a uniform temperature. Therefore, the heat rate to the liquid below the separator would be

$$q = kA \frac{\Delta T}{\Delta X}$$

where

k = the insulation's effective thermal conductivity

A = the insulation area (below the separator)

ΔT = the temperature differential through the insulation

ΔX = the insulation thickness

The critical item in attaining the correct q through that portion of the insulation is that the proper value for k be chosen. The other parameters A, ΔT and ΔX are known within close tolerence relative to k and are listed below:

$A = 15.41\ ft^2$ (calculated from tank dimensions)

$\Delta T = 70^o - (-320) = 390^oF$

$\Delta X = 1.37$ in.

The insulation used on the tank test was the 1/4-mil double aluminized Mylar and red polyurethane foam material. At the average temperature of the insulation (-125^oF) the experimental value of thermal conductivity is

$5 \pm 1 \times 10^{-5}$ Btu/hr-ft-^{o}F. This results in a heat flux to the liquid below the separator of 2.19 rate ± 0.44 Btu/hr.

The remaining heat rate to the liquid is that entering above the separator which includes flow down the tank wall, down the fill line and conduction down the gas. All these heat rates are calculated in the thermal analysis. The analysis was performed assuming that the tank was insulated with 1.37 in. of the double aluminized Mylar and red polyurethane foam HPI. The thermal conductivity normal to the insulation was assumed to be $5 \pm 1 \times 10^{-5}$ Btu/hr-ft-^{o}F while the thermal conductivity parallel to the insulation was assumed to be 1.55×10^{-2} Btu/hr-ft-^{o}F. Temperature-dependent thermal conductivity values for the aluminum tank, the stainless steel fill and vent lines and the nitrogen gas were used in the analysis. All the heat flow paths in the analysis were conduction except those convection heat paths connecting the gas to the tank wall and to the fill and vent lines. The convection heat flow paths (thermal resistors) were

$$R_{conv} = \frac{1}{hA} \tag{2}$$

where

h = the convection heat transfer coefficient

A = heat flow area

The heat transfer coefficient was obtained from the laminar and turbulent natural convection equations on a vertical wall. (Ref. 2)

Laminar

$$Nu_x = 0.508 \left(\frac{Pr}{0.952 + Pr} \quad Gr_x Pr \right)^{1/4} \tag{3}$$

Turbulent

$$Nu_x = \frac{0.0295\ (Gr_x)^{2/5}\ Pr^{7/15}}{\left(1 + 0.495\ Pr^{2/3}\right)^{2/5}} \tag{4}$$

In the test, the tank contained liquid nitrogen which boils at -320°F. The gas properties were evaluated for gaseous nitrogen at -280°F. A value somewhat higher than -320°F was chosen since the gas temperature will increase as it passes up the vent lines. The error caused by not using temperature dependent gas properties is insignificant. Therefore, the temperature at which the properties are evaluated is not critical. The resulting equations were:

$$Gr_x = 3.45 \times 10^8\ (T_w + 320)\ x^3 \tag{5}$$

Laminar

$$h = 0.294(T_w + 320)^{1/4}\ \ x^{-1/4}\quad Btu/hr\text{-}ft^2\text{-}{}^oF \tag{6}$$

Turbulent

$$h = 0.323(T_w + 320)^{2/5}\ \ x^{1/5}\quad Btu/hr\text{-}ft^2\text{-}{}^oF \tag{7}$$

The transition Gr number was set at 10^9 in the analysis. In both of these equations, h is dependent on the surface temperature, T, and the distance, x, along the surface from the leading edge (the liquid surface).

In order to treat the temperature and position dependence in Eqs. (6) and (7), they were evaluated at five different values of axial position, x, up the neck of the tank. The x positions were 3.52, 10.6, 17.6, 24.6 and 31.7 in. Each axial node in the analysis was assigned one of these x values. The step transition from Eq. (6) to Eq. (7) was prescribed by the point at which Gr $\geq 10^9$.

Treatment of conduction and convection by the thermal analyzer was straightforward. One aspect of the physical system, however, was not so easy to model. This was the heat transfer effect of mass flow of the boiloff gas. The measured boiloff rate of the liquid was 0.0508 lbm/hr. From this the heat flux up the vent line due to mass flow can be calculated.

$$q(x) = \dot{m}\, C_p\, T_g(x)$$

$$= \left(0.0508\ \frac{\text{lbm}}{\text{hr}}\right)\left(0.25\ \frac{\text{Btu}}{\text{lbm}\,^{\circ}\text{F}}\right)\ T_g(x)\ (^{\circ}\text{F})$$

$$= 0.0127\ T_g(x)\ \text{Btu/hr.} \tag{8}$$

Neither the 3-D IPM nor any other thermal analyzer available at HREC at the time was capable of handling this mass flow option. Therefore, two limiting cases were solved to bracket the real problem. At the same time, the Lockheed/Huntsville Computer program was modified to handle Eq. (8). A third case was then run on the mofified program to solve the "flowing gas" case. The three cases analyzed are:

1. The venting gas was assumed to be stationary, but its temperature constrined only at the liquid surface and the top of the next line.
2. The venting gas was assumed to be stationary, and its temperature was fixed at the evaporation temperature at all points.
3. The venting gas was assumed to be flowing out the vent line at the rate dictated by boiloff, and its temperature was constrained only at the liquid surface and the top of the next line.

Case 1

The fallacy in Case 1 is that the heat exchanger effect of the moving gas is ignored. The gas nodes reach a steady state temperature in the vent line at a value higher than the real case. The heat leakage to the liquid is calculated to be higher than the real case. The heat leakage to the liquid is itemized in Table 1. The errors are from the uncertainty of HPI thermal conductivity.

Table 1

ITEMIZED HEAT LEAKAGE TO LN_2 STORED IN MSFC 20-INCH TANK WITH 5% GAS FRACTION AND INSULATED WITH 1 INCH OF 1/4-MIL DOUBLE-ALUMINIZED MYLAR AND RED POLYURETHANE FOAM

Heat Path	Case 1 — Stationary Gas Nodes/Unconstrained Gas Temperature (Btu/hr)	Case 2 — Stationary Gas Nodes/Fixed Gas Temperature (Btu/hr)	Case 3 — (Mass Flow Treatment) (Btu/hr)
Stainless Steel Vent Line and Aluminum Tank Wall	5.17 ± 0.06	1.85 ± 0.04	3.42 ± 0.06
Stainless Steel Fill Line	0.05	0.00016	0.03
Boiloff Nitrogen Gas	0.05	—	0.03
HPI Over Node 1*	0.12 ± 0.02	0.11 ± 0.02	0.12 ± 0.02
HPI Below Node 1*	2.19 ± 0.44	2.19 ± 0.44	2.19 ± 0.44
Total Heat Rate to Liquid	7.58 ± 0.52	4.15 ± 0.50	5.79 ± 0.52

*See Fig. 1

The temperature distributions of the gas, the fill line and the vent line (tank wall) are shown in Fig. 2. Note that for each case the curve represents the fill and the vent line temperatures. Also, for Cases 1 and 2 the gas temperature is very close to this same temperature. Inspection explains why this is the case. The convective heat transfer of the gas is much greater than the pure conduction down the gas. Therefore, radial heat transfer is much greater than longitudinal with the result that at any longitudinal position the temperatures of all components are very nearly the same value.

Case 2

The error in Case 2 is obviously that the temperature increase of the gas as it ascends the vent annulus is ignored since the gas is held stationary and at a constant temperature in the analysis. The heat leakage to the liquid is itemized in Table 1. The temperature distributions of the fill line and the vent line (tank wall) are shown in Fig. 2. Note that the temperature distribution is flat in the vent and fill lines in all but the very top of the neck. This is due to the constant gas temperature approximation. The real case is expected to be somewhere between Case 1 and Case 2.

Case 3

This case most closely represents the real situation since the gas motion is considered in the heat balance. The resulting heat leak to the liquid is itemized in Table 1, and the temperature distribution in the tank is shown in Fig. 2. As expected, the distribution for Case 3 fell between the two "bracketing" cases. Also, the heat leakage to the liquid for Case 3 fell between that of the two others.

The three calculated values of total heat leakage to the liquid are now compared to the test value of 4.38 Btu/hr.

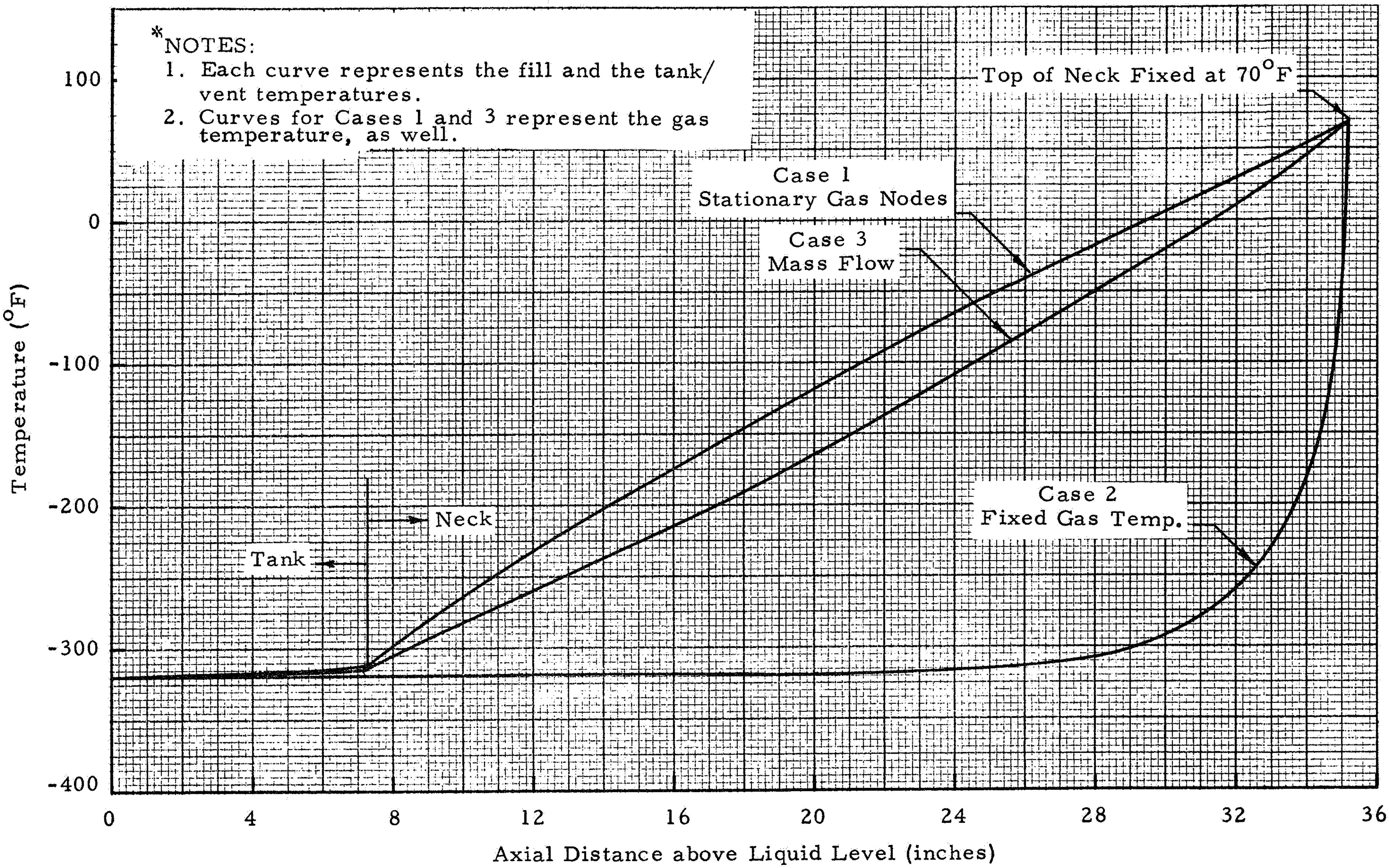

Fig. 2 - Plot of Temperature* vs Axial Distance above the Liquid for the NASA 20-Inch Cryogenic Tank Insulated with 1-Inch of Mylar and Foam and 95% Fill with LN_2

$$\text{Case 1: Percent difference} = \frac{7.58 - 4.38}{4.38} = 73\%$$

$$\text{Case 2: Percent difference} = \frac{4.15 - 4.38}{4.38} = -5.25\%$$

$$\text{Case 3: Percent difference} = \frac{5.79 - 4.38}{4.38} = 32.2\%$$

These results indicate that the method used does have a considerable effect on the calculated heat rate to the liquid. In this particular comparison, although the mass flow case was expected to produce the closest comparison, the fixed gas temperature assumption actually produced the closest. Two factors should be noted in judging this comparison. First, the accuracy of the analysis is based on the accuracy of the assumed thermal conductivity value, 5×10^{-5} Btu/hr-ft-^{o}F. The thermal conductivity of the mylar and foam composite is dependent on factors such as its preconditioning, its degree of outgassing and its compression. These and other effects can cause an error ($\pm$20%) in estimating the applied conductivity value of the HPI. The second factor is that there is some question concerning the reliability of the test data because of changes in the ullage pressure during the test.

The significant fact is that when a venting tank is analyzed, careful consideration should be given to the method of modeling the vent gas temperature. Different assumptions, although all reasonable, predict substantially different heat leaks.

2.2 THE NASA/MSFC 105-INCH CRYOGENIC TANK

The NASA/MSFC 105-in. cryogenic tank was analyzed using the Lockheed/Huntsville Computer program. Figure 3 shows a schematic of the tank. The liquid level was chosen by setting the gas fraction in the top of the tank at approximately 5%. Only the portion of the tank at and above the adiabatic separator, including the tank neck, was analyzed. The cryogen was LH_2 which has a boiling temperature of $-423^{\circ}F$. As in the case of the 20-in. tank, the heat flux to the liquid below the separator was calculated by hand.

$$q = kA \frac{\Delta T}{\Delta X}$$

where

$$k = 5 \times 10^{-5} \text{ Btu/hr-ft-}^{\circ}\text{F}$$

$$A = 239.63 \text{ ft}^2 \text{ (calculated from tank dimensions)}$$

$$\Delta T = 70 - (-423^{\circ}\text{F}) = 493^{\circ}\text{F}$$

$$\Delta X = 1 \text{ in.}$$

$$\therefore \; q = 70.8 \text{ Btu/hr}$$

The remaining heat flux to the liquid is calculated in the thermal analysis. The analysis was performed for 1 in. of the double aluminized Mylar and red polyurethane foam insulation. The thermal conductivity normal to the insulation was assumed to be 5×10^{-5} Btu/hr-ft-$^{\circ}$F while that parallel to the insulation

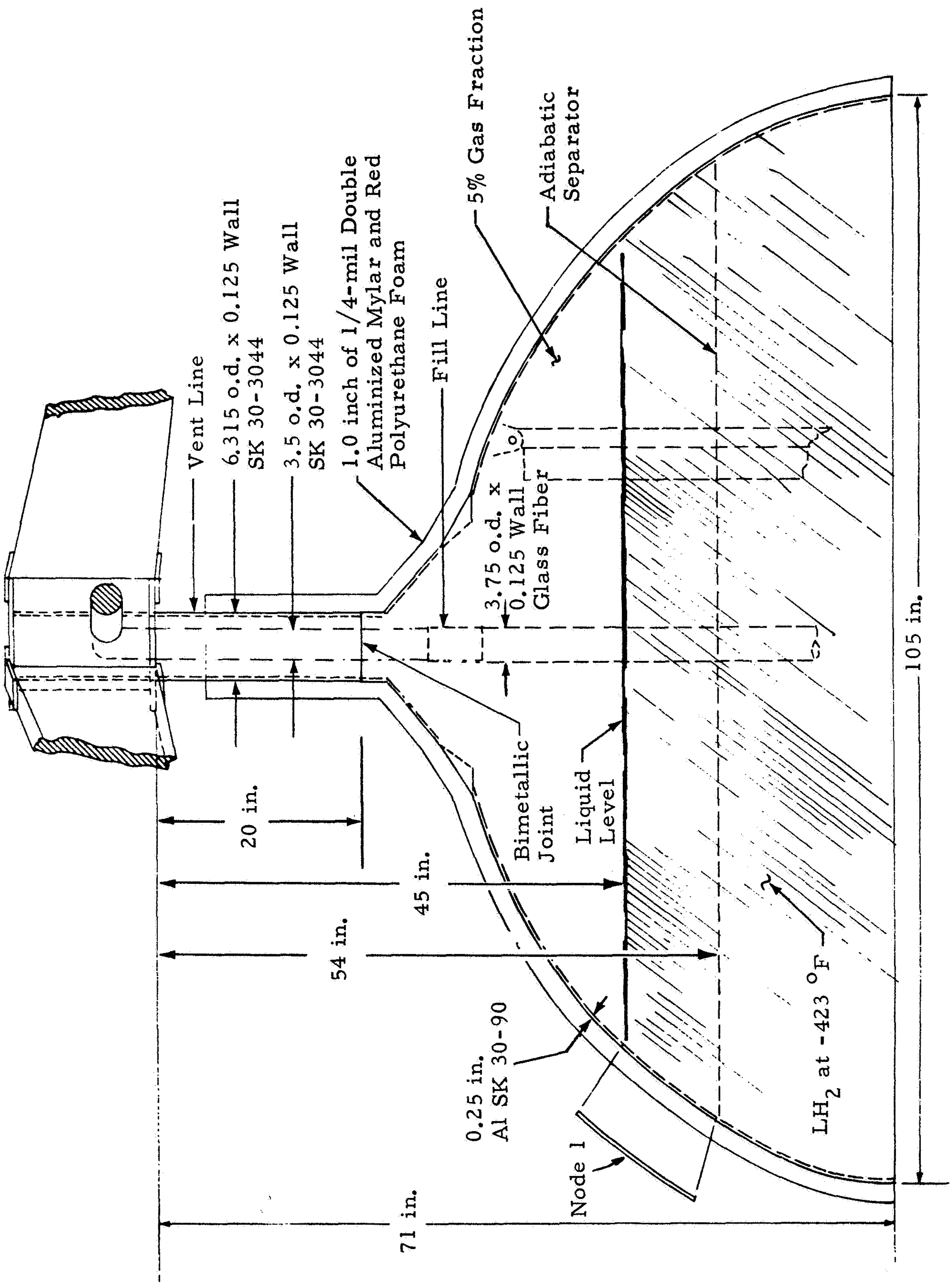

Fig. 3 - Schematic of MSFC 105-Inch Cryogenic Tank

was assumed to be 1.55×10^{-2} Btu/hr-ft-$^{\circ}$F. The analysis includes heat flux through the insulation as well as flux down the tank wall, the fill line, the gas and the instrumentation wires. Temperature-dependent thermal conductivity values for the aluminum tank, the stainless steel and fiberglas fill and vent lines, the copper liquid level sensor, the constantan lead wires and the hydrogen boiloff gas were used in the analysis. The curve used for the fiberglas is shown in Fig. 4. All the heat flow paths in the thermal analysis were the conduction type except the convection paths connecting the gas to the tank wall, the fill line and the instrumentation wires. The convection equations used were the laminar and turbulent natural convection equations on a vertical wall, Eqs. (3) and (4). These were evaluated for -400°F hydrogen gas to obtain:

Laminar

$$h = 0.922(T - 37)^{1/4} \, x^{-1/4} \text{ Btu/hr-ft}^2\text{-}^{\circ}\text{F}$$

Turbulent

$$h = 1.17(T - 37)^{2/5} \, x^{1/5}$$

The transition Gr number was fixed at 10^9 in the analysis. As for the 20-in. tank, the two equations were solved at five axial positions which, in this case were 4.5, 13.5, 23, 32 and 41 in., respectively. Each axial node in the analysis was assigned one of these x values.

As for the 20-in. tank, three cases were analyzed, two approximations and a "flowing gas" case. These three cases were:

- The venting gas was assumed to be stationary, and its temperature was constrained only at the liquid surface and at the top of the vent line.
- The venting gas was assumed to be stationary, and its temperature was fixed at the evaporation temperature at all points.
- The venting gas was assumed to be flowing out the vent line at the rate dictated by boiloff, and its temperature was constrained only at the liquid surface and the top of the vent line.

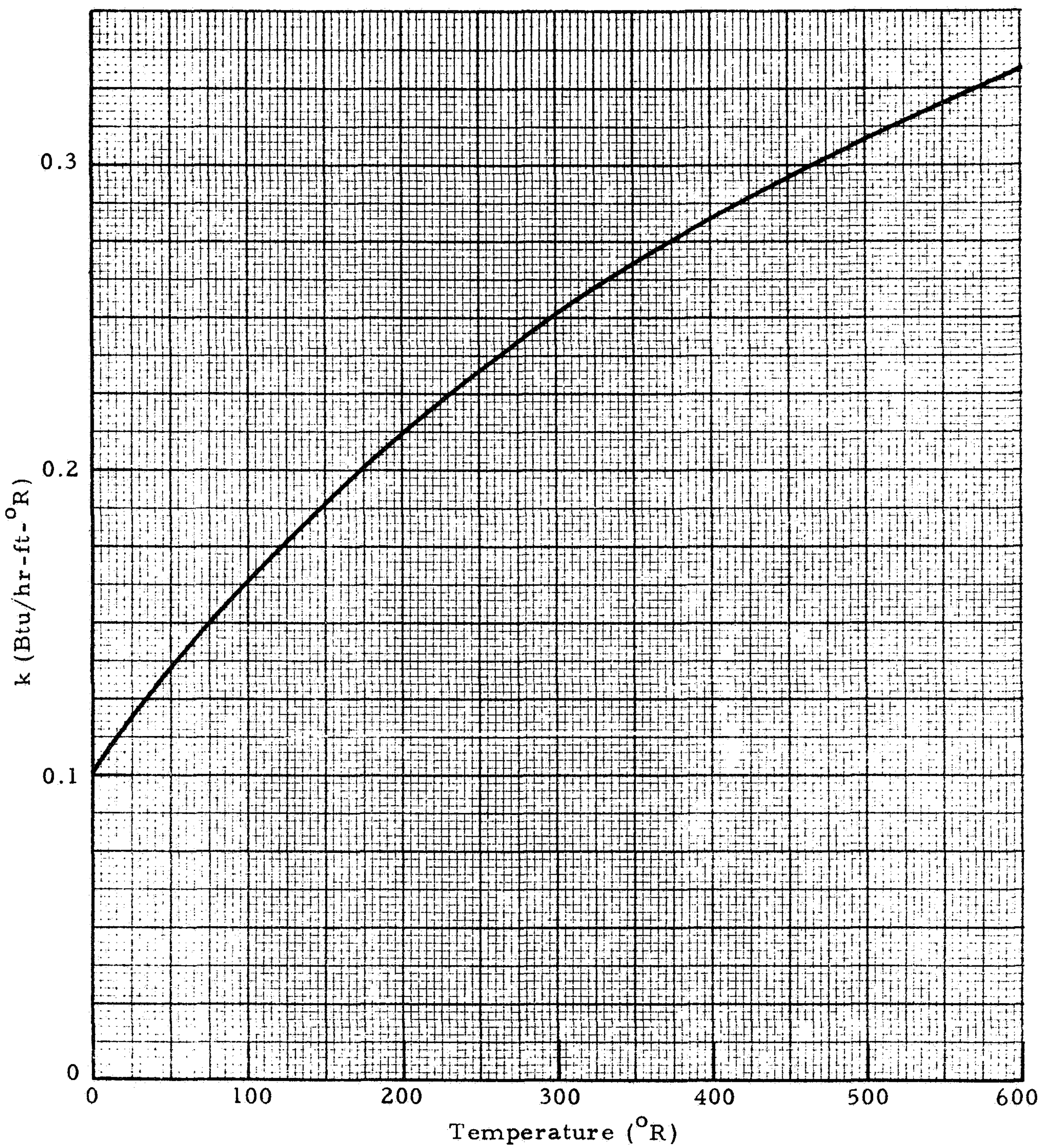

Fig. 4 - Thermal Conductivity of Fiberglas (Epoxy, Parallel Glass Fiber)

Results for all three cases are shown in Figs. 5 and 6 which present the temperature distributions for the vent and fill lines, respectively. The heat leakage to the liquid is itemized for each case in Table 2. As expected, the temperature distribution and heat leakage for Case 3 fell between the two "bracketing" cases.

Table 2

ITEMIZED HEAT LEAKAGE TO LH_2 STORED IN MSFC 105-INCH TANK WITH 5% GAS FRACTION AND INSULATED WITH 1 INCH OF 1/4-MIL DOUBLE-ALUMINIZED MYLAR AND RED POLYURETHANE FOAM

Heat Path	Case 1 – Stationary Gas Nodes/Unconstrained Gas Temperature (Btu/hr)	Case 2 – Stationary Gas Nodes/Fixed Gas Temperature (Btu/hr)	Case 3 – (Mass Flow Treatment) (Btu/hr)
Stainless Steel Vent Line and Aluminum Tank Wall	52.77	3.04	8.87
Stainless Steel and Glass Fiber Fill Line	0.03	0.000001	0.004
Boiloff Hydrogen Gas	3.97	0.0	0.62
Thermocouple Wires (100 Constantan Wires)	0.01	0.0	0.0008
Liquid-Level Sensor (Copper Wire)	0.07	0.000001	0.006
HPI Over Node 1*	5.72	5.68	5.69
HPI Below Node 1*	70.80	70.80	70.80
Total Heat Rate to Liquid	133.37	79.52	85.99

*See Fig. 3

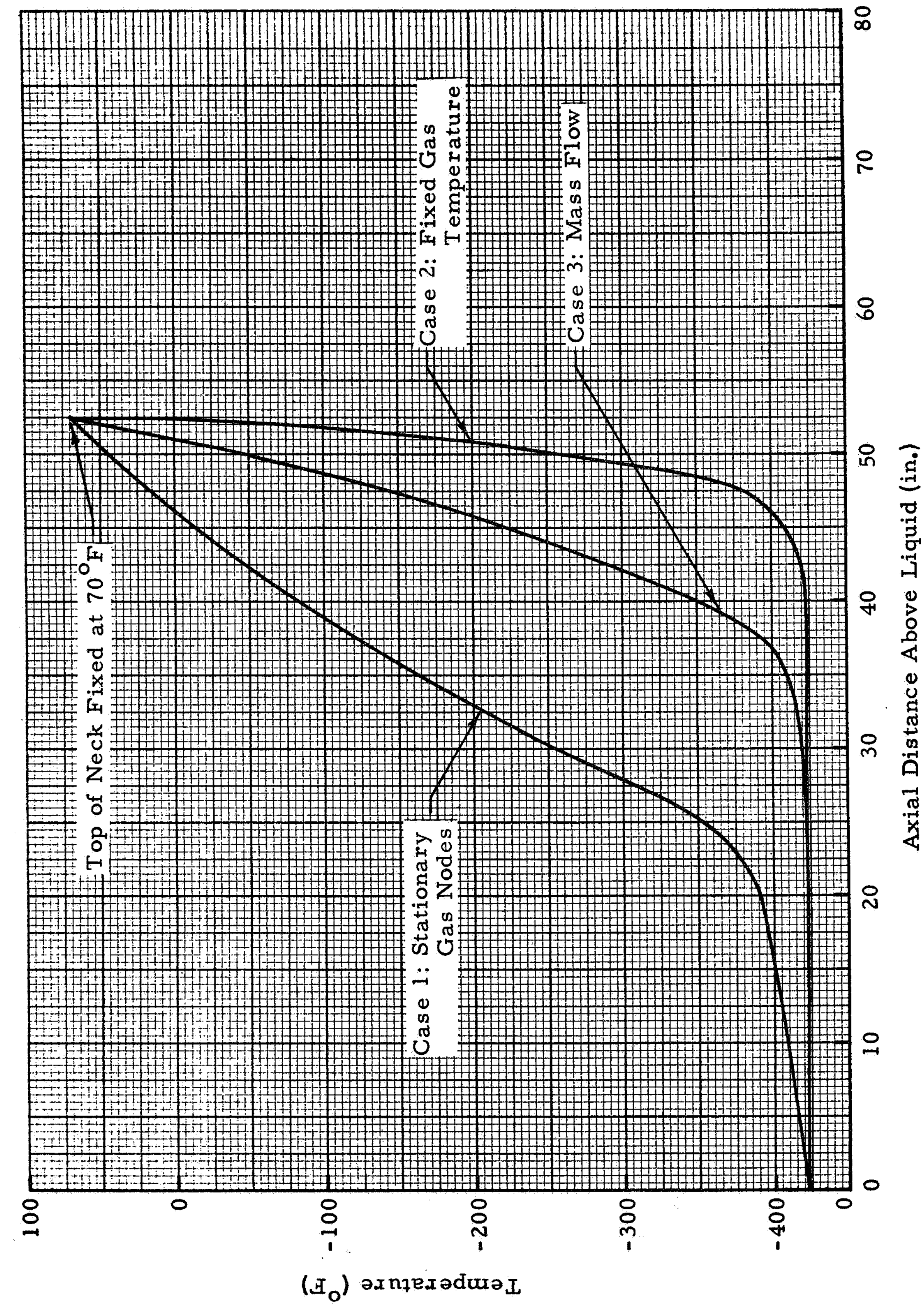

Fig. 5 - Plot of Tank Wall and Vent Temperature Distribution for MSFC 105-Inch Cryogenic Tank 95% Filled with LH_2 and Insulated with 1-Inch Mylar and Foam Insulation

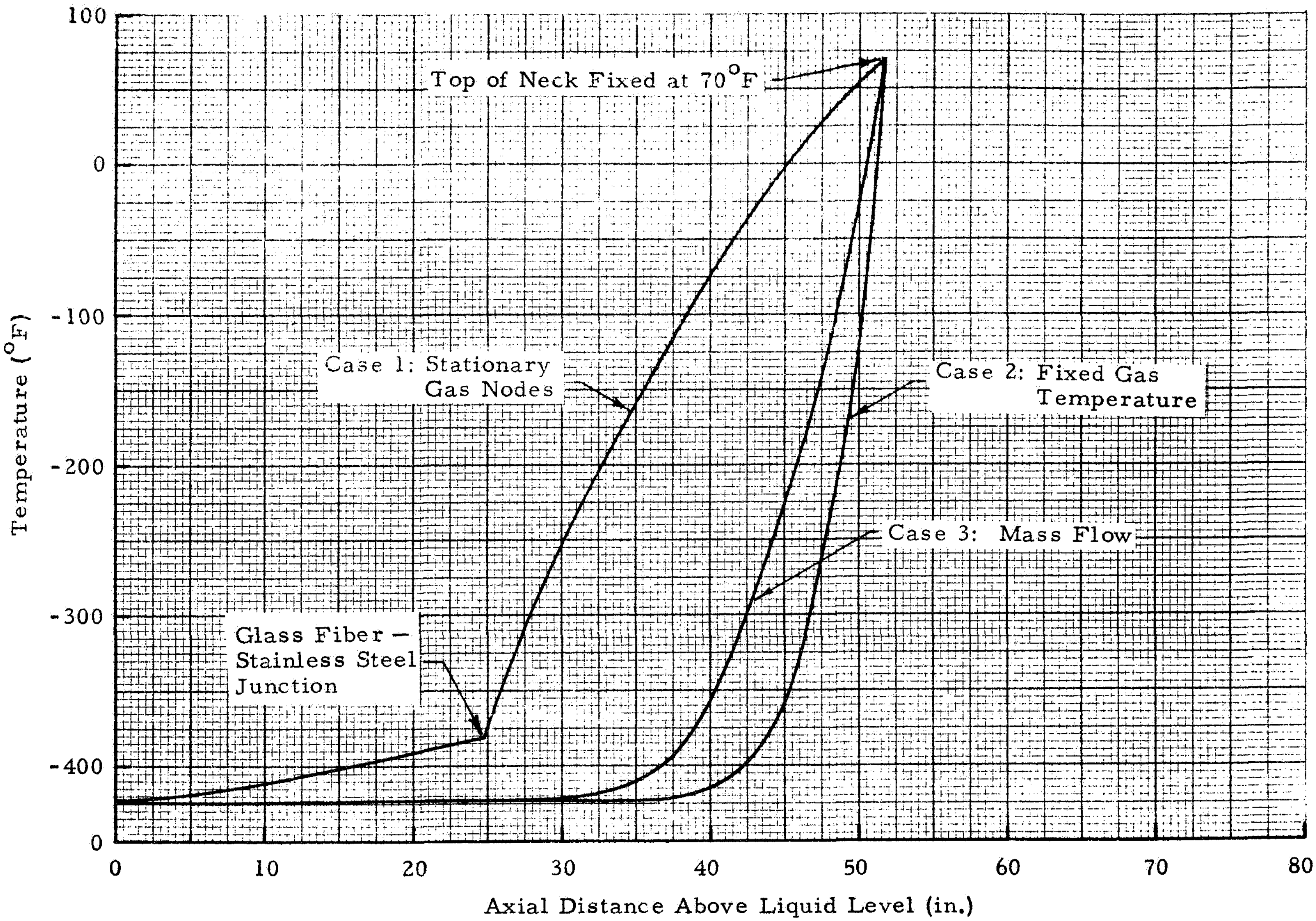

Fig. 6 - Plot of Fill Line Temperature Distribution for MSFC 105-Inch Cryogenic Tank 95% Filled With LH_2 and Insulated with 1-Inch Mylar and Foam Insulation

Section 3
CONCLUSIONS AND RECOMMENDATIONS

3.1 CONCLUSIONS

Thermal analysis of venting insulated cryogenic tanks requires a proper treatment of the convective and mass flow effects of the venting gas. The natural convection from the walls of the fill and vent lines, the tank wall and the instrumentation wires to the cooler gas results in a heat rate that is several times greater than that which would exist with pure conduction between the components and the gas. Because this natural convection exists in a gravity field, it is essential that it be included in the calculations.

The heat transport capability of a flowing gas with a temperature gradient is considerable. This effect must be included in any detailed thermal analysis. The actual heat rate associated with mass flow is

$$q = \dot{m}\, C_p\, T.$$

If the temperature of the gas flowing into a nodal position is different from that flowing out of the nodal position, then there exists a net heat flow due to the gas motion coupled with its thermal capacitance. There is a temperature gradient in cryogenic vent lines; therefore, there is a net heat flow out of the gas node defined by

$$q_{net} = \dot{m}\, C_p\, (T_{out} - T_{in})$$

This net heat flow is ignored in a standard thermal analysis using the stationary node approximation where the only heat flow in the direction of the gas motion is pure conduction in the gas. In the mass flow treatment case the vertical heat transport in the gas due to boiloff gas flow out the tank vent is approximately one order of magnitude greater than the vertical heat flow due to pure conduction alone in the gas. As seen in the comparison between cases, the effect of this heat transport on the final temperature distribution and the final heat leakage to the liquid is significant.

It is interesting to note a significant difference between the two tanks. For the 20-in. tank (Fig. 2),"the mass flow" (Case 3) temperature distribution is nearer the "stationary gas node" (Case 1) temperature distribution, while for the 105-in. tank (Figs. 5 and 6),"the mass flow" (Case 3) temperature distribution is nearer the "fixed gas temperature" (Case 2) temperature distribution. The reason for this is readily apparent. From a gas temperature gradient standpoint, Case 1 corresponds to a zero boiloff rate while Case 2 corresponds to an infinite boiloff rate. Therefore, the greater the boiloff, the closer Case 3 will come to Case 2 and the further from Case 1. The larger tank, obviously, has the larger boiloff with the result that Case 3 came closer to Case 2.

3.2 RECOMMENDATIONS

The Lockheed/Huntsville Thermal Analyzer Computer program is now specifically applicable to cryogenic tank analysis. Laminar and turbulent, natural and forced convection equations can be used to produce a heat transfer coefficient, h, as a function of distance from the leading edge, wall temperature, gravitational acceleration and gas type.

Mass flow equations have been incorporated into the program and are properly treating the thermal effect of the moving gas. However, its use currently is predicated on a foreknowledge of the mass flow rate. When this parameter is not known from test results, an estimate must made by hand calculations. It is recommended that the computer program be further modified to incorporate an iterative procedure to arrive at the analytically accurate value for mass flow rate. Heat rates to the stored liquid would be first calculated based on the value of mass flow rate input. A new value could then be calculated based on this total heat leakage and the known latent heat of vaporization of the cryogen. This iterative procedure will continue until the converged value of mass flow rate is reached.

The Lockheed/Huntsville Thermal Analyzer program has other options such as the transient capability. The current program capabilities coupled with those recently incorporated or proposed produces a meaningful computer tool for application to the storage of cryogenic fluids insulated vented or non-vented storage vessels in any g field and under any external thermal conditions. It is recommended that this computer program be utilized in future thermal analysis work dealing with vented cryogenic tanks.

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