FACILITY FORM 602

N65-24302	
(ACCESSION NUMBER)	(THRU)
64	1
(PAGES)	(CODE)
CR-62859	32
(NASA CR OR TMX OR AD NUMBER)	(CATEGORY)

THERMAL PROTECTION SYSTEMS FOR LIQUID HYDROGEN TANKS

BY PETER E. GLASER

GPO PRICE $ ______

OTS PRICE(S) $ ______

Hard copy (HC) $3.00

Microfiche (MF) .75

LIQUID PROPELLANT LOSSES DURING SPACE FLIGHT

PREPARED FOR
THE NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
CONTRACT NO. NAS5-664

THERMAL PROTECTION SYSTEMS FOR

LIQUID HYDROGEN TANKS

By

PETER E. GLASER

November 1962

C-65008-3

Arthur D. Little, Inc.

TABLE OF CONTENTS

TABLE OF CONTENTS (Continued)

I. INTRODUCTION

The brilliantly glowing nose cone successfully resisting the effects of aerodynamic heating while re-entering the earth's atmosphere is proof that thermal protection systems can be developed to meet conditions which only a few years ago were beyond technological feasibility. Less spectacular, but of equal importance to the continued progress of space exploration, is the development of thermal protection systems for cryogenic fuels, in particular liquid hydrogen, to meet the demands of a wide spectrum of space missions. The difficulties that impede thermal protection development and application have only recently been fully appreciated after it has been demonstrated that thermal insulations and application techniques cannot be relegated to the category of engineering design details.

In fact, thermal protection systems adequate to meet the demands of fuel conservation for space missions which may extend beyond a few days duration are not within the present state of the art. The development efforts required before liquid hydrogen tanks can be adequately insulated to permit space vehicles to explore the regions beyond the vicinity of the earth appear to be at least equivalent to efforts already expended on the development of nose-cone materials.

The following discussion deals with the requirements of thermal protection systems subjected to ground and space environments; typical materials either presently available or under development; the influence of the physical variables governing the thermal properties of those materials and the apparatus required for their measurement; possible improvements in thermal protection systems; and design considerations for thermal protection systems, dictated by production, handling, and test requirements.

II. ENVIRONMENTS AFFECTING THERMAL PROTECTION SYSTEMS

From the time a thermal protection system is assembled in the production plant to the successful completion of its mission, it will have been exposed to a number of environmental interactions which need to be recognized and considered in the various design approaches and performance expectations.

A. THERMAL INTERACTION

A thermal protection system has to be designed to reduce the boil-off rate caused by heat sources external to the vehicle and by heat sources on-board. Among the external heat sources are the thermal conditions existing on the ground prior to flight, the aerodynamic heating encountered during boost, and the various radiation heat inputs, such as direct solar radiation and planet shine. On-board heat sources consist of radiative inputs from other portions of the vehicle and heat leaks to the propellant tank from supports, pipes and other structural members.

B. IONIZING RADIATION INTERACTIONS

The ionizing radiation existing in a space environment may affect the performance of thermal protection system materials, particularly those of organic origin.

C. METEOROID INTERACTIONS

Destructive effects due to meteoroids of high kinetic energy can markedly reduce the reliability of thermal protection systems unless suitable bumpers to mitigate these effects are employed.

D. MECHANICAL INTERACTIONS

The handling and transportation of the insulated tanks prior to firing may place severe restrictions on the design of thermal protection systems so as not to decrease reliability and not to increase the complexity of maintenance procedures. It is in this area that increased communication between designers and producers is required so that designers may take into account human errors which could result in damage to the thermal protection system.

III. SELECTION OF MATERIALS FOR THERMAL PROTECTION SYSTEMS

For liquid-hydrogen-fueled vehicles, a thermal protection system is absolutely essential. However, considerable choice can be exercised in the combination and selection of materials for a system based on meeting the conditions imposed by the above environment. The thermal performance, thickness, and weight of insulation materials have to be optimized to accomplish a specific mission so as to minimize weight penalties.

For example, weight penalties can be associated with different methods for preserving a cryogenic fuel depending on whether vented storage or non-vented storage, with or without a refrigerator, is being used.[1] Figure 1 illustrates the effect of these different methods on the thickness of high efficiency insulation required for a given stay-time in space for liquid hydrogen tanks of different diameters. Although refrigeration can be seen to greatly reduce the thickness requirement, thereby leading to a several-fold reduction in weight penalties for extended stay times, a finite insulation thickness is still essential.

Figure 2 compares the thermal conductivity of insulating materials applicable to thermal protection systems. Although there is nearly a three-order-of-magnitude difference in the insulating effectiveness of these various materials, their strength properties, temperature capabilities, and application feasibility also have to be considered in applying these materials to a thermal protection system.

The following review presents the salient characteristics of thermal insulators which are of particular interest in a thermal protection system for liquid hydrogen tanks.

A. FOAMS

Great strides have been made in the development of various types of foamed organic plastics using either fluorinated hydrocarbons or carbon dioxide as expanding agents. The thermal conductivity of foams as a function of temperature and the effect of foam structure on thermal conductivity are useful criteria of the insulating effectiveness of foams[2]. Figure 3 shows the dependence of thermal conductivity on temperature for a Freon expanded foam of two lb/cu. ft. density. The non-linearity in this dependence appears to be due to increased heat transfer by convection currents resulting from re-boiling of condensed Freon and the saturation of the foam-cell surfaces with liquid Freon resulting in increased heat transfer by conduction. Furthermore, the addition of heat to the warmer side of the foam cell may cause a mass transfer of

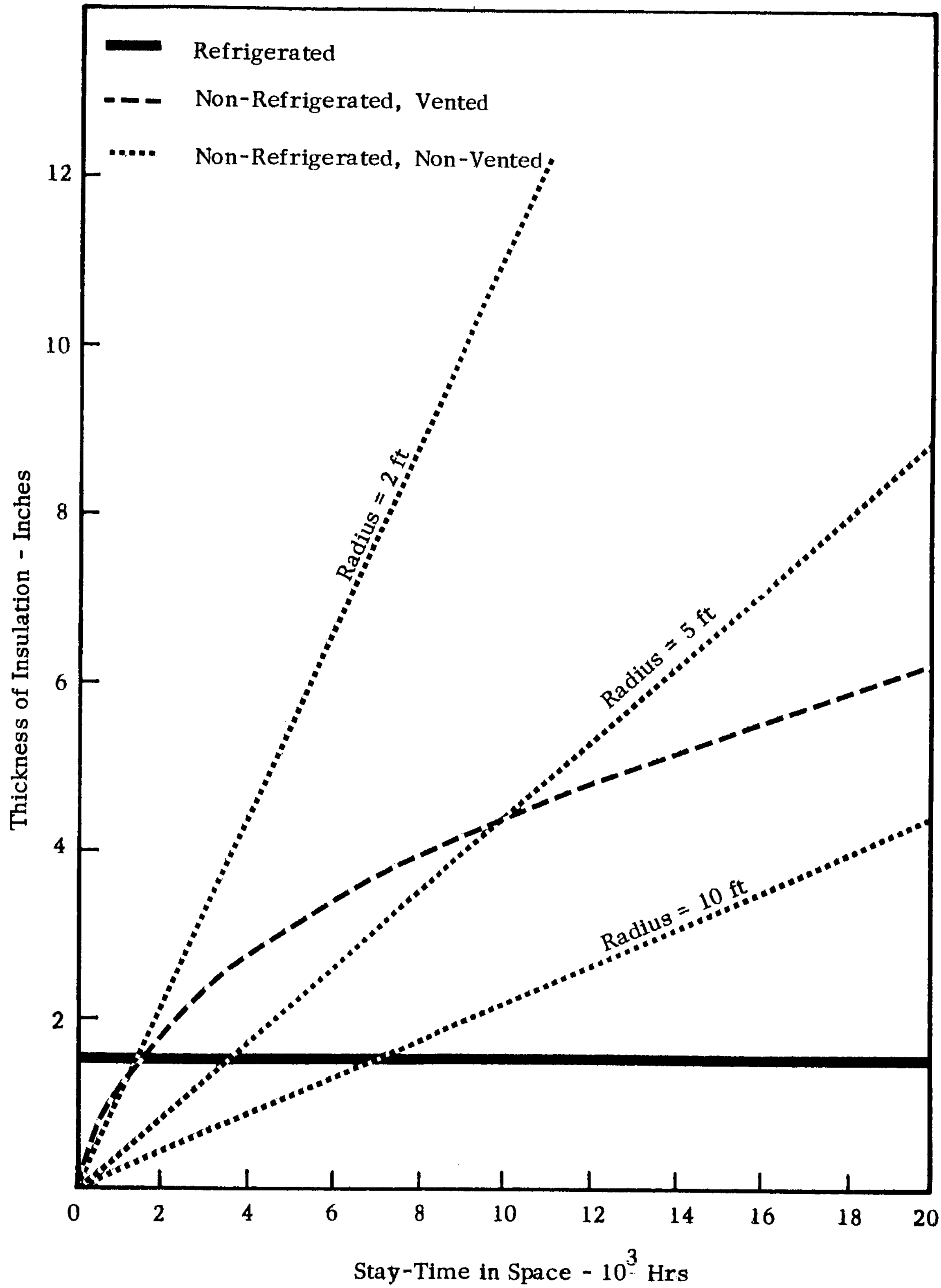

FIGURE 1 COMPARISON OF INSULATION REQUIREMENTS FOR ALTERNATE METHODS OF STORING LIQUID HYDROGEN

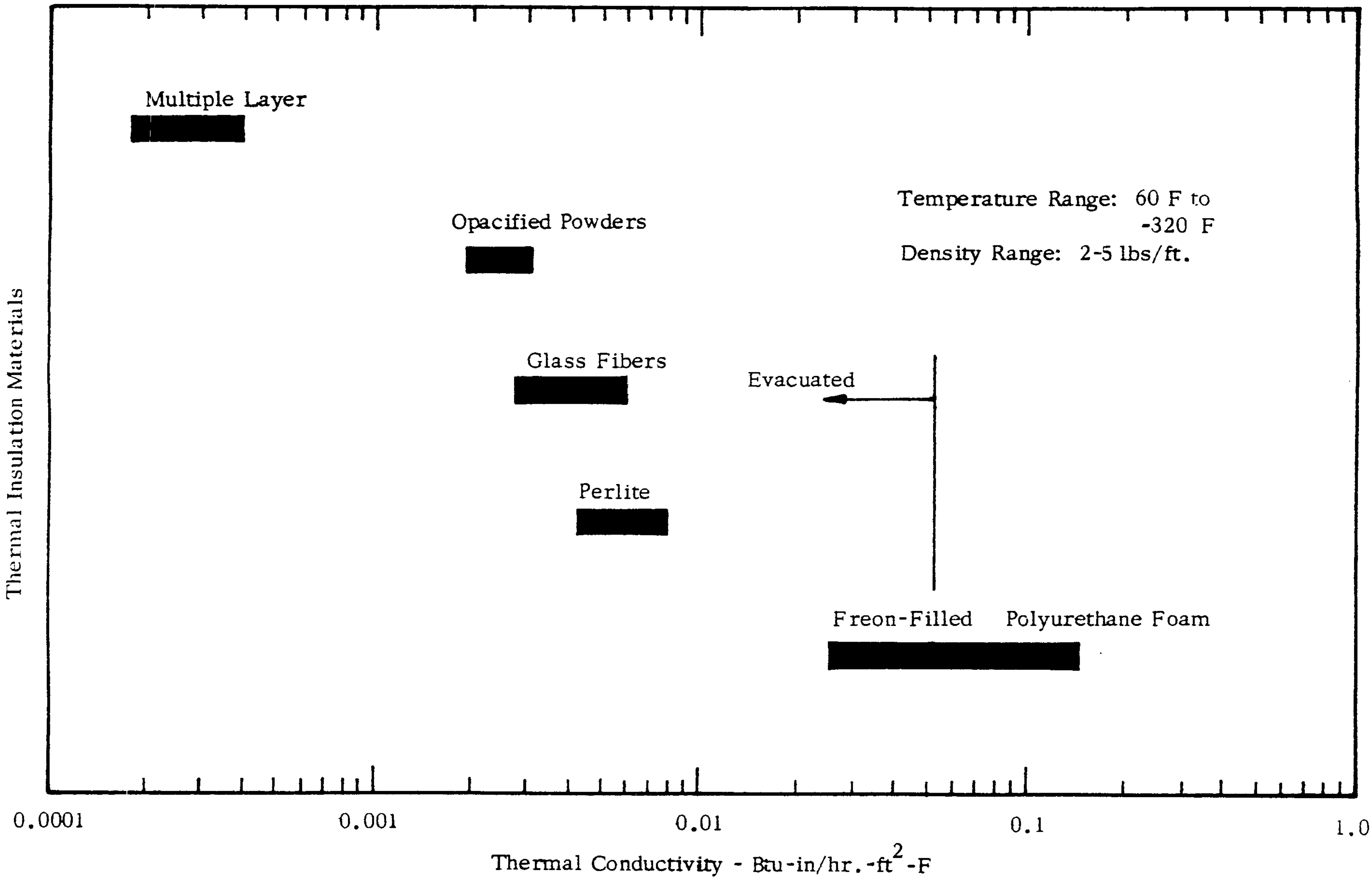

FIGURE 2 THERMAL CONDUCTIVITY OF TYPICAL THERMAL INSULATORS

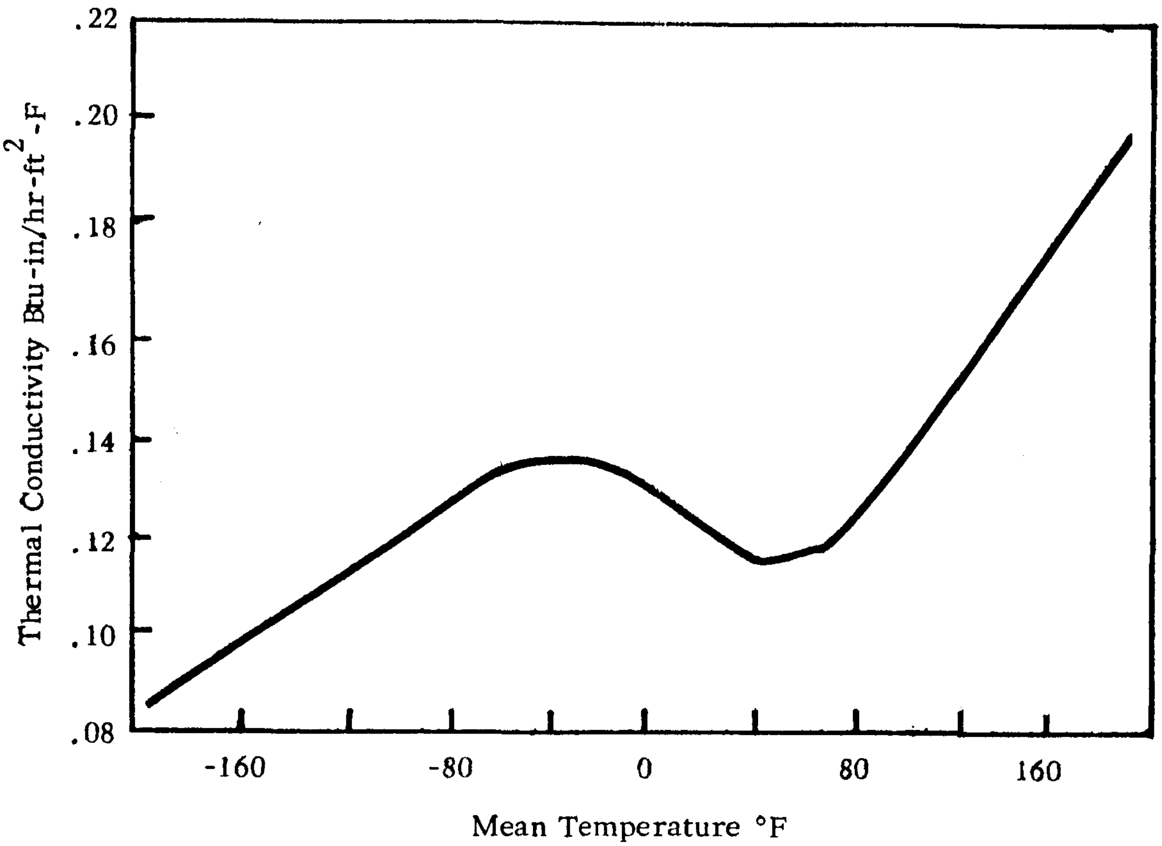

FIGURE 3 THE DEPENDENCE OF THERMAL CONDUCTIVITY ON TEMPERATURES OF FREON EXPANDED POLYURETHANE FOAM

Source: R. E. Knox, "Insulation Properties of Fluorocarbon Expanded Rigid Urethane Foam" saturated Freon vapor to the cold side with subsequent release of latent heat. This behavior will be of particular interest when the foam is to be used in combination with other thermal insulators.

Figure 4 illustrates the effect of foam cell size on thermal conductivity and indicates that the smaller the cell diameter the greater the resistance of heat flow because of the increased heat flowpaths per unit thickness.

Foam insulation becomes a partially evacuated insulation when it is exposed to liquid hydrogen, because of cryopumping effects. Thus precautions have to be taken to assure that gas-tight barriers are provided to stop the penetration of moisture and air into the insulation, which would result in a marked increase in thermal conductivity and the possible disruption of the cells if surface temperature suddenly increased.

B. POWDERS AND FIBERS

Powder and fiber insulations have a low thermal conductivity when gases have been removed from the void spaces. As far back as 1910, it was recognized that thermal insulators far more effective than air could be obtained when finely divided materials were subjected to a vacuum.[3] Although remarkable improvements are obtained by the removal of gas, other heat transfer mechanisms must be minimized. Heat flow through an evacuated insulating material can be attributed to the simultaneous operation of several different mechanisms:

(1) Solid conduction through the materials making up the insulation and conduction between the particles across areas of contact;

(2) Residual gas conduction in spaces between the particles;

(3) Radiation across the spaces between and through the particles.

Solid conduction can be reduced by breaking up the conduction paths through the use of finely divided materials so that resistances to heat flow are formed at the surface of each particle or fiber. To maintain these devious heat flow paths, the contact areas between the individual particles must be reduced to point contacts. These resistances are a function of the load imposed on the material and depend upon the amount of deformation at such point contacts. The particle size of a specific powder or the diameter of the fiber will influence the area of contact as well as the resistance per unit thickness of insulation. It is also desirable that the void spaces between particles be small enough so that gas conduction can be greatly reduced at reasonably low pressures. Therefore, powders with a range of particle sizes are desirable so

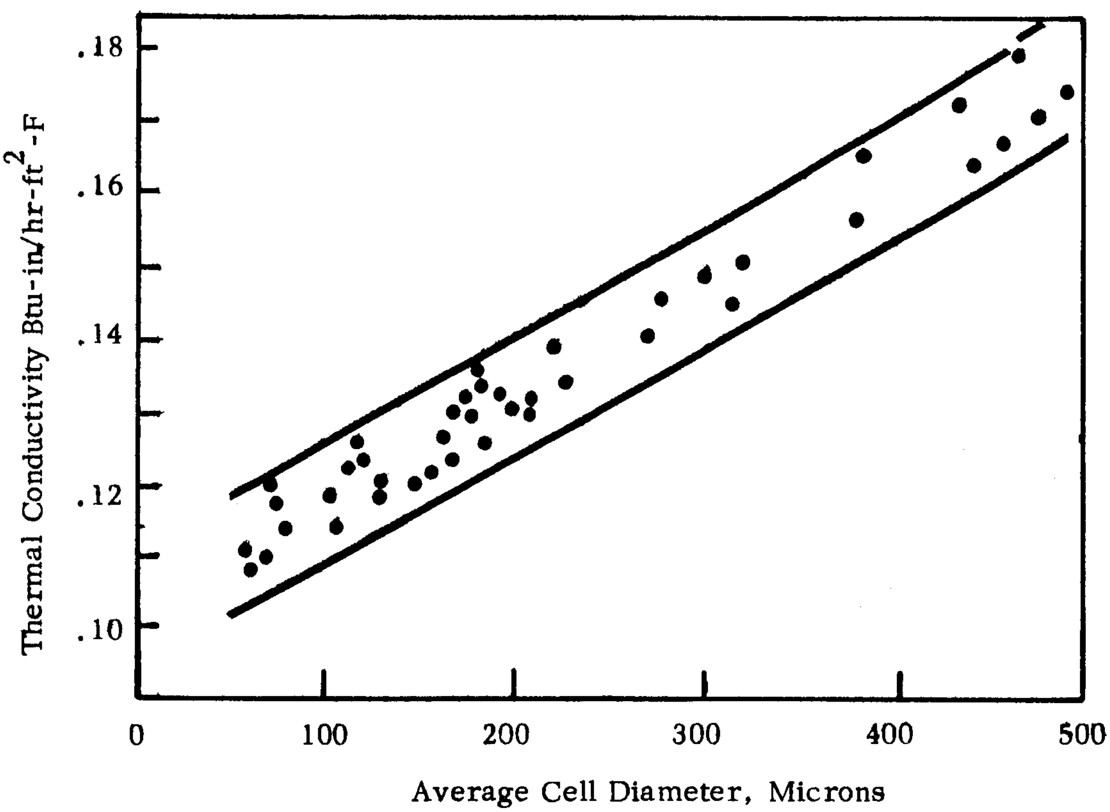

FIGURE 4 THE EFFECT OF CELL SIZE OF FREON EXPANDED POLYURETHANE FOAM ON THERMAL CONDUCTIVITY

Source: R.E. Knox, "Insulation Properties of Fluorocarbon Expanded Rigid Urethane Foam" that large voids are filled by smaller particles and small voids, in turn, still smaller particles. Submicron fibers with parallel orientation tend to exhibit an increased insulating effectiveness.

Evacuated insulating materials require that most of the gas be removed to obtain the desired insulating effectiveness; the degree of vacuum necessary to achieve this is of great practical importance. The mechanism by which heat flows by means of gas conduction between the particles can be explained on the basis of the kinetic theory of gases. The effect of gas conduction can be divided into two separate regions:

(1) The region ranging from atmospheric pressure down to a few millimeters of mercury in which gas conduction is independent of pressure.

(2) A region at pressures below a few millimeters of mercury in which gas conduction depends on pressure.

The transition from one type of conduction to the other depends upon the mean diameter of the powder or fiber, the arrangement of individual particles, and the pressure required to obtain a specific mean-free path. When the mean-free path of the gas molecules is greater than the distance between the individual particles, then the pressure-dependent region is obtained. The thermal conductivity of the gas at that point decreases substantially until a lower limit is obtained at a pressure when most of the gas molecules have been removed. The larger the particle spacing, the lower the pressure required for the thermal conductivity to approach its limiting value. At this lower limit, the thermal conductivity still has a finite value because heat can be transferred by radiation, by solid conduction within and through the particles, and by residual gas conduction. The effect of residual helium or nitrogen gas on thermal conductivity of perlite is shown in Figure 5.[4]

Fine powders and sub-micron diameter fibers exhibit a very low solid conductivity, but they are relatively transparent to radiation. The radiation heat-transfer component in powders can be reduced by the addition of absorbing and scattering media, such as metallic flakes or carbon powder.[5] Figure 6 shows the effect of the addition of aluminum flakes on thermal conductivity.[6] The addition of larger amounts of opacifiers tends to increase solid conduction so that there is a limit to the improvement that can be obtained by this technique.

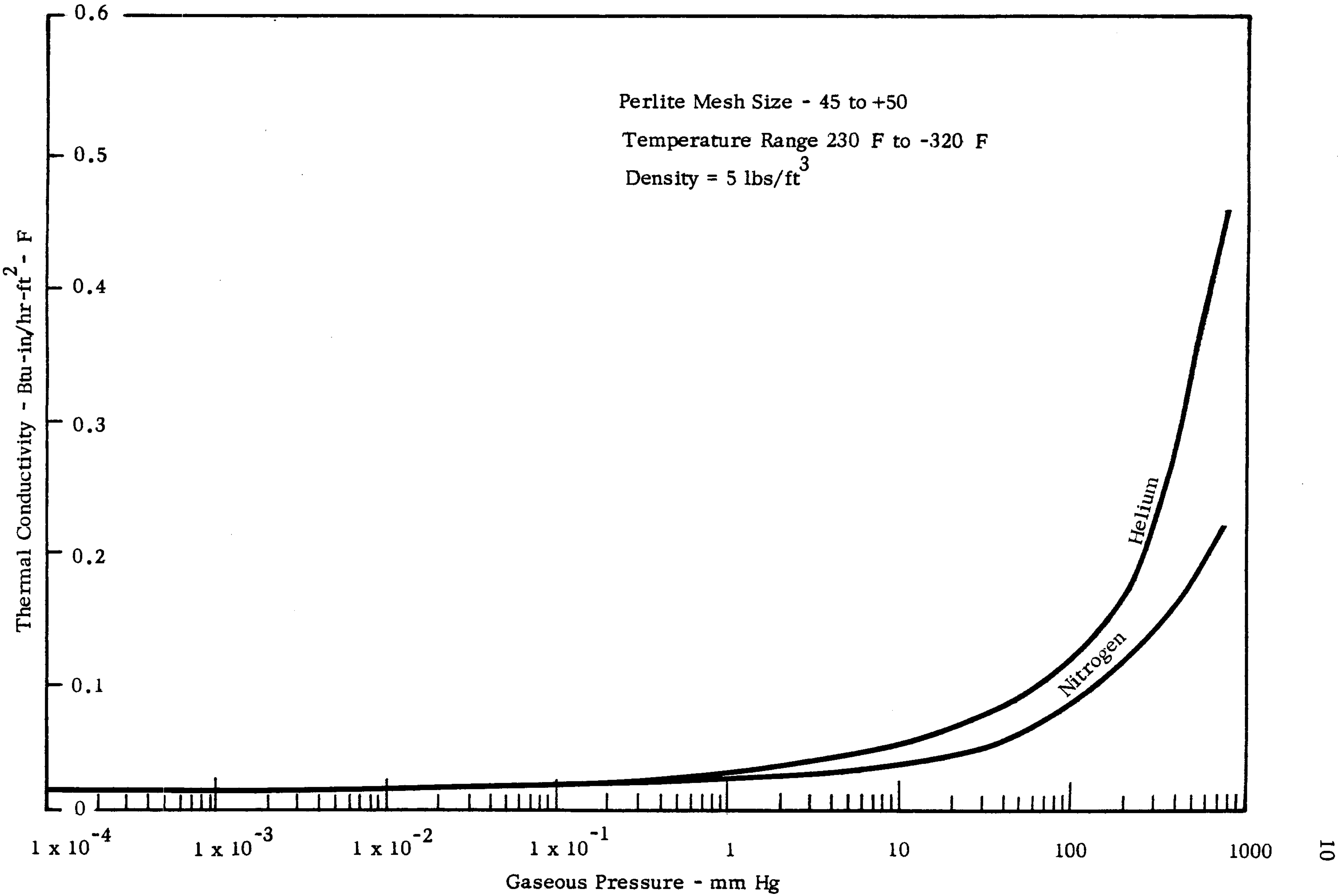

FIGURE 5 THE EFFECT OF THE TYPE OF RESIDUAL GAS ON THE THERMAL CONDUCTIVITY OF PERLITE POWDER

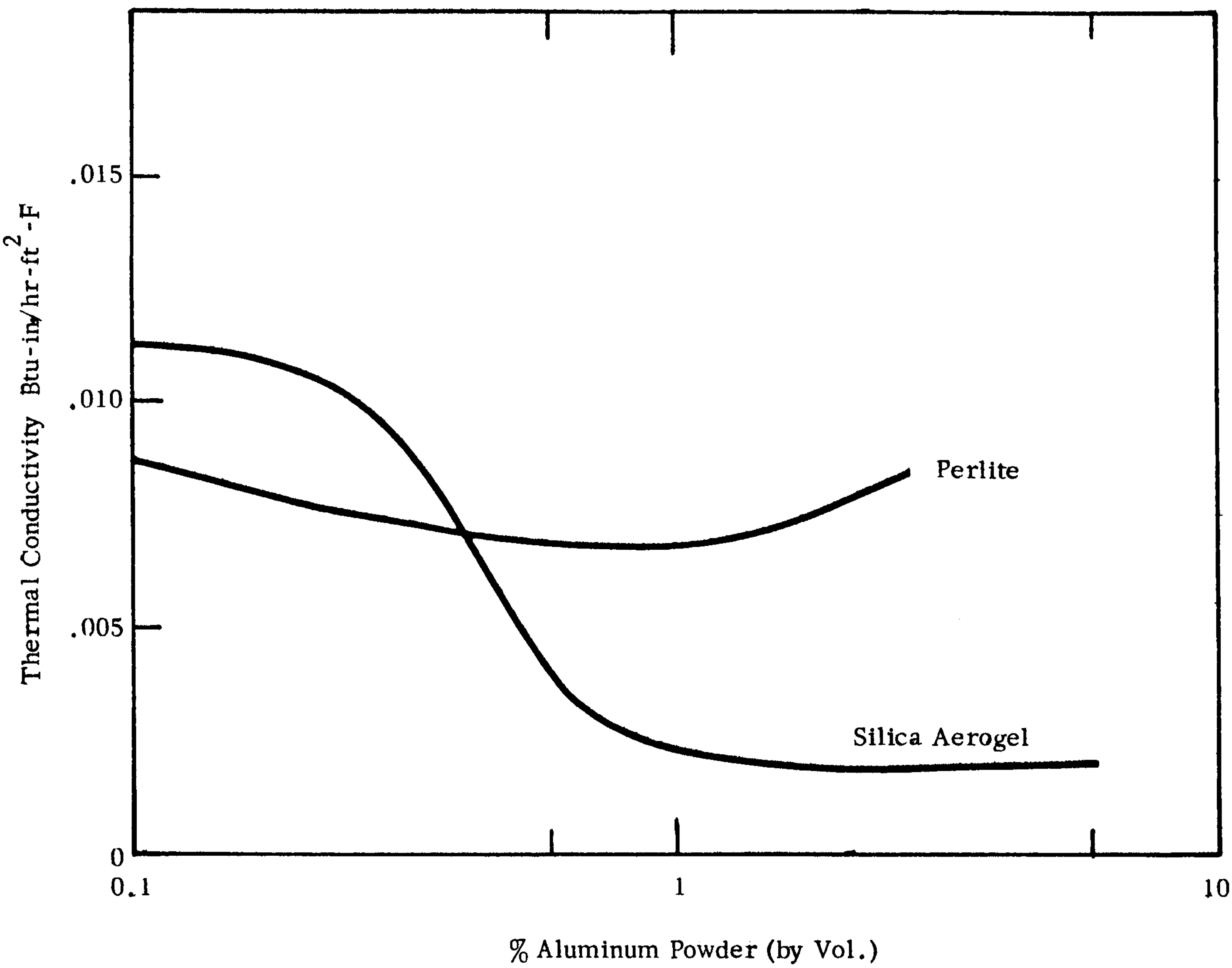

FIGURE 6 THE EFFECT OF ADDITION OF ALUMINUM POWDER ON THERMAL CONDUCTIVITY OF PERLITE AND SANTOCEL

Source: D. Cline and R.H. Kropschot, "Thermal Properties of Powder Insulations in the Temperature Range 300° to 4°K"

C. MULTILAYER INSULATION

The work on multilayer insulations carried out by Peterson[7] and the further improvements carried out by researchers in this country[8][9][10] have resulted in an insulating effectiveness substantially better than that achieved by evacuated powders and fibers. In this type of insulation the principles already used in the improvement of evacuated insulation were carried to their logical conclusion by the recognition of the importance of radiation heat transfer once the mechanisms of gas conduction and solid conduction have been reduced. Ideally in this insulation radiation shields of the highest reflectivity are separated by nearly non-conducting spacers in a high vacuum. The term superinsulation has been applied to this insulation system to signify the importance of this technological advance. Table I lists typical multilayer insulations now under consideration.

TABLE I

TYPICAL RADIATION SHIELD AND SPACER MATERIAL COMBINATIONS FOR MULTILAYER INSULATIONS

RADIATION SHIELD	SPACER
0.00025 inch crinkled Mylar, aluminized one side. One to two ohms per cm^2. National Research Corporation	
0.0005 inch Aluminum Foil Alcoa	3 gm/ft^2 fiberglass Libby Owens Ford
0.0005 inch Aluminum Foil Alcoa	0.004 inch Dexter Paper
0.00025 inch crinkled aluminized Mylar National Research Corporation	0.004 inch Dexter Paper 50% perforations
0.002 inch Aluminum Foil Alcoa	0.022 inch thick resin covered fiberglass netting with 1/8" x 1/8" mesh

The successful application of multilayer insulations to the solution of cryogenic storage and transportation problems of large cryogenic tanks on earth points to the promising solution of the problem of thermal protection of liquid hydrogen during long periods of space travel. However, before multilayer insulations can be incorporated in a thermal protection system, a number of basic problems will have to be solved. An indication that this task is indeed of greater magnitude than had been at first realized when these high efficiency insulations became generally available is the fact that, at present, no space vehicle of major size is using such a system.

IV. THERMAL CONDUCTIVITY MEASUREMENTS

As part of the work which has been undertaken by Arthur D. Little, Inc., to investigate liquid propellant losses in space[11], a number of the variables which influence the performance of multi-layer insulation under projected use conditions are being investigated.

The variables influencing insulation performance which are of particular interest are the following: (1) temperature, (2) gas pressure, (3) outgassing, (4) mechanical pressure, (5) emittance, and (6) the effects of thermal shorts and discontinuities.

To permit experimental measurement of the influence of these variables, a suitable thermal conductivity apparatus had to be constructed. Several basic designs for thermal conductivity apparatus have been used for investigations of specific variables.[12] None of them is standard and capable of meeting all the required test conditions for the measurement of the above variables. The designs presently in use can be classified according to the shape of the sample test chamber as spherical, cylindrical, or flat plate.

A. SPHERICAL APPARATUS

Figure 7 shows a spherical thermal conductivity apparatus consisting of an inner sphere filled with a cryogenic liquid and an outer sphere kept at a desired temperature by means of an electrical heater or a constant temperature bath. The test sample is placed between the two spheres. The boil-off rate of the cryogenic liquid is an indication of the heat flow through the material. A variation of this type of apparatus has an electrical heater installed inside the inner sphere while the outer sphere is placed into a cryogenic liquid. The heat input to the heater is used in the calculation of thermal conductivity.

This type of apparatus is simple to construct and it eliminates edge effects. However, it is difficult to produce a spherically shaped sample, except for loose-fill materials; the density of the specimen cannot be easily controlled; only one sample size can be tested; it is difficult to apply mechanical pressure to the sample; and the heat leak along the neck of the inner sphere cannot be easily estimated.

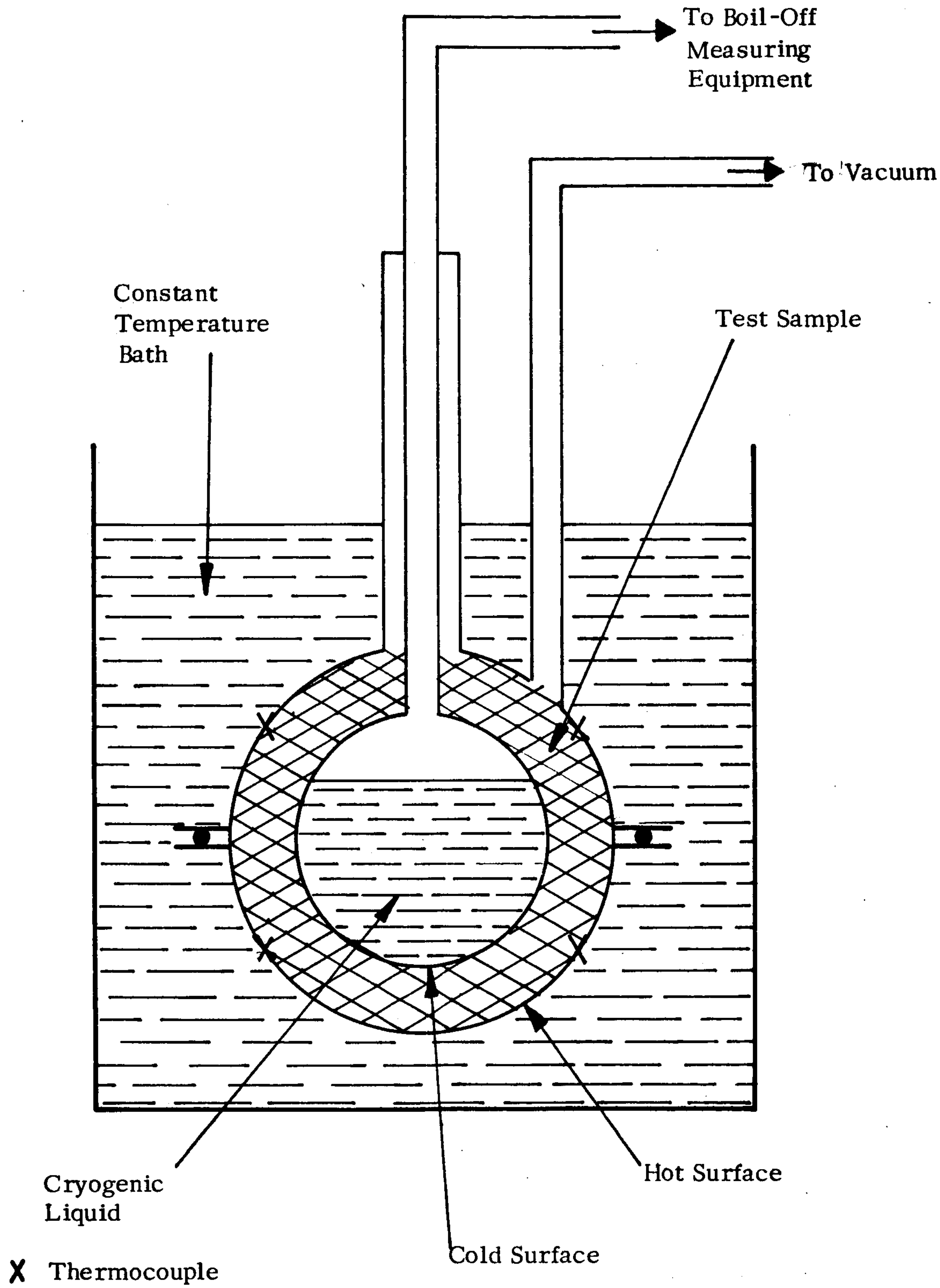

FIGURE 7 DIAGRAM OF THE SPHERICAL THERMAL CONDUCTIVITY APPARATUS

B. CYLINDRICAL APPARATUS

Figure 8 shows a cylindrical thermal conductivity apparatus[8]. It consists of a measuring vessel provided with end guard vessels. The upper guard vessel reduces the heat loss through the neck of the measuring vessel. All three vessels are filled with the same cryogenic liquid. The outer jacket can be kept at a uniform temperature by an electrical heater or a constant temperature bath. The boil-off rate of the cryogenic liquid from the measuring vessel is an indication of the heat flow through the sample.

This apparatus has been used for the testing of powders, foams, fibers and multi-layer insulations[13]; it cannot be easily adapted to measure the influence of thermal conductivity changes caused by mechanical pressure on the sample; and the thickness of the sample cannot be accurately controlled.

C. FLAT PLATE APPARATUS

The flat plate thermal conductivity apparatus consists of hot and cold plates between which a sample can be inserted. This type of apparatus approaches closest to the ASTM guarded hot plate apparatus[14] in concept and has been shown to be versatile and useful in a number of investigations. The following variations of the apparatus are in use:

1. Guarded Hot Plate

The guarded hot plate apparatus shown in Figure 9 consists of an electrically heated hot plate placed inside a ring shaped guard heater.[15] Two test specimens are placed on each side of the hot plate. The complete assembly is enclosed in a container which can be evacuated and is then immersed in a constant temperature bath. This apparatus permits the measurement of thermal conductivity over an easily adjustable range of temperature. However, the physical environment of the sample tends to be harder to control. This apparatus has been particularly useful in the measurement of the thermal conductivity of foams.

A variation of this type of apparatus is the single-guarded hot plate apparatus shown in Figure 10[16]. Only one sample needs to be installed between the hot plate and the single cold plate. The temperature of the guard ring has to be kept equal to that of the measuring hot plate to prevent heat transfer between guard and measuring plate, otherwise errors due to the departure from one dimensional heat flow may be introduced. This problem tends to be accentuated when materials of high insulating effectiveness are used, although it can be overcome by sensitive heater controls.

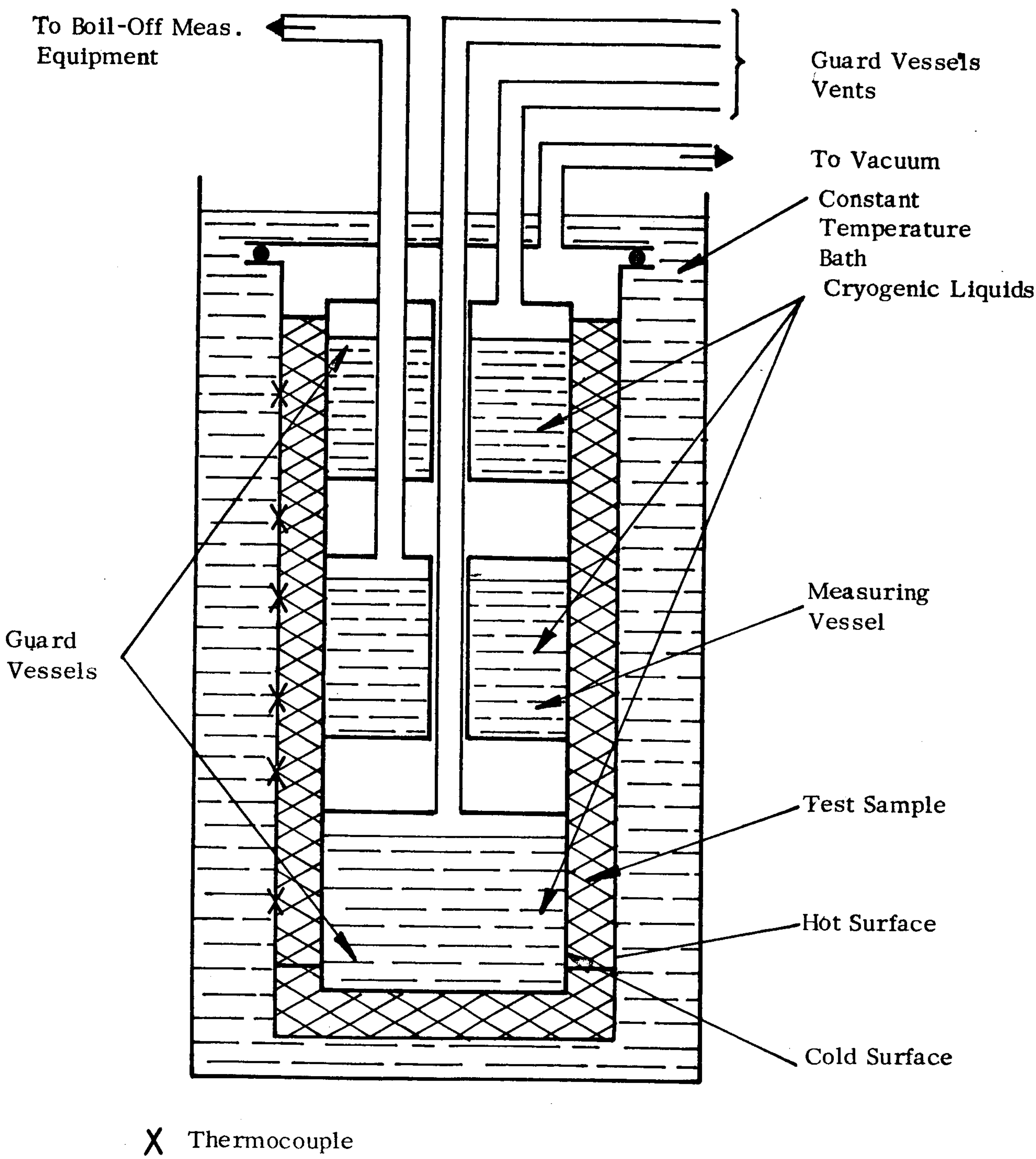

FIGURE 8 DIAGRAM OF THE CYLINDRICAL THERMAL CONDUCTIVITY APPARATUS

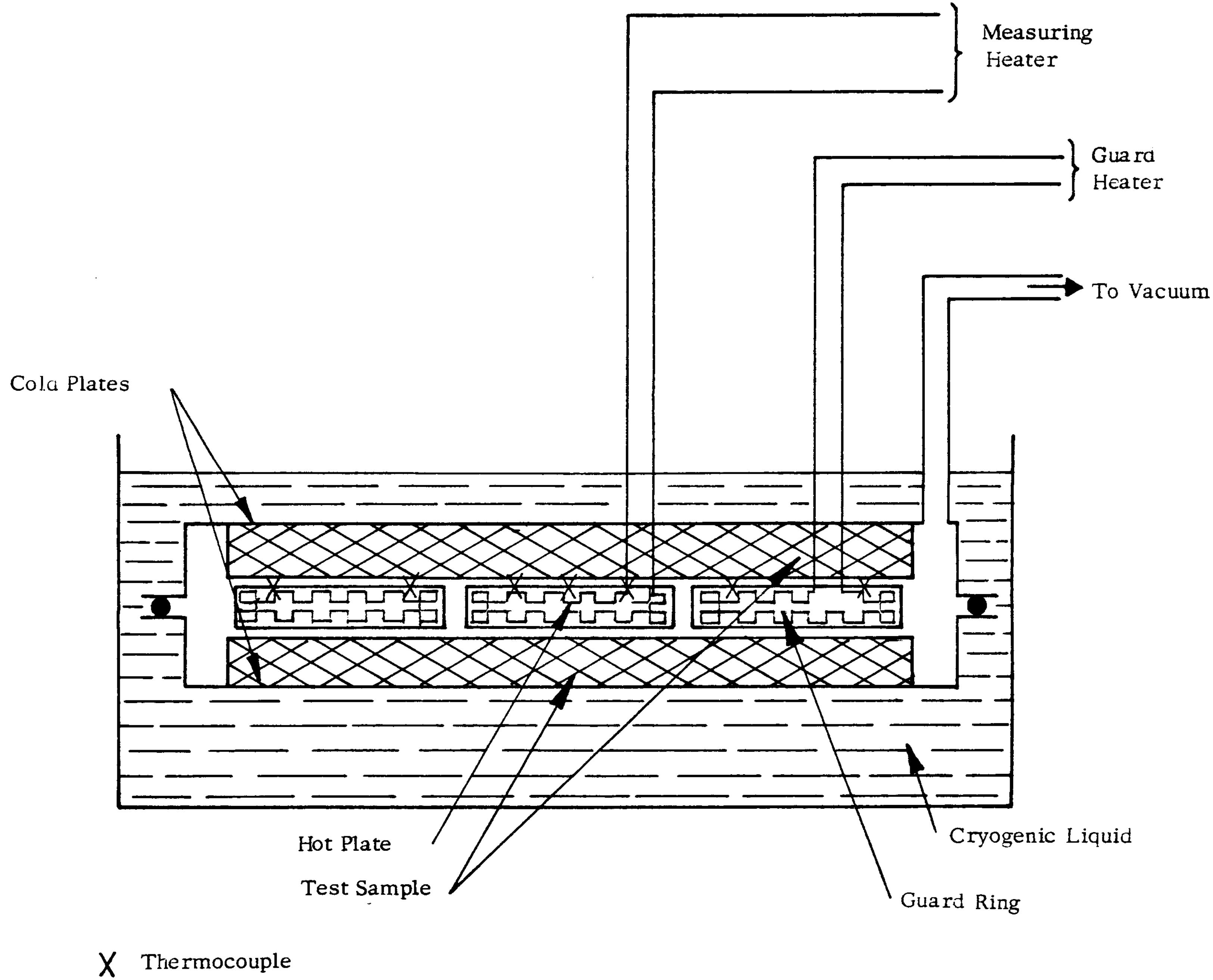

FIGURE 9 DIAGRAM OF THE GUARDED HOT PLATE THERMAL CONDUCTIVITY APPARATUS

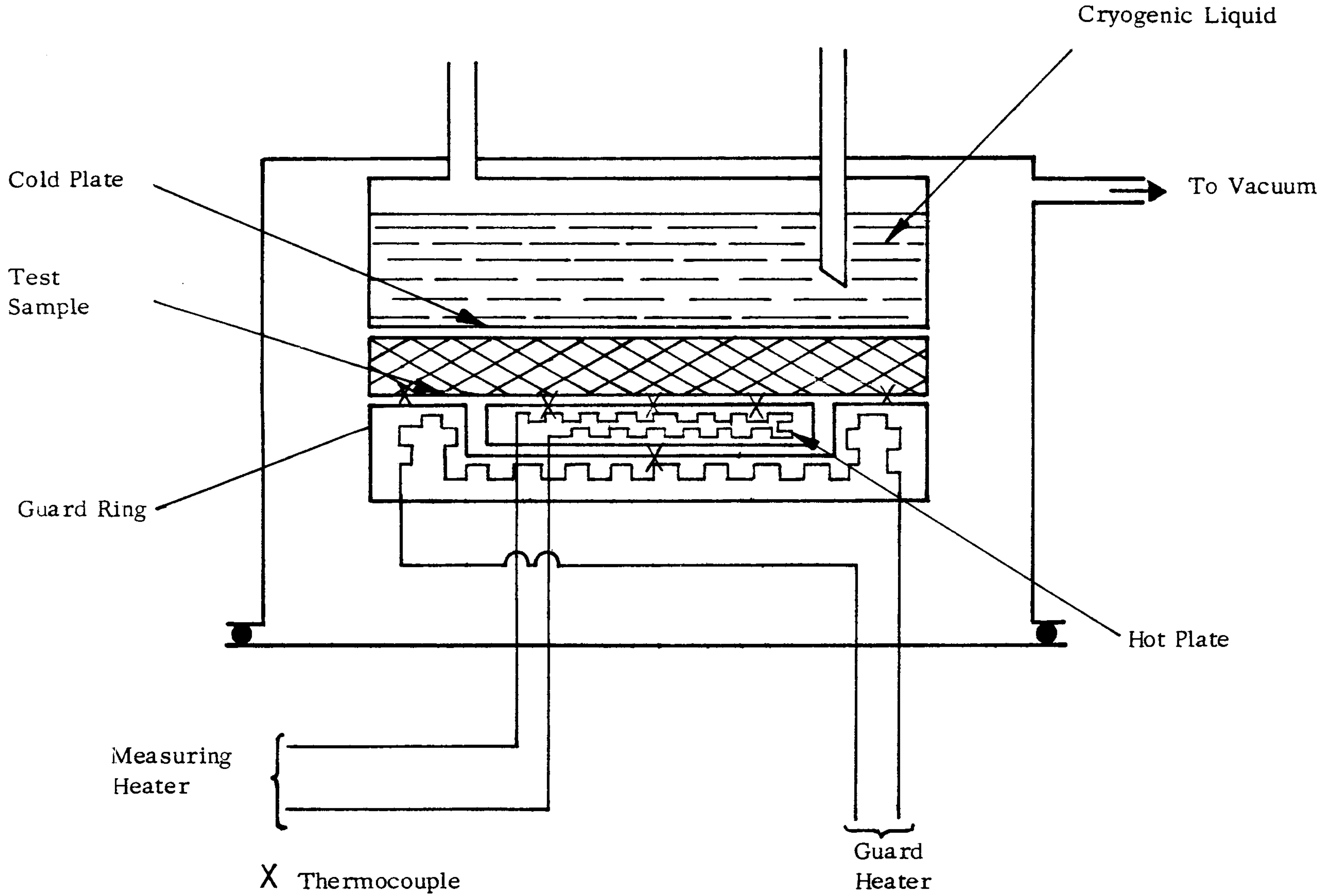

FIGURE 10 DIAGRAM OF THE SINGLE GUARDED HOT PLATE THERMAL CONDUCTIVITY APPARATUS

2. Guarded Cold Plate

Figure 11 shows a single-guarded cold plate apparatus[17]. In this apparatus the bottom of the measuring vessel constitutes the cold plate. The measuring vessel is surrounded by a guard vessel. The hot plate is kept at a desired uniform temperature by a circulating fluid. In this apparatus the boil-off rate of a cryogenic liquid from the measuring vessel is an indication of the heat flow through the sample.

The use of a calibrated heat flow meter is an alternative method for measuring the heat input into the sample[18]. However, calibration of the heat flow meter, particularly at very low heat flow rates, is difficult.

Because the measurement of the boil-off rate relies on calorimetric principles and the conditions of the cryogenic liquid can be closely controlled, we have given preference to this approach compared to the electrical heat input measurements in the design of an improved double-guarded cold-plate apparatus for measurement of thermal conductivity at liquid hydrogen temperatures.

Figure 12 shows the design details of the apparatus, which consists of the following major components: a measuring vessel enclosed in a ring-shaped guard vessel, an outer guard jacket which can be filled with liquid nitrogen while the other vessels are filled with liquid hydrogen, a warm plate on which the sample is placed, and a sample chamber which allows for control of the atmosphere and pressure surrounding the sample separate from that of the bell jar enclosing the rest of the apparatus. Figure 13 shows a photograph of the assembled apparatus.

The apparatus is designed to measure thermal conductivity of multilayer insulations, powders, fibers, and foams having thermal conductivities as low as 10^{-4} Btu-in./hr.-ft^2-F.

Figure 14 shows the instrumentation which is required to carry out sensitive heat flow measurements. Because the measurement of heat flow relies on the measurement of the boil-off rate from the cryogenic fluid in the measuring vessel, the volume of gases boiled-off can either be measured with a gas flow meter or at very low rates, by collecting gases above a low vapor pressure oil in a graduate.

The apparatus was designed so that one side of the sample can be exposed over a range of discrete temperatures from 4.2 to 243 K depending on the boiling point of the specific fluid used. The other side of the sample can be exposed to a range of temperatures from 20 to 500 K by proper choice of fluids and automatic temperature control. The test sample in the form of a disc can

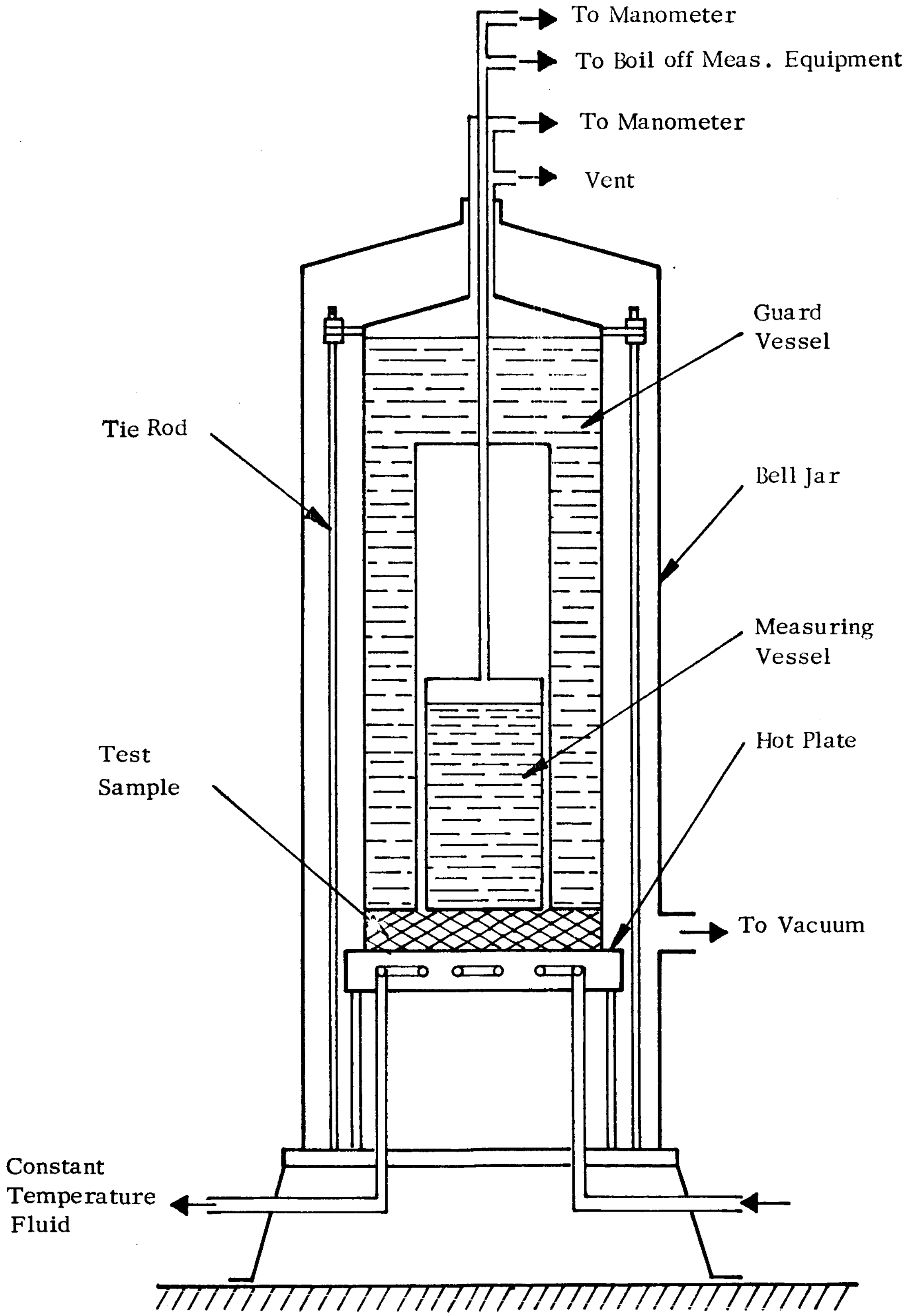

FIGURE 11 DIAGRAM OF THE SINGLE GUARDED COLD PLATE THERMAL CONDUCTIVITY APPARATUS

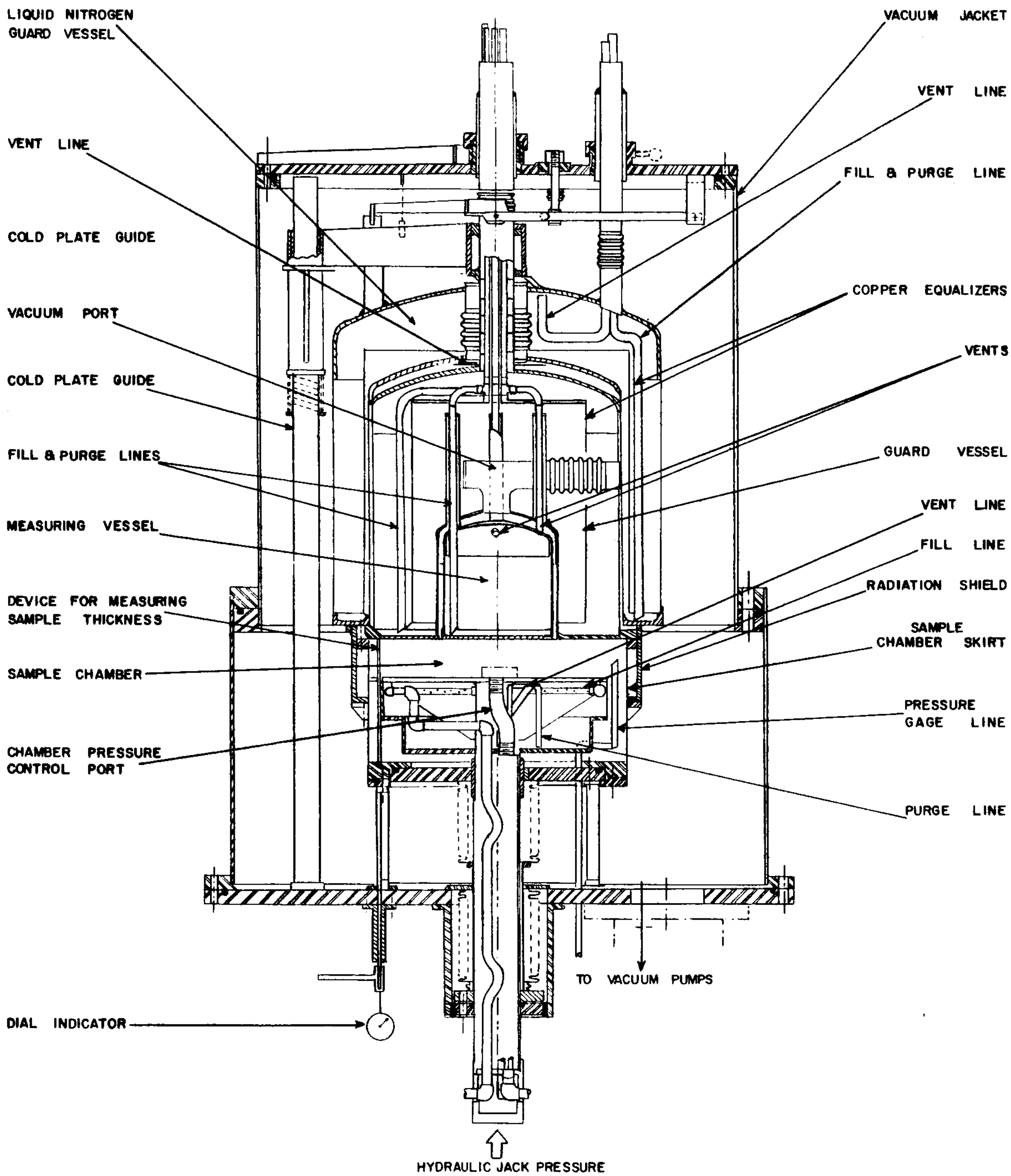

FIGURE 12 DIAGRAM OF THE DOUBLE GUARDED COLD PLATE THERMAL CONDUCTIVITY APPARATUS

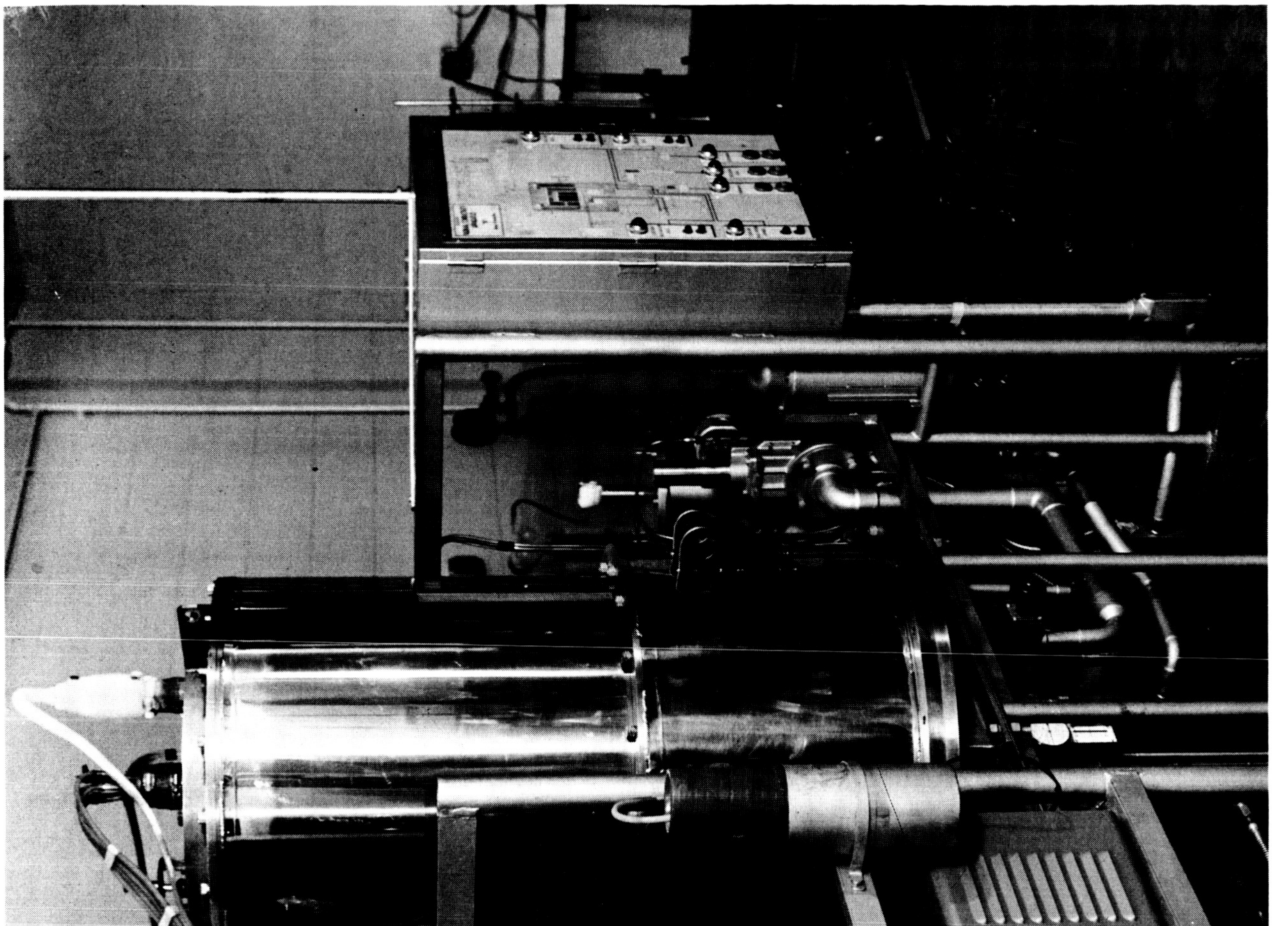

FIGURE 13 ARRANGEMENT OF DOUBLE-GUARDED THERMAL CONDUCTIVITY APPARATUS

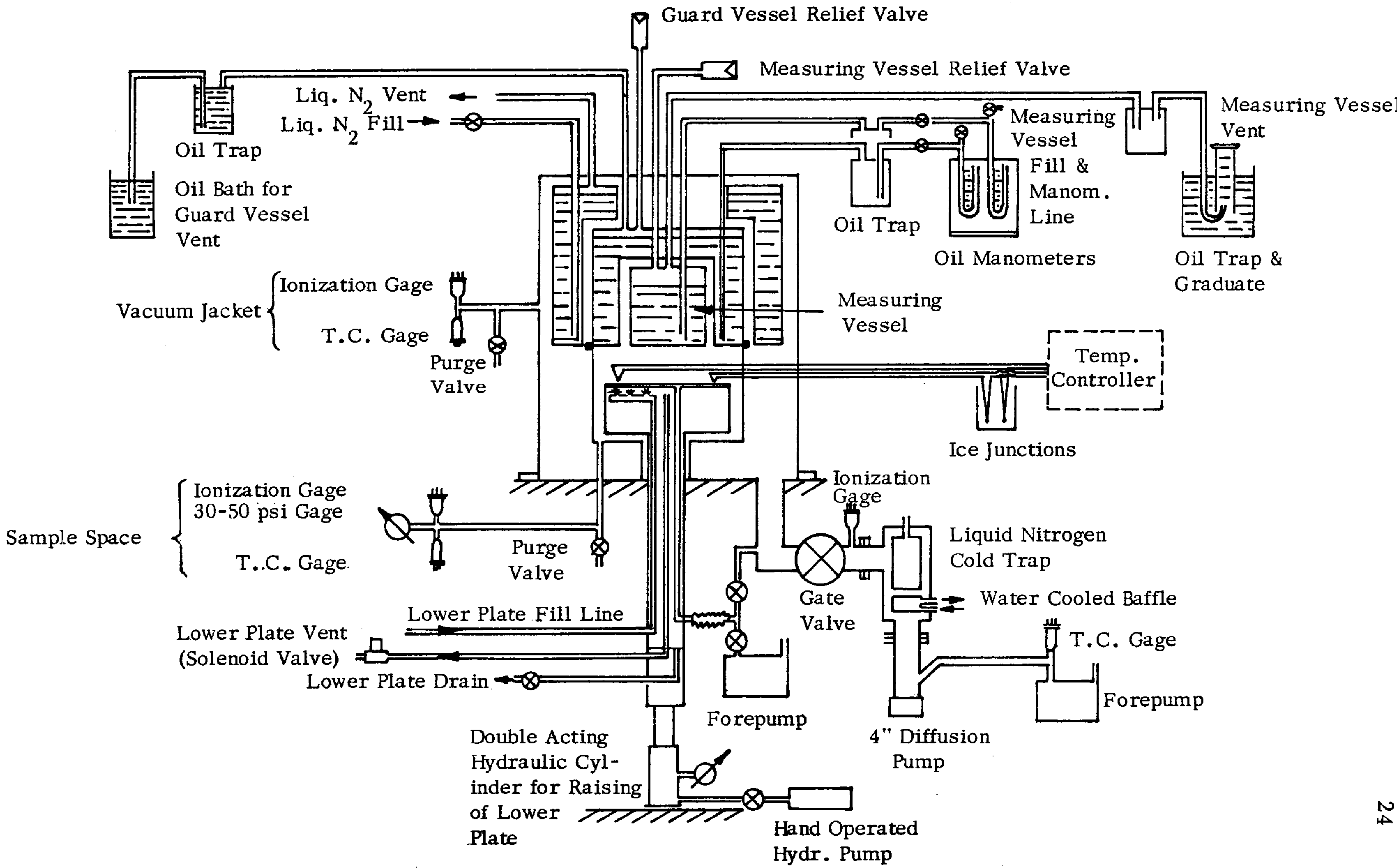

FIGURE 14 DIAGRAM OF INSTRUMENTATION REQUIRED FOR DOUBLE-GUARDED COLD PLATE THERMAL CONDUCTIVITY APPARATUS

range in thickness from 0 to 2 inches, with diameters of up to 12 inches. The measuring section of the sample is 6 inches in diameter. The sample can be exposed to any desired gas pressure from 10^{-6} Torr up to one atmosphere. In addition the test chamber can be filled with different gases while a test is in progress. The sample thickness can be changed during a test by moving the hydraulically actuated warm plate. Parallel alignment of the warm and cold plates and the thickness of the test sample can be checked during the test by three dial indicators within an accuracy of 0.001 inch. While the test is in progress, mechanical pressure from 0 to 50 psi can be applied to the sample by a hydraulically controlled and calibrated pressure unit. During a test, control can be exercised over the temperatures of the side walls of the sample chamber to control edge effects. A guard shield can be placed in thermal contact with a low temperature cryogenic cooled plate or heated by an electric heater should that be desirable.

Precautions have to be taken in the operation of the thermal conductivity apparatus so that accurate and reproducible data are obtained. Attention has to be given to the following:

a. The measuring vessel should not be filled during a test, otherwise equilibrium conditions tend to be disturbed. For a typical good insulator the measuring vessel need not be filled for several days. For low-conductivity samples several hours will elapse before equilibrium can take place.

b. Stratification of the liquid at very low boil-off rates may occur. To reduce the probability of this occurring copper wool has been introduced into the measuring vessel to provide better temperature equalization.

c. Close control over the pressure difference in the measuring and guard vessels has to be maintained to prevent stray heat leaks between them. By controlling the pressure difference to about one millimeter of oil, as measured by manometers connected to both the measuring and the guard vessels, the guard vessel can be arranged to be at a slightly higher temperature than the measuring vessel, thereby preventing boil-off gases from the measuring vessel from condensing on the walls of the guard vessel.

d. Changes in atmospheric pressure due to weather conditions during a test period are not infrequent and may lead to a corresponding change in the enthalpy of the cryogenic liquid. A correction factor can be applied to the data or a device installed to compensate for these changes by keeping a constant pressure within the measuring and guard vessels.

e. The different heights of cryogenic liquids in the guard vessel and measuring vessel result in a slightly higher temperature in the guard vessel because of the increased head of liquid above the cold plate.

f. It is essential that the warm and cold plates are kept parallel throughout a test; otherwise the samples could be subjected to unequal compression resulting in a departure from the postulated one-dimensional heat flow conditions.

g. It is not desirable to have the diameter of the sample coincide with the diameter of the measuring vessel and then surround it by a guard ring made out of the same sample material. Radiation from the warm plate could be transferred through the small opening between the sample and the guard ring and then be trapped in the clearance space between the measuring and guard vessel, leading to an increase in boil-off rates.

h. Because the sample is not infinite in extent the effect of edge losses has to be considered. The design of the sample chamber permits control over the temperature gradient of the surfaces viewed by the sample. In addition, treatment of the sidewalls permits emittance to be controlled so that reradiation or reflection from the warmer parts of the sample chamber can be reduced. In some experiments it is advantageous to know how exposed edges of the sample respond to various types of radiative heat inputs which can be provided by the aforementioned techniques.

i. Although the gas pressures in the bell jar and sample chamber are measured, no direct measurements can be made of the pressure existing within the sample. Thus, some uncertainty exists regarding the contribution of gas conduction to the overall heat transfer, particularly in those materials which are known to have a finite vapor pressure.

j. The characterization of the sample with respect to physical properties, such as the emissivity, vapor pressure, and internal structure is essential. Variation in any of these properties can lead to a difference in thermal conductivity between nearly identical samples.

The accuracy of the thermal conductivity apparatus can be established by two techniques: (1) The measuring vessel, guard vessel, and outer guard, as well as the warm plate, are filled with liquid nitrogen. If no stray heat leaks are present, the boil-off rate from the measuring vessel will be zero. (2) The apparatus is calibrated with an accurately measured electrical input to a heater placed in contact with the measuring vessel. In this test all of the parts of the apparatus are kept at liquid nitrogen temperatures so that the heat evolved by the eletrical heater is the only heat source causing the boil-off in the measuring vessel.

Figure 15 shows a typical curve for calibration with the electric heater. The sensitivity of the apparatus is sufficient to permit the measurement of heat flows on the order of 0.1 Btu/hr. through the six-inch diameter

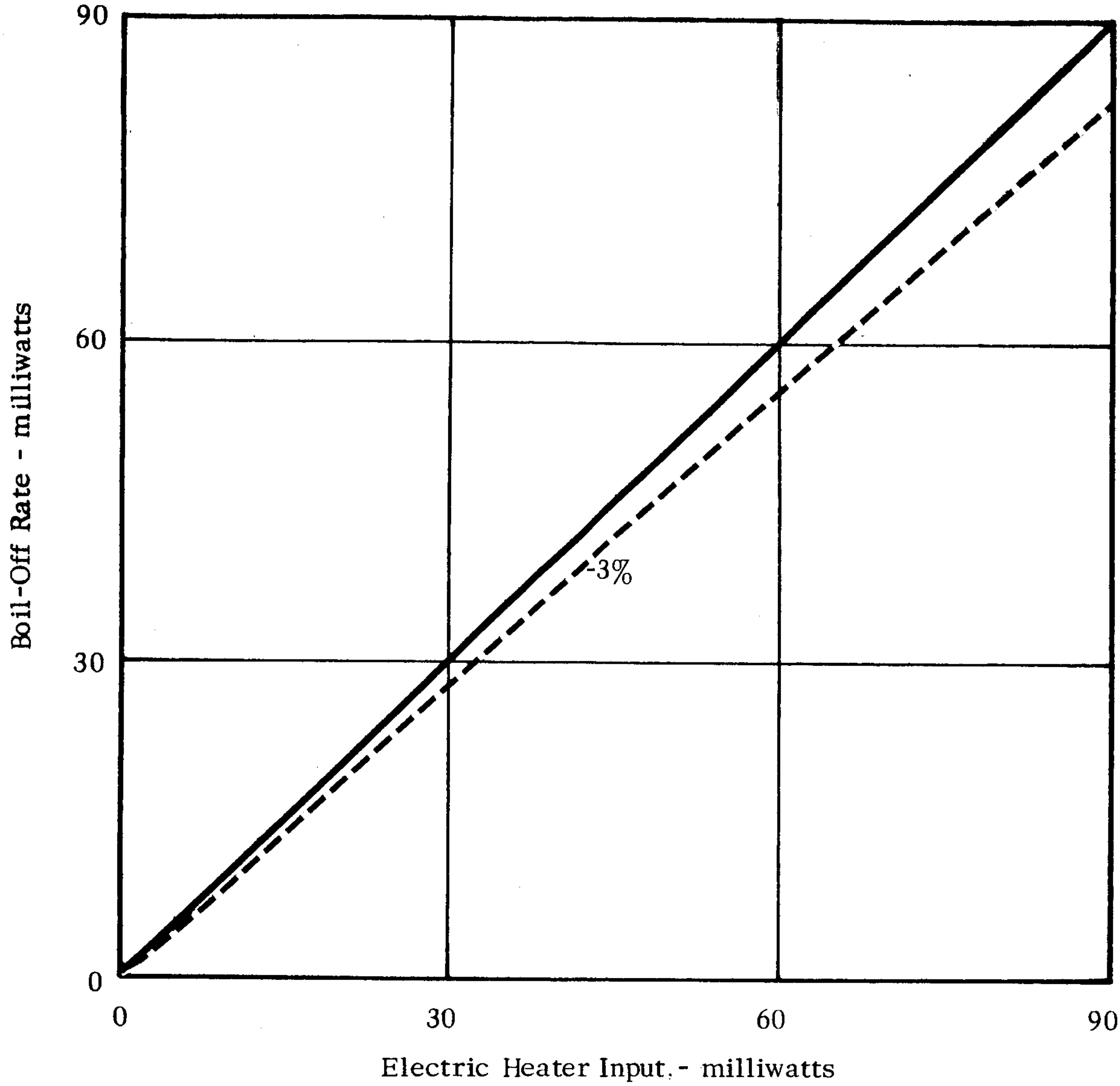

FIGURE 15 RESULTS OF CALIBRATION TESTS OF LIQUID NITROGEN TEMPERATURE ON DOUBLE GUARDED COLD PLATE APPARATUS

measuring section. Its over-all accuracy is estimated to be ±10%. Table II indicates the reproducibility of test data. A typical test on multilayer insulations takes three to five days, depending upon test conditions.

TABLE II

REPRODUCIBILITY OF TESTS WITH THERMAL CONDUCTIVITY APPARATUS

Sample: Aluminum foil and fiberglass mesh

Warm Plate Temperature: 40 F
Cold Plate Temperature: -320 F
Sample Density: 18.7 lb/ft^3

TEST #	THICKNESS (in.)	HEAT FLUX $\frac{\text{Btu}}{\text{hr-ft}^2}$	THERMAL CONDUCTIVITY $\frac{\text{Btu - in}}{\text{hr-ft}^2\text{-F}}$
1005d	0.278	0.32	0.00024
1006	0.278	0.34	0.00026
1022b	0.293	0.38	0.00028
1005b	0.390	0.28	0.00033
1022a	0.383	0.36	0.00035

V. EXPERIMENTAL RESULTS

At this time three double-guarded cold plate apparatus are in use for the measurement of the thermal conductivity of various thermal insulators at liquid hydrogen and liquid nitrogen temperatures. Two units are located at the Lewis Research Laboratory in Cleveland, and one unit is located at the Arthur D. Little Laboratory in Cambridge. The units in Cleveland have been in operation since May and August, 1962, while the unit in Cambridge has been operating since September, 1962. The following data obtained on a number of variables affecting thermal conductivity is indicative of the program being followed in this work.

A. TEMPERATURE

Table III shows the influence of temperature on the thermal conductivity of foam and multilayer insulations at liquid nitrogen and liquid hydrogen boundary conditions. In general, the thermal conductivity obtained at liquid hydrogen temperatures is slightly lower than that obtained at liquid nitrogen temperatures. This decrease in thermal conductivity appears to be due to lower pressures obtained in the sample because of more efficient cryopumping, as well as decrease in the solid conduction contribution with temperature.

Figure 16 indicates a technique which can be used to obtain a dependence of thermal conductivity on temperature during one test run. The figure shows a variation with mean temperature of the thermal conductivity of a glass wool sample as measured by a number of thermocouples imbedded in it.

B. THICKNESS

Table IV compares the influence of the thickness of the sample on thermal conductivity. The slight decrease in thermal conductivity appears to be due to a reduction in the radiation contribution to the heat transfer. One can conclude from this that for the measured thickness of foam and multi-layer insulations no major reduction in thermal conductivity will occur. In the case of the foam most of the radiation has already been absorbed. In the case of the multilayer insulation, the radiation contribution with increasing number of layers decreases; however, the solid conduction contribution increases at a greater rate so that it is possible to find an optimum number of radiation shields for a specific material combination.[19] Figure 17 illustrates this behavior.

The effect of mechanical pressure on the thickness and thereby on the heat flux through a sample of a multi-layer insulation is of considerable practical

TABLE III

INFLUENCE OF TEMPERATURE ON THERMAL CONDUCTIVITY

SAMPLE	TEST #	HOT PLATE TEMP (F)	COLD PLATE TEMP (F)	HEAT FLUX $\frac{Btu}{hr\ ft^2}$	THERMAL CONDUCTIVITY $\frac{Btu - in}{hr - ft^2 - F}$	SAMPLE THICKNESS (in.)	SAMPLE DENSITY (lb/ft^3)
Fiberglass reinforced foam	1018a	66	-320	22.6	0.0458	0.782	5
Fiberglass reinforced foam	1018c	69	-423	21.6	0.0346	0.782	5
Aluminum foil and fiberglass mat 50% perforations	1031a	63	-320	0.231	0.00018	0.301	12
Aluminum foil and fiberglass mat 50% perforations	1031b	61	-423	0.200	0.00012	0.301	12
Aluminum foil and fiberglass mesh	1006	40	-320	0.341	0.00026	0.320	19
Aluminum foil and fiberglass mesh	1030c	64	-423	0.323	0.00019	0.320	19

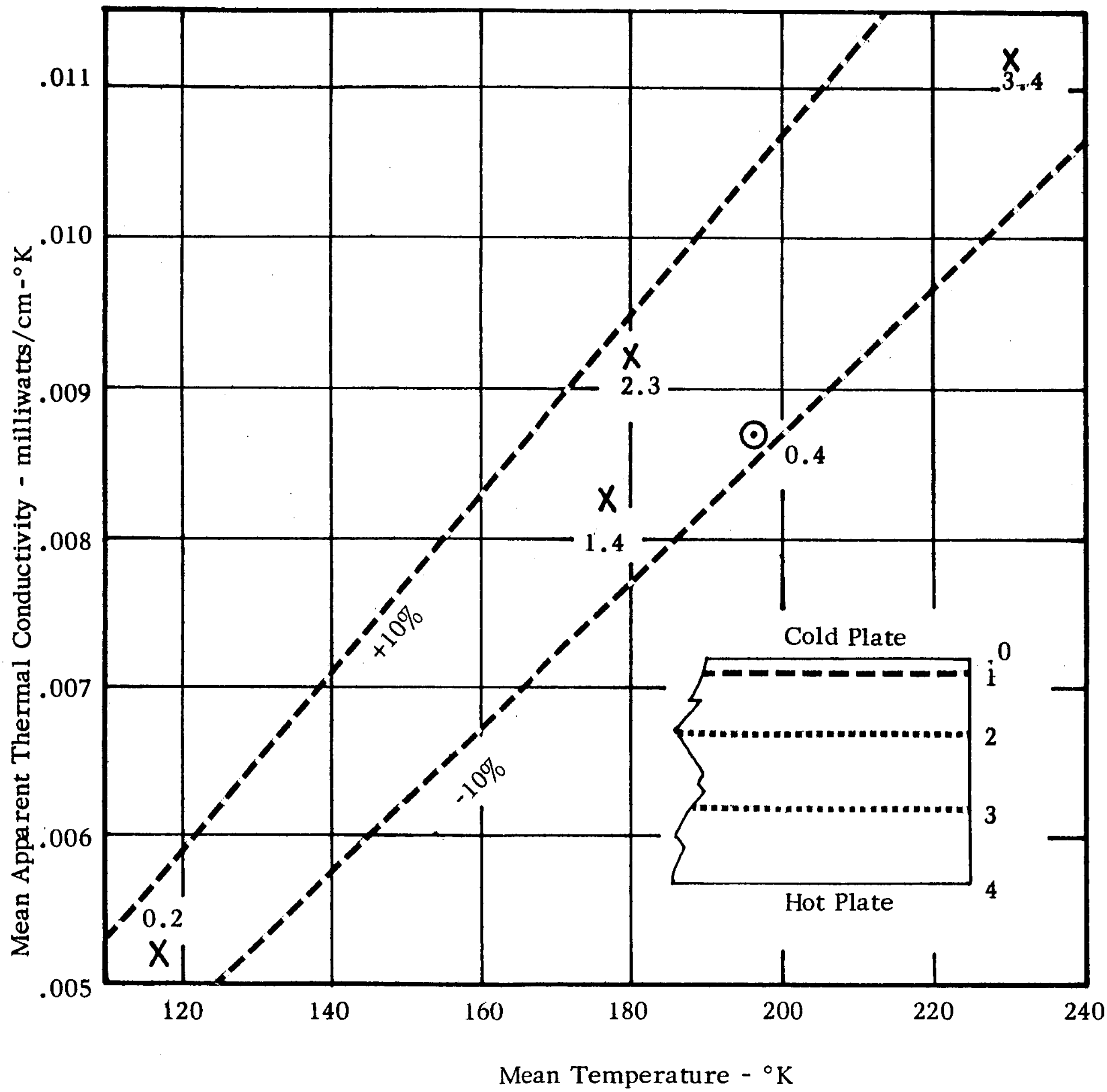

FIGURE 16 THE DEPENDENCE OF THERMAL CONDUCTIVITY ON TEMPERATURE FOR GLASS WOOL

TABLE IV

INFLUENCE OF THICKNESS ON THERMAL CONDUCTIVITY

SAMPLE	TEST #	DENSITY $\frac{lb}{ft^3}$	THICKNESS in.	HEAT FLUX $\frac{Btu}{hr-ft^2}$	THERMAL CONDUCTIVITY $\frac{Btu - in}{hr - ft^2 - F}$	PRESSURE IN SAMPLE CHAMBER mm Hg	HOT PLATE TEMP. (^oF)	COLD PLATE TEMP. (^oF)
Polyurethane foam	1014c	4	0.246	224.0	0.149	760(N_2)	50	-320
Polyurethane foam	1015d	4	1.00	50.8	0.145	760(N_2)	50	-320
Aluminum foil and fiberglass mesh	1006	19	0.278	0.341	0.00026	10^{-5}	40	-320
Aluminum foil and fiberglass mesh	1007	19	0.560	0.122	0.00016	10^{-5}	40	-320

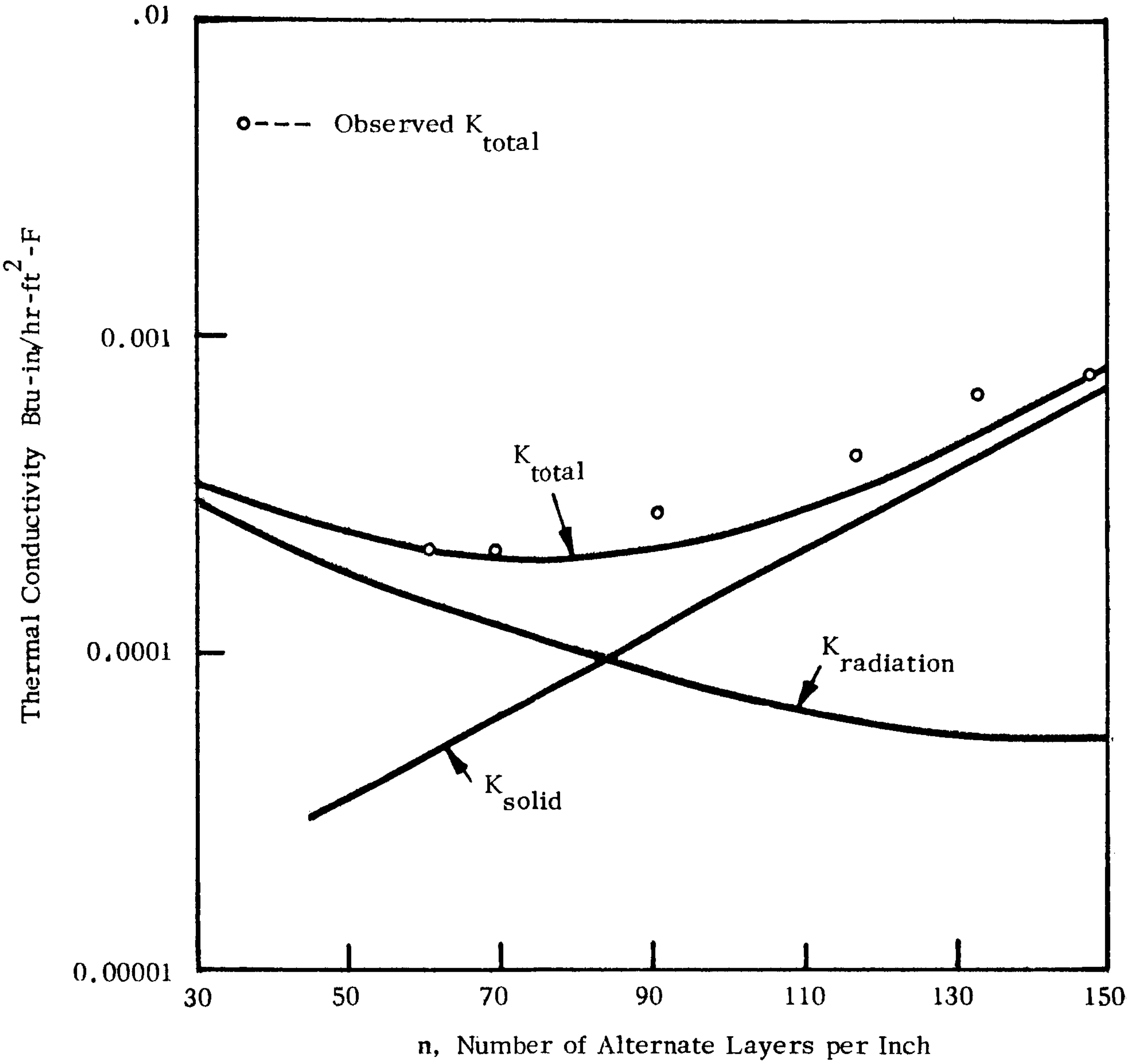

FIGURE 17 THE CONTRIBUTION OF SOLID CONDUCTION AND RADIANT HEAT TRANSFER TO THE THERMAL CONDUCTIVITY OF MULTI-LAYER INSULATIONS

Source: D. I. -J. Wang, "Multiple Layer Insulations" significance. An increase in mechanical pressure will cause the compression of the radiation shields and spacers into a thinner sandwich of higher density. The density, rather than the mechanical pressure, is used as a variable in the results presented in Figure 18. The increase in heat flow through an insulation is proportional to the ratio of thermal conductivity to thickness. Thus, doubling the conductivity and decreasing thickness by one-half leads to a four-fold increase in heat flow.

The aluminized and crinkled Mylar is very strongly dependent on mechanical pressure. Table V illustrates the results of compression on a sample consisting of 20 Mylar shields and indicates the hysteresis effect obtained by repeated compression. The density of the sample was calculated from the thickness, which was measured during the test in a fixture constructed for this purpose.

C. CONTACT RESISTANCE

Table VI shows the results of tests of the influence of contact resistance between a surface and a foam sample. When the space around the sample was evacuated, the contact resistance between the sample and the surface increased to the extent that the heat flux decreased by a factor of four. Since the foam sample had approximately a 90 percent closed cell structure, the conductivity of the bulk of the foam appeared to be unaffected by the evacuation of the gas on the outside. Thus, the increase in the resistance appears to be due to a removal of gas from broken cells near the surface. A similar increase in contact resistance was observed on a second sample of foam when a thin evacuated gap was formed between the cold plate and the sample.

D. EMISSIVITY

The effect of emissivity on the thermal conductivity of a multilayer sample can be illustrated by tests made on an assembly of aluminized Mylar. In one sample the thermal conductivity was 0.001 Btu-in/hr-ft^2-F. After a sample was made up from a new shipment from the supplier, a thermal conductivity of 0.0004 Btu-in/hr-ft^2-F was measured. On checking with the supplier, we learned that the aluminized coating in the sample with the higher thermal conductivity was near the border line of acceptable quality (electrical resistance of 7 ohms/cm^2).

In another test, aluminum foils in a sample were replaced by foils just received from the supplier. An 80 percent decrease in thermal conductivity was obtained, indicating that either oxidation or a different treatment of the foil had resulted in a reduced emissivity and thereby a lower thermal conductivity.

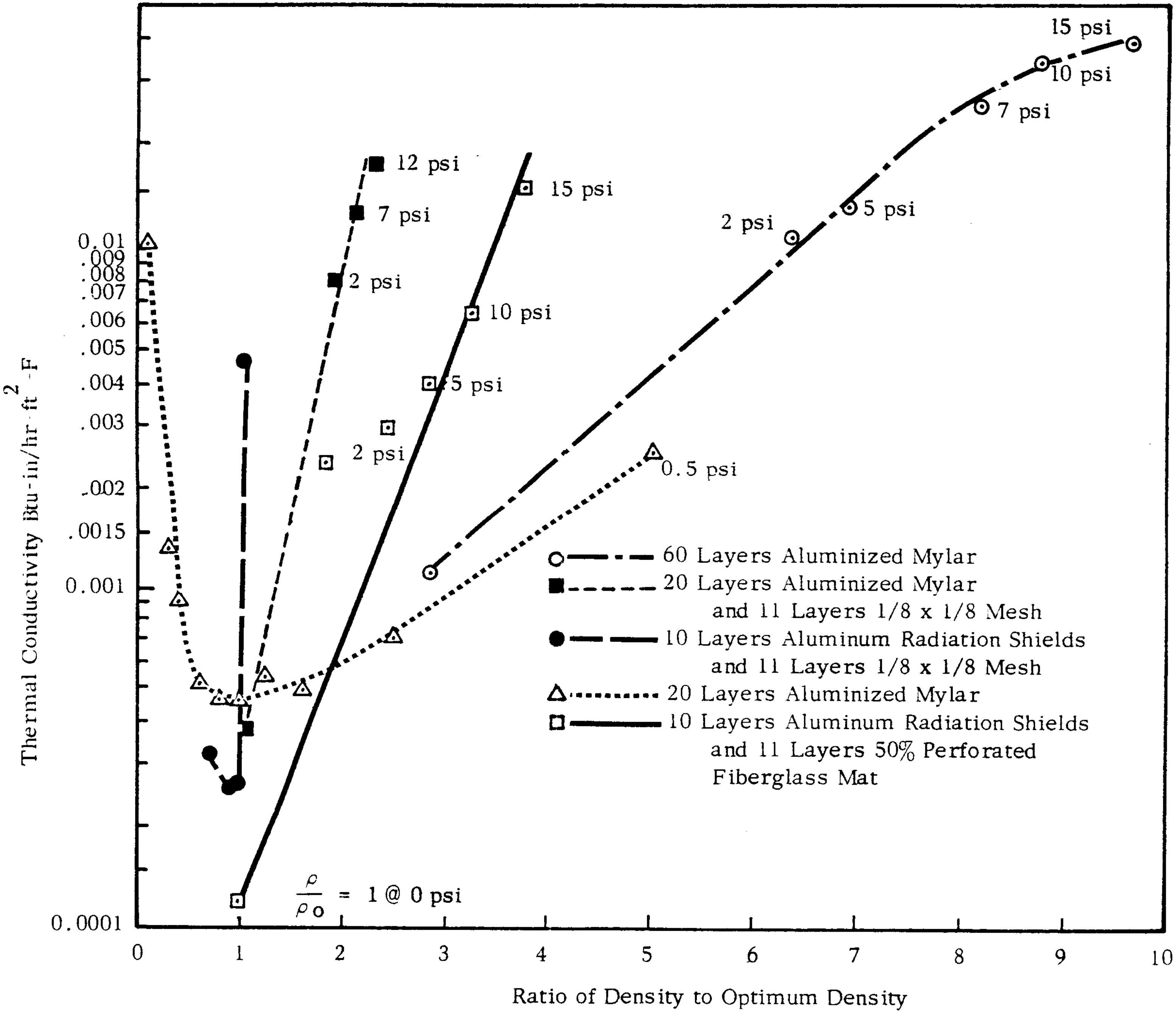

FIGURE 18 THE DEPENDENCE OF THERMAL CONDUCTIVITY ON DENSITY OF MULTI-LAYER INSULATIONS

TABLE V

EFFECT OF COMPRESSION ON MYLAR FOILS

	Compressive Load (lbs/in^2)	Sample Thickness (in.)	Density (lbs/ft^2)
A. 0.00025 inch crinkled Mylar, uncompressed	0.000	1.000	0.44
	0.006	0.500	0.87
	0.018	0.310	1.39
	0.054	0.180	2.31
	0.117	0.125	3.46
B. Repeat Tests	0.017	0.142	3.05
	0.054	0.080	5.42
	0.164	0.055	7.90
	0.185	0.044	9.87
C. Repeat Tests	0.018	0.059	7.36
	0.054	0.051	8.50
	0.164	0.047	9.22
D. Repeat Tests	0.185	0.042	10.31

TABLE VI

THE INFLUENCE OF CONTACT RESISTANCE BETWEEN PLATES AND SAMPLE ON THERMAL CONDUCTIVITY

	TEST #	DENSITY $\frac{lb}{ft^3}$	THICKNESS (in.)	HEAT FLUX $\frac{Btu}{hr - ft^2}$	THERMAL CONDUCTIVITY $\frac{Btu - in.}{hr - ft^2 - F}$	PRESSURE IN SPECIMEN CHAMBER (mm Hg)
Polyurethane foam	14c	3.85	0.246	224	0.141	$760(N_2)$
Polyurethane foam	14b	3.85	0.246	63.8	0.040	10^{-5}
Polyurethane foam	14a	3.85	0.280	23.3	0.0176	10^{-5}(gap)
Fiberglass reinforced foam	18b	5.0	0.758	41.2	0.0808	10^{-5}
Fiberglass reinforced foam	18a	5.0	0.782	22.6	0.0458	10^{-5}(gap)

Since the contribution of radiation heat transfer is proportional to the emissivity, considerable quality control is required to assure that reproducible emissivities are obtained.

It is possible that radiation shields placed very close to each other may cause a substantial increase in the radiation heat transfer rate.[20] This increase could be caused by the effect of constructive interference when the wave emitted by a foil is reflected back and forth in the small gap between two foils. This can be of importance in the transmission of energy across the gap when the spacing is very small compared to the wavelength of the radiation, as may be occurring at cryogenic temperatures.

E. GAS PRESSURE

The influence of gas pressure on residual gas conduction is of considerable practical importance. Figure 19 illustrates the behavior of a multi-layer insulation as a function of gas pressure. The insulating effectiveness of multi-layer insulations tends to decrease rapidly with increased gas pressures, so that at pressures between 10 and 100 microns their thermal conductivity tends to be higher than for an opacified evacuated powder.

This dependence of thermal conductivity on residual gas conduction indicates that every effort has to be made to obtain a low pressure within the multi-layer insulation, particularly after reaching space. If the low pressure existing in space is to be relied upon to assist in the evacuating of the multi-layer insulation, steps have to be taken so that the outgassing rate from the insulation surfaces, as well as gas diffusion from the tank due to very small leaks, can be accommodated. Consideration can be given to the geometrical arrangements of the foils such that they can enhance the pumping of the gas without reducing the effectiveness of the foils as a radiation shield. One such technique is to perforate the foil; however, the perforations have to be small enough so as not to lower the effective emissivity of the foil, since radiation can enter by the same path by which molecules leave.[21] Thus, 1 percent of perforation would cause an increase of emissivity from 0.05 for the undisturbed foil to 0.07, corresponding to a possible 40 percent increase in heat flow.

An analysis of gas conduction through multi-layer insulation shows that for a typical assembly of 100 foils, the allowable diffusion rate from the liquid hydrogen fuel tank is about 1 pound of hydrogen per year for each 1000 sq. ft. of tank surface assuming 1% perforation. This would correspond to a maximum pressure of about 10^{-4} Torr within the insulated space when exposed to space vacumm. This pressure would not tend to lower the insulating effectiveness appreciably. However, this diffusion rate on the ground could not be easily removed by a vacuum pump connected to an insulation system and could

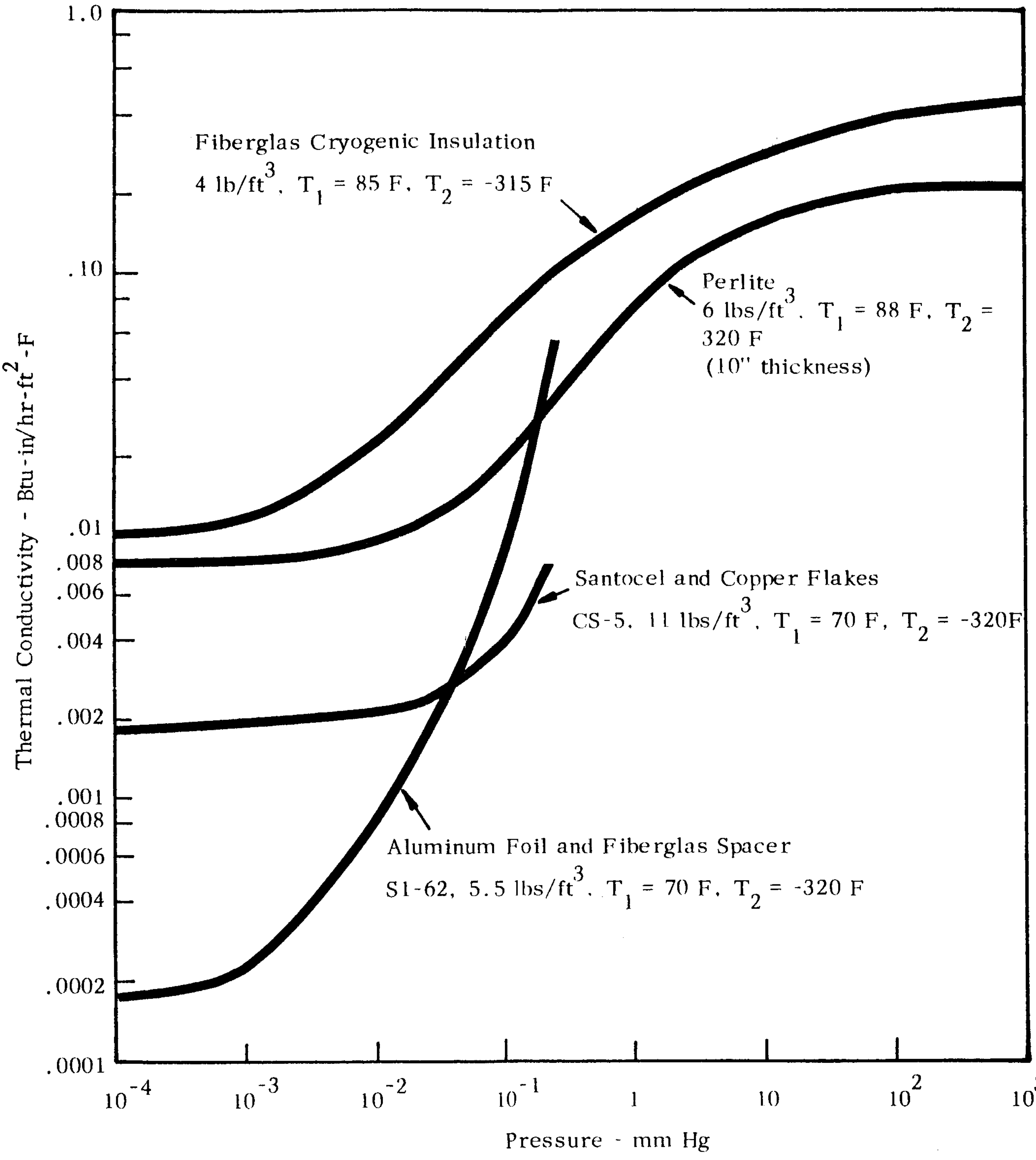

FIGURE 19 THE DEPENDENCE OF THERMAL CONDUCTIVITY ON GAS PRESSURE OF TYPICAL POWDERS, FIBERS, AND MULTI-LAYER INSULATIONS

Source: M.A. Dubs and L.I. Dana, "Superinsulation for the Large Scale Storage and Transport of Liquified Gases"
D.B. Chelton and D.B. Mann, Cryogenic Data Book
R.M. Christiansen, M. Hollingsworth, Jr., and H.N. Marsh, Jr., "Low Temperature Insulating Systems"

lead to a pressure rise of undesirable proportions. Therefore, more stringent criteria for diffusion rates would have to be used to assure that pressure would not rise above a maximum value consistent with desired insulating effectiveness.

VI. REQUIREMENTS FOR APPLICATION OF MULTILAYER INSULATION TO LIQUID HYDROGEN TANKS

After examining the data on thermal conductivity of multi-layer insulations, it is not surprising that the impression is gained that an adequate solution is near at hand for overcoming the insulation problems of large liquid hydrogen tanks for space vehicles. We can conclude from available data that multi-layer insulations of quite reasonable thickness will be adequate to reduce the boil-off losses to acceptable proportions. In fact, only insulation systems of such high effectiveness are capable of meeting the objectives of long-term space missions. However, multi-layer insulations have been successfully applied only to ground-based, double-walled storage vessels and no large insulated cryogenic tanks have yet undergone the rigors of space flight.

In the following discussion, the design parameters influencing the selection of insulations and their application to a tank are reviewed and areas where design problems still remain unsolved are pointed out.

A. DESIGN PARAMETERS

Prior to the selection of a suitable insulation system, several design parameters have to be established:

1. Heat Balance For Liquid Propellant Storage Systems

The heat balance will be influenced by:[22]

a. The radiative heat input from the sun and other solar system bodies. This input will depend upon the degree of orientation of the vehicle. In many cases, controlled orientation can materially simplify the thermal insulation problem.

b. Radiative heat inputs from other portions of the vehicle. These inputs can be minimized by judicious placement of liquid-oxygen tanks to shield liquid-hydrogen tanks from the engine.

c. Conductive heat inputs through pipes and supports. If the insulating effectiveness of the multiple-foil insulation is not to be offset, these heat inputs must remain a small percentage of the over-all heat flow into the tank.

d. Radiative heat losses to space. The outside surface coating should accept as little radiation as possible while radiating strongly to the surroundings.

2. Methods Used For The Storage Of Liquid Propellants

Three principal methods can be identified. Liquid propellants can be stored in:

a. A vented tank,

b. A nonvented tank with a variable pressure, or

c. A refrigerated tank in combination with the above.

The specific method chosen will have an influence on the thermal stratification and condition of the liquid hydrogen during both ground standby and space flight. The approach selected may also influence the tank shape, e.g., in a nonvented, variable-pressure tank, an increase in pressure may lead to a change in the shape of the vessel. Analysis has shown(1) that a definite advantage exists in supplying refrigeration for a number of mission profiles of interest because it eliminates boil-off losses and reduces the thickness of insulation required. (See Figure 1.)

3. Meteoroid Protections

Based on our present understanding of the hazards to large liquid propellant tanks by meteoroid impact, it appears that an effective meteoroid bumper has to be provided as a means of assuring the survival of the tank in space during extended missions. This bumper which may be erected after the vehicle has left the earth's atmosphere could perform an additional thermal protection function by having its surfaces coated so as to minimize radiant heating by outside sources.

4. Production Consideration

Large liquid hydrogen tanks will undergo a number of certification tests, static firing tests, and inspection tests, all involving transportation and handling by a variety of means before erection at the launch site. These conditions impose practical limits on the type of thermal protection systems selected, from the point of view of sensitivity to damage, ease of repair, maintenance under field conditions, and compatibility with inspection procedures.

Among these inspection procedures leak detection of the complete tank structure at various points of tests and at the launch site is of great importance. The thermal protection system has to be compatible with procedures for leak detection, as any undetected leakage into the protection system can nullify its insulating effectiveness under space conditions or even create explosion hazards.

B. METHODS OF INSTALLATION

Once a heat-transfer analysis and optimization procedure has been carried out, and the thermal insulation system has been characterized as to the number and type of foils, the type of spacer, the over-all density requirements, and the thermal conductivity of selected thermal insulation systems has been measured, installation techniques can be investigated.

The following typical installation techniques can be used as examples. (See Figure 20):

1. Straight wrap, with alternate layers arranged to cover joints formed by the previous layer.

2. Spiral wrap, which can be more easily adapted to curved geometries.

3. The shingle assembly in which preassembled layers are arranged to overlap at the joints.

4. Segmented layers, which can be more easily applied over hemispherical ends.

5. The successive skirt wrap.

6. The orange peel wrap, which is particularly suited for spherical tanks.

Once a form of application has been selected, based on tank size and material and fabrication considerations, the insulation has to be designed to stay in place in spite of aerodynamic and vibration forces without sacrificing insulation performance. The most obvious method for achieving this would be to wrap the foils tight enough so that the frictional forces would be sufficient to withstand the applied forces. However, compression of the insulation can lead to considerably higher thermal conductivity. (See Figure 18.) A compromise solution has been suggested involving the use of bands which can provide a controlled degree of compression sufficient to withstand an axial force, e.g.

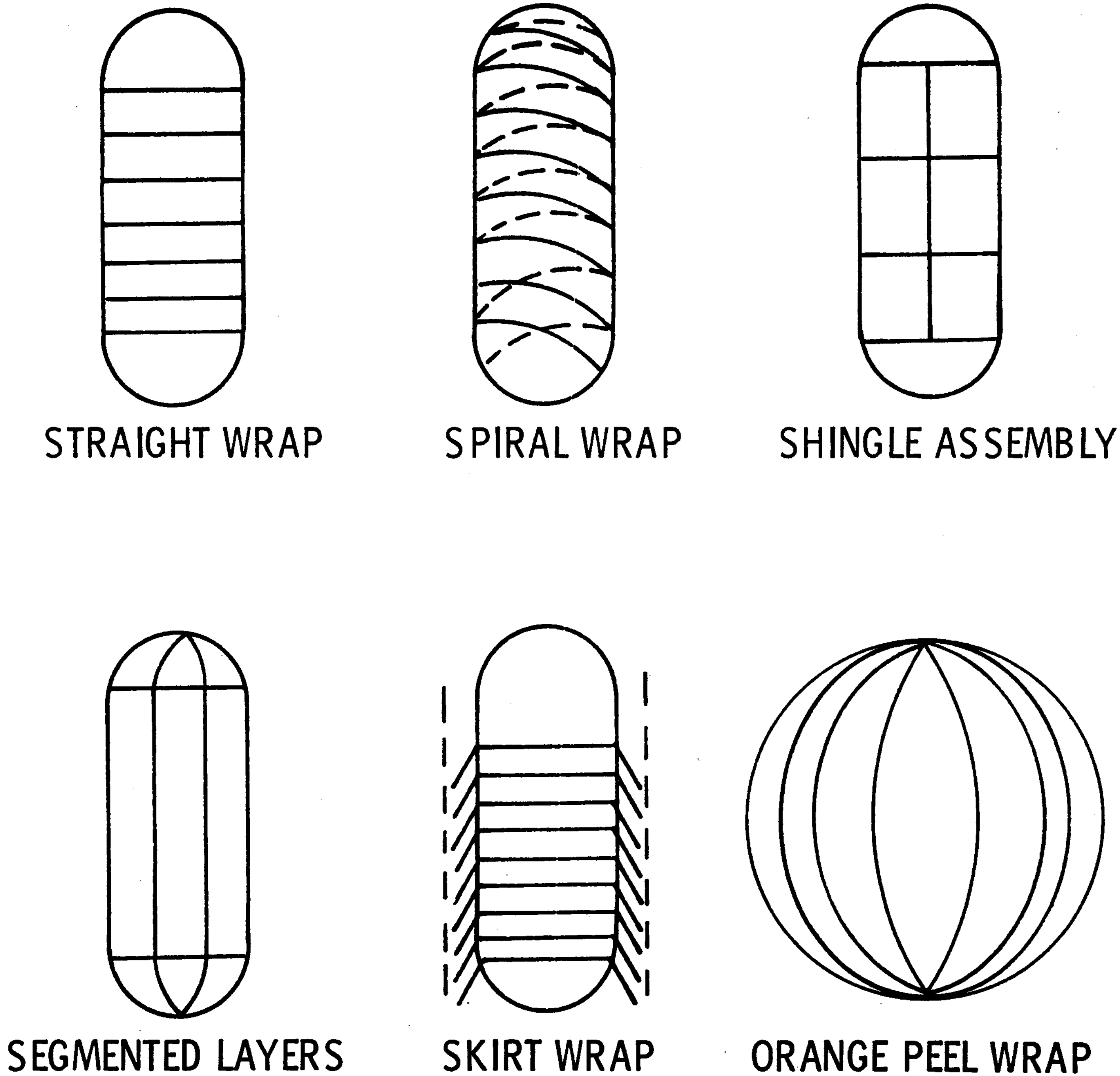

FIGURE 20 TECHNIQUES FOR INSTALLING MULTI-LAYER INSULATION ON A TANK

compression to withstand 4g resulted in a 27% increase in heat flux.[23] The larger diameter tanks may not permit this approach to be used. Another means would be to provide attachments designed to penetrate the insulation without increasing the over-all heat flux appreciably.

The particular method of application may be further influenced by the following considerations:

1. The insulation, when applied to a warm tank, will have to accommodate thermal contraction when the tank is filled, and, conversely, in a tank used for nonvented storage, insulation will have to accommodate expansion due to an increase in gas pressure. In a sense, the insulation has to be capable of breathing with the tank without opening up joints and seams or suffering deterioration.

2. Ground standby conditions may require that the insulated space be ventilated with a noncondensing gas to prevent condensation of air and other vapors. Provisions have to be made allowing this gas to escape from within the insulation as the vehicle leaves the earth's atmosphere. Two conditions have to be met: (a) the gas has to escape at a sufficient velocity but without causing pressure differentials which could lead to rupture of the foil insulation and (b) the thermal conductivity of the multifoil insulation must be reduced to the desirable low values in the shortest possible time by using the external space vacuum for pumping.

3. The temperatures generated by aerodynamic heating on the outside of the insulation system require that suitable materials be employed as a means of forestalling a serious deterioration in the desirable emissive properties of the outside coatings or a marked deterioration in the strength of the outside layers of the insulation making them incapable of withstanding aerodynamic forces. Thus, it is necessary to select and to install materials with desirable strength characteristics and physical properties for the specific temperature range.

C. CONDUCTIVE HEAT LEAKS AT DISCONTINUITIES DUE TO JOINTS, SUPPORTS, AND PIPING ATTACHMENTS.

Once the basic requirements for the payload and the weight of the cryogenic propellants to complete specific missions are known, the insulating effectiveness of the thermal protection system and the maximum allowable heat leak through discontinuities caused by joints, supports, and piping attachments can be specified. As a rough design guide, no more than 30 percent of the total heat leak into the tank should be caused by such discontinuities. Heat leaks that are higher would tend to nullify the effectiveness of the insulation and might not warrant the application of multi-layer insulation.

Analyses have been carried out on the heat leaks due to insulation penetrations and discontinuities.[24] These can be classified as:

a. Weak thermal shorts where a linearized radiation boundary condition can be applied with an acceptable accuracy, e.g., where the surface temperature is within 10% of the adiabatic wall temperature.

b. Strong thermal shorts when the depression of the temperature at the outer surface of a foil is such that the boundary condition is no longer linear, and

c. Absolute thermal shorts where the penetration is so highly conductive that the tank wall and the outside surface of the short are nearly at the same temperature.

Conductive heat leaks are of particular concern after the space vehicle has left the earth's atmosphere and is subject to the various radiative heat inputs from the sun and other solar system bodies.

1. Weak Thermal Shorts

Multi-layer insulations can be used with materials such as Teflon to meet structural requirements and to satisfy the weak thermal short condition. Steel or other separators would not fall into the category of weak thermal shorts unless insulator could be installed between the multilayers and the penetration. In the case of aluminized Mylar weak thermal shorts could be obtained for reasonable thicknesses of very poor conducting structural materials when the thickness of the Mylar assembly is greater than one inch. Figure 21 shows the maximum widths of various materials penetrating multilayer insulations when the weak thermal short approximation is applicable.

The design of the thermal protection system even with weak thermal shorts may have to be modified so that the over-all heat leak is within permissible limits. To illustrate this point a cylindrical tank 10 feet in diameter and 20 feet long will contain about five thousand pounds of liquid hydrogen and its cylindrical surface area would be 628 sq. ft. If there are two 10-mil plastic dividing strips around this tank, then it can be shown that the heat leak through these strips would be equivalent to that of 520 sq. ft. of insulation. Therefore, the over-all heat leak would be increased by more than 80 percent.

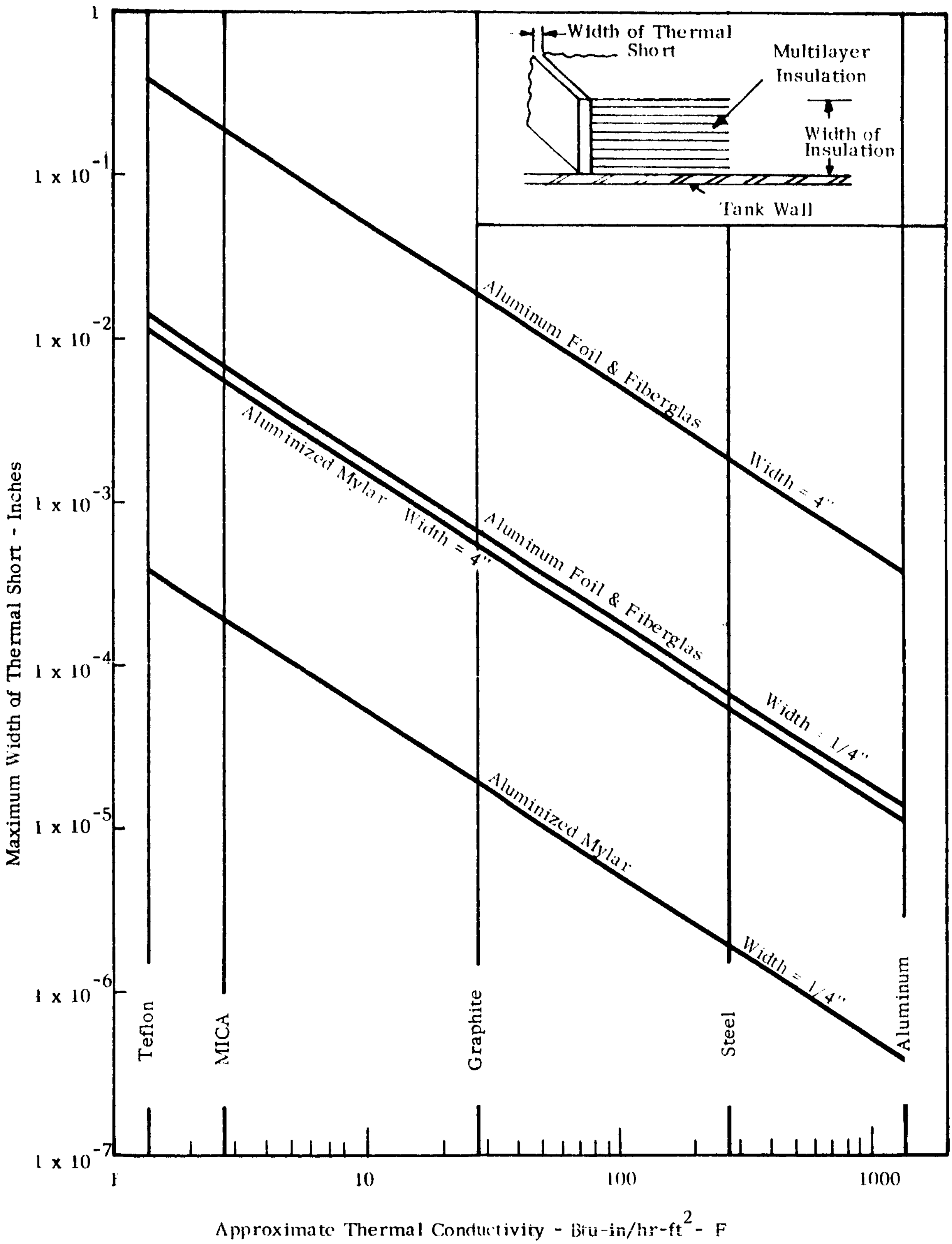

FIGURE 21 THE MAXIMUM WIDTH OF VARIOUS THERMAL SHORT MATERIALS PENETRATING MULTI-LAYER INSULATIONS

2. Strong Thermal Shorts

Because heat leaks associated with strong thermal shorts are greater than would occur from weak thermal shorts, it is evident that strong shorts have to be avoided or if unavoidable the design has to be arranged so that they can be converted to weak thermal shorts.

The approach in reducing the heat leak due to a strong thermal short is to separate them from the ends of the multilayer insulation, thus bringing them into the range of weak thermal shorts. Examples of the approach that can be followed to decrease the effect of a strong thermal short are shown in Figure 22. This figure shows the following:

a. A strip of evacuated insulation (fibers or foam) separating multilayer insulation from a thermal short.

b. A square section of evacuated insulation placed in a corner to protect the foils in one direction from being shorted by those in the other direction when the radiant flux impinges equally on both surfaces of the insulation.

c. A square section of evacuated insulation the upper side of which is kept adiabatic to protect a joint between a tank wall and a large pipe or between a tank wall and a structural support with negligible heat flow from the warm end to which these elements would be connected.

d. A square section of evacuated insulation provided with a protective radiation shield, the upper side of which receives the same radiant flux as does the multi-layer insulation.

Calculations have shown that for a surface temperature of 300 K and a liquid temperature of 20k, the heat leak due to the presence of a one-foot length of evacuated insulation is equivalent to from 0.5 to 1.5 sq ft of undisturbed multilayer insulation.

Based on these considerations it can be seen that joints, supports, and piping attachments have to be given considerable attention and special design techniques have to be employed so that the heat leak can be kept to acceptable proportions.

3. Methods of Tank Support

When the hydrogen tank is to be placed inside a shroud, the support methods shown in Figure 23 can be employed. The supports are arranged in such a manner that loads in several axial directions can be accommodated. The

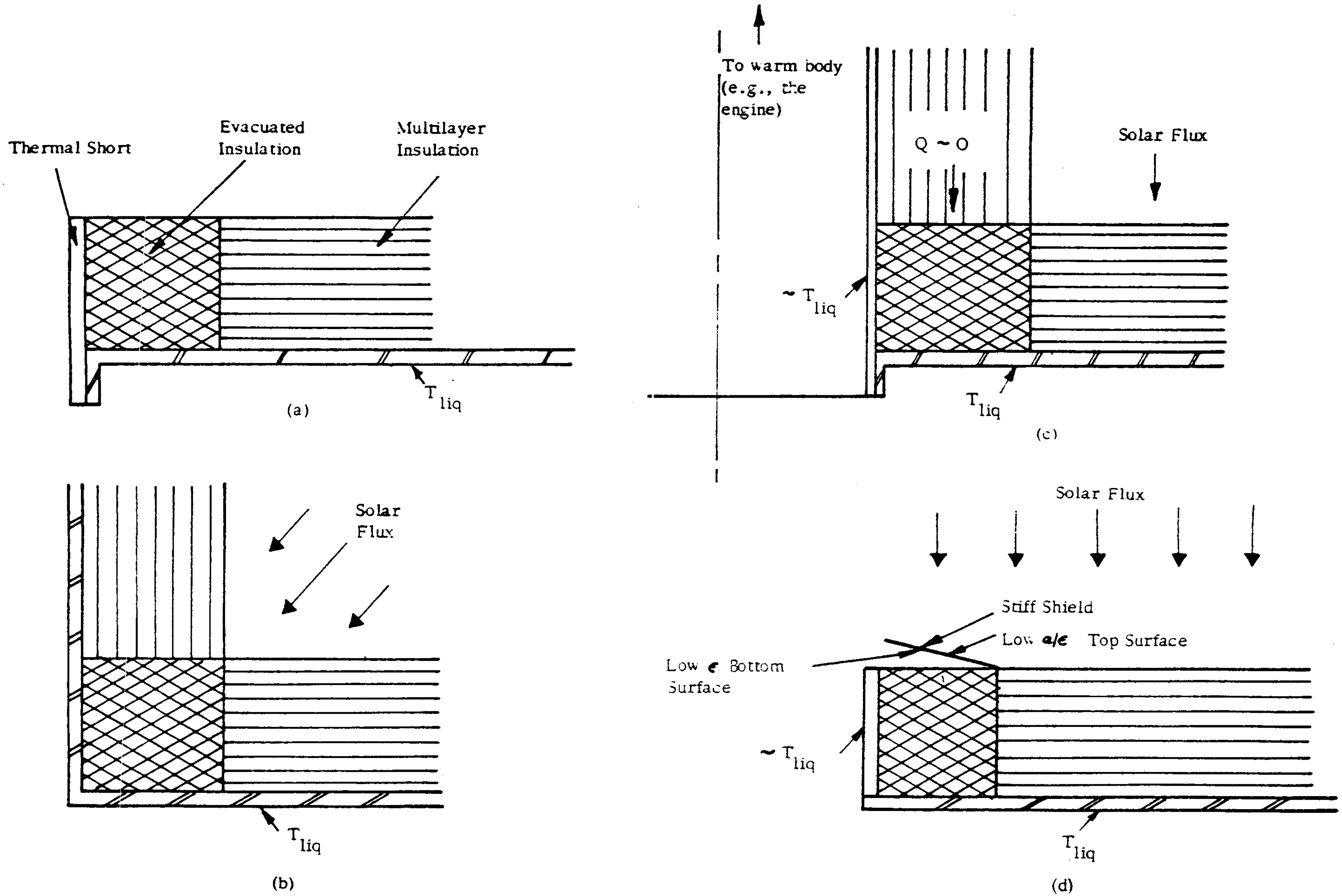

FIGURE 22 TECHNIQUES FOR DECREASING THE EFFECT OF THERMAL SHORTS

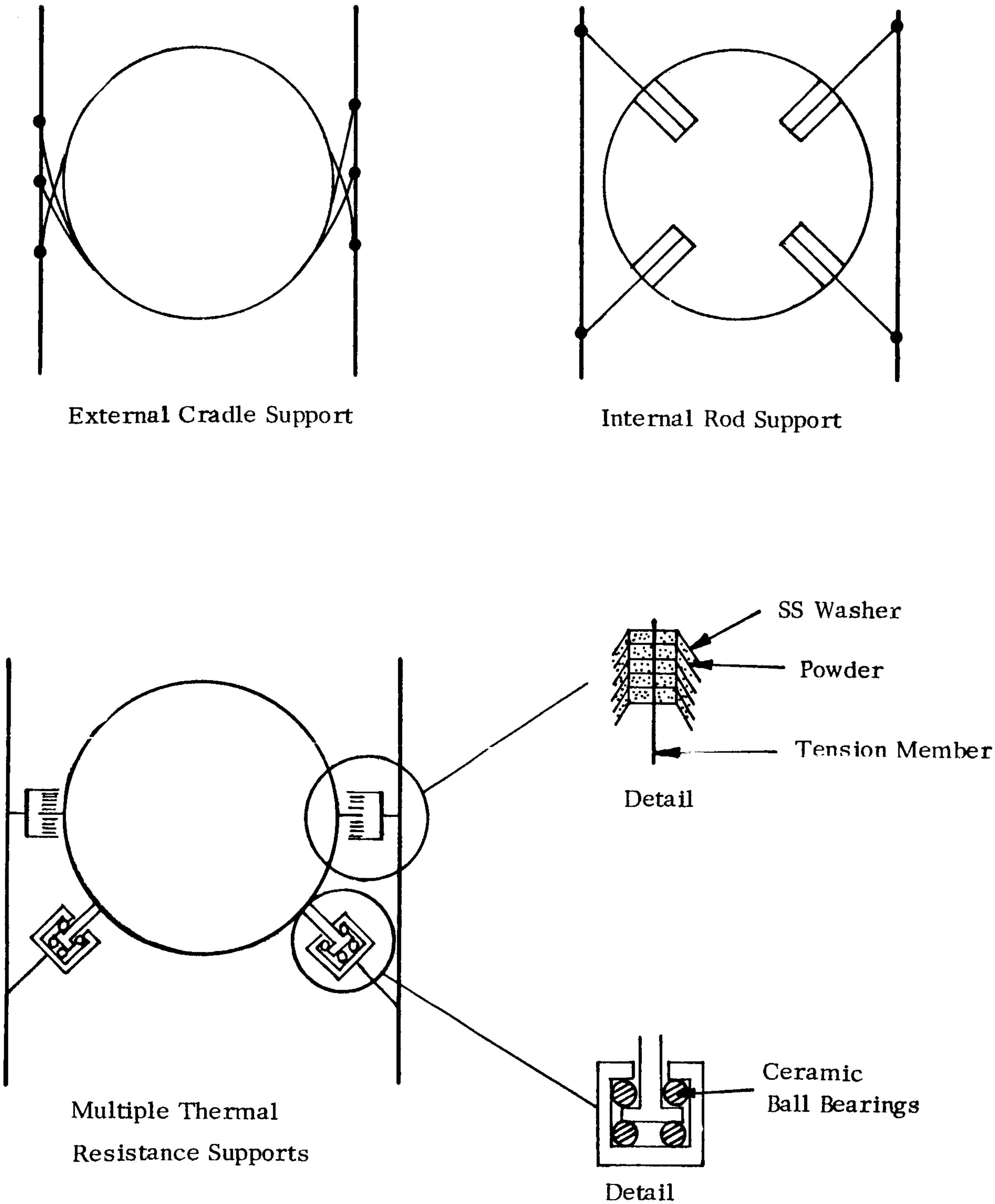

FIGURE 23 TECHNIQUES FOR SUPPORTING TANKS WITHIN A SHROUD

external cradle support requires that the insulation be under a compressive load during acceleration of the vehicle. However, a system of springs can be arranged to lift this compressive load once zero gravity conditions have been achieved. Rod supports or cables can be anchored outside the tanks with connection to re-entrant pipes.

Stainless steel washers dusted with insulating powders have been found to give excellent performance in the support of heavy loads.[25] Such washers are primarily designed to withstand compressive loads, although they can be formed to take limited radial loads. (See Figure 23.) Passing tension members through the washers can provide additional structural rigidity. A limited amount of work has been done in using ceramic bearings to take radial and axial thrust. The object here is to minimize the heat flow across the contact area between the ball and the bearing surface by minimizing thermal conductivity and maximizing the modulus of elasticity.

D. LAUNCH ENVIRONMENTAL INFLUENCES

Although the design of the thermal protection system is primarily directed towards meeting the space-flight portion of the mission, the design will be greatly influenced by the ground-standby and boost-phase portions of the mission. Because these phases may control the practical application problems, it is of interest to consider them in detail.

1. Ground Standby

To allow for the delay time which can be expected while the fueled vehicle is kept on the launch pad, insulation has to be adequate to reduce the magnitude of boil-off losses. The following approaches can be used to achieve this:

a. An outside jacket is provided, which permits the insulation to be evacuated on the ground by cryopumping after the insulation has been flushed with an easily condensible gas such as CO_2 to exclude residual hydrogen in the air. Because the weight of a rigid shell capable of withstanding atmospheric loads would be prohibitive, a light-weight jacket which encloses the insulation can be considered. (See Figure 24a.) This jacket can be designed to be completely flexible and gas tight. It would be used in conjunction with a tank enclosed by a shroud capable of withstanding aerodynamic forces. In this case the insulation is compressed by atmospheric pressure, which disappears when high altitudes are reached. The compressed insulation, although suffering a considerable deterioration in insulating effectiveness, (see Figure 18) would reduce the boil-off losses to manageable proportions. A layer of foam next

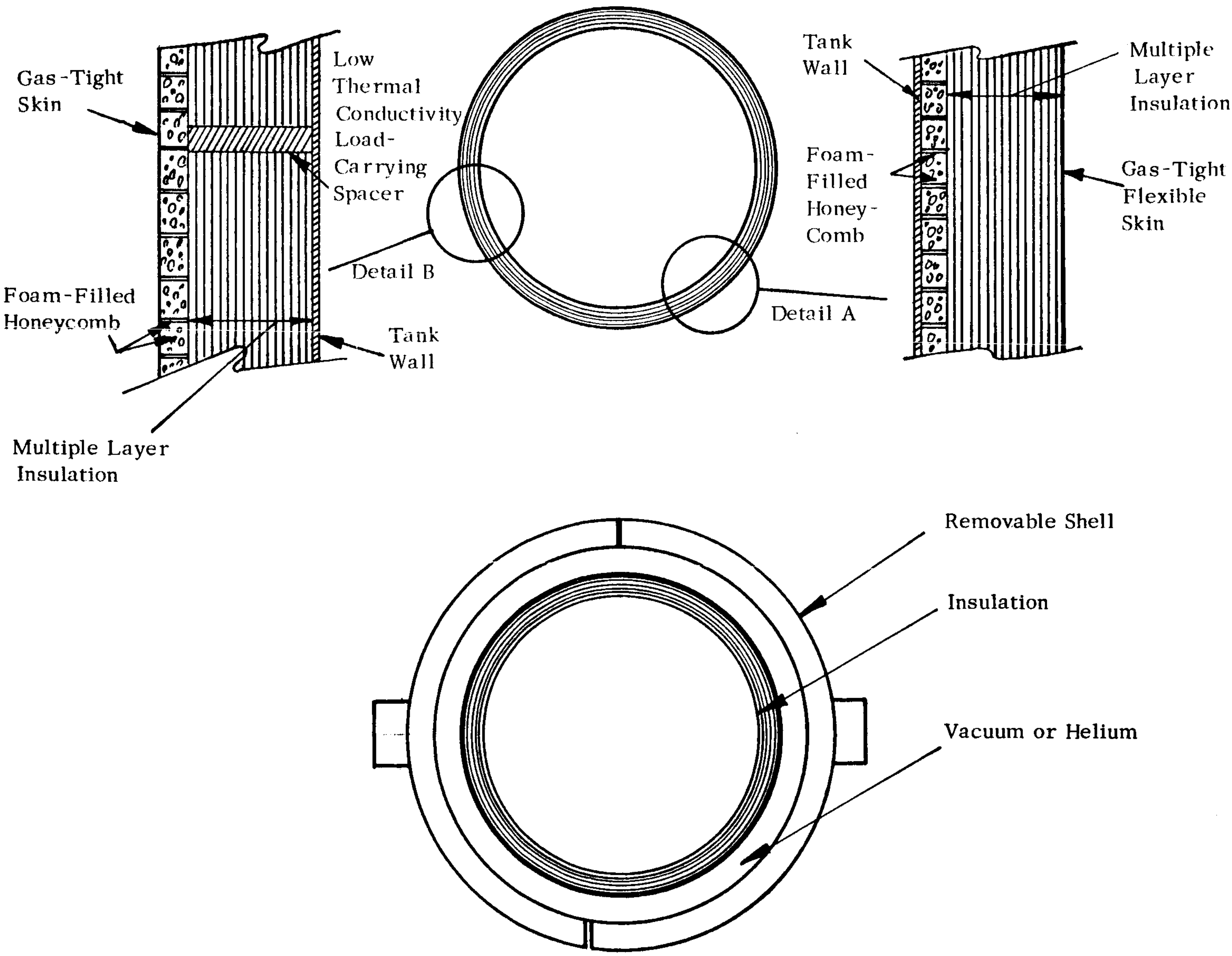

FIGURE 24 TECHNIQUES TO DECREASE BOIL-OFF LOSSES DURING GROUND STANDBY

to the tank could further reduce these losses and could provide a safety factor in case hydrogen should leak into the insulated space. At high altitudes, the insulation can regain almost all of its high insulating effectiveness if it is not subject to hysteresis effects.

Although this scheme is an attractive one, care has to be taken to achieve tight seals at the many projections caused by supports and piping attachments.

b. An alternative approach to evacuating insulation on the ground is to provide a light-weight rigid insulating cover capable of supporting limited compressive forces and permitting the insulated space to be filled with helium at a slight overpressure (See Figure 24b). The helium prevents air or water vapor from condensing and freezing in the insulation next to the liquid-hydrogen tank. The presence of liquid air, in addition to providing a LOX compatibility hazard, would cause the insulating effectiveness to deteriorate and, upon being heated, could lead to the formation of explosive gases during the boost phase. The cover would aid in preventing damage to the insulation and would be capable of withstanding aerodynamic forces and boost heating conditions. In addition, the multilayer insulation would not be exposed to mechanical loads because any applied loads could be transmitted to the tank wall by load-carrying, low-conductivity spacers.

c. The alternative to having a flexible shell is an arrangement of clam shells which make a tight seal with the tank. (See Figure 24c.) At the time of the launch, the vacuum is broken and the clam shells are removed. Because of the large size of most tanks, the need to have connections made to the tank during the ground standby period, and the complexity involved in arranging for removable clam shells, this approach is not recommended.

2. Boost Phase

Depending upon the trajectory, the aerodynamic heating during the boost phase may reach as high as 800° F on the sides of the vehicle. Although this heating effect occurs only for a short time, it can have the following consequences:

a. The surface layers of the insulation system could be materially degraded because of lack of strength at the high temperatures and the action of aerodynamic forces during this period.

b. The heat applied could penetrate into the insulation system, depending upon the system's diffusivity and heat capacity, and thus be responsible for boil-off losses even after the vehicle has successfully withstood the boost phase.

It is, therefore, desirable to use materials which (a) can withstand such temperatures without deterioration and are capable of having a coating with a low α/ε applied to them which can effectively radiate to the surroundings or (b) can act as effective ablators without a sacrifice in strength properties.

The buffeting forces set up during the boost phase by aerodynamic loads may excite various oscillating frequencies in the tank. The action of these forces on the tank walls will have to be established so that the insulation system can be designed to withstand these forces without suffering major damage. It may be useful to arrange for a portion of the insulation system to be jettisoned after the buffeting has subsided, so as to minimize the weight penalty.

E. LEAK DETECTION TECHNIQUES

Liquid hydrogen, because of its low viscosities, is capable of flows through tiny cracks and imperfections. Because of the difficulties of predicting the location, size, and frequency of leaks, sufficient quantities of liquid hydrogen could escape to lead to: (a) failure of the tank structure, (b) failure of the thermal insulation, (c) intolerable deterioration of the insulating effectiveness of the multi-layer insulator, or (d) an explosion hazard.

The liquid hydrogen tank can be treated as if it were a vacuum-type vessel and it can be subjected to the tests which are common practice in vacuum techniques. Such a procedure can greatly enhance the reliability of the system and permit more accurate predictions of thermal protection system performance.[26]

Flaw and leak detection methods, such as ultra-sonic testing, x-ray radiography and helium mass spectrometry, are standard procedures in vacuum technology. The helium mass spectrometer, in particular, should be used to test components, sub-assemblies, and final assembly welds at various stages of the construction process to assure maximum reliability.

Cold leak testing of critical parts at either liquid nitrogen or liquid hydrogen temperatures is highly recommended because leaks which cannot be detected while the tank is warm tend to become apparent after the tank has been cooled.

Attention will have to be given to the leak detection procedure after the thermal protection system has been installed, so that any leaks which may occur during static-firing tests or final assembly at the launch site can be detected. The sensitivity of the helium mass spectrometer is sufficient to indicate whether thermal protection performance will be impaired by the presence of a leak. Careful consideration has to be given to methods of applying the helium mass spectrometer so as to check large areas of the tank surface in a reasonable time consistent with conditions prevailing during pre-flight checkout.

The design of the thermal protection system has to be such as to permit the location and repair of a leak after one has been indicated. If careful attention is not paid to leak detection, the high insulating effectiveness of multilayer insulations may be compromised to such a degree that the insulation can no longer perform its function.

VII. CONCLUSIONS

The developments carried out in the field of thermal protection systems using various material combinations suitable for multilayer insulations show sufficient progress to make their use for large cryogenic tanks for extended space missions attractive. Many of the missions now contemplated can be achieved only if the high insulation effectiveness measured in the laboratory can be maintained after the thermal protection system has been installed on a tank and subjected to the various phases of space flight.

Before thermal protection systems can be routinely applied to cryogenic tanks, further development is necessary. Based on work already in progress, system design can be more clearly established and the additional efforts required in the various branches of science and technology better defined to meet the objectives of the future space-flight missions.

ACKNOWLEDGMENTS

The author gratefully acknowledges the many valuable discussions and contributions of his associates, in particular Igor A. Black, Jacques M. Bonneville, Alfred E. Emslie, Arthur A. Fowle, and Norman M. Wiederhorn, and the wholehearted cooperation extended by Dr. J. R. Brun, of NASA, Lewis Research Laboratories, under whose direction portions of the work reported on are being carried out.

REFERENCES

1. A. A. Fowle, "Estimation of Weight Penalties Associated with Alternate Methods for Storing Cryogenic Propellants in Space," Contract No. NAS5-664, Report No. 63270-11-01, May 1962.

2. R. E. Knox, "Insulation Properties of Fluorocarbon Expanded Rigid Urethane Foam," ASHRAE Journal, Vol. 4, October 1962, p. 43.

3. M. Smoluchowski, "Sur la Conductibilite Calorifique des Corps Pulverises," Bull. Int. de L'Academie des Sciences de Cracovie, A., (1910), p. 129.

4. A. Everest, P. E. Glaser and A. E. Wechsler, "Thermal Conductivity of Non-Metallic Materials", Contract No. NAS 8-1567, George G. Marshall Space Flight Center, Summary Report, April 1962.

5. I. A. Black, A. A. Fowle and P. E. Glaser, "Development of High Efficiency Insulation," Adv. in Cryogenic Engineering, K. D. Timmerhaus (ed.), Vol. 5, Plenum Press, Inc., (New York, 1960), p. 181.

6. D. Cline and R. H. Kropschot, "Thermal Properties of Powder Insulations in the Temperature Range 300° to 4°K," in Radiative Transfer from Solid Materials. H. H. Blau and H. Fischer (ed.), Macmillan Co., (New York, 1962) p. 75.

7. P. Peterson, "The 'Heat-Tight Vessel' A Transport and Storage Vessel for Liquid Oxygen and Liquid Nitrogen, etc." University of Lund, Sweden, December 1951, Office of Naval Intelligence, Translation No. 1147.

8. R. H. Kropschot, J. E. Schrodt, M. M. Fulk and B. J. Hunter, "Multiple-Layer Insulation", Adv. in Cryogenic Engineering, K. D. Timmerhaus (ed.) Vol. 5, Plenum Press, Inc., (New York, 1960), p. 189.

9. M. P. Hnilicka, "Engineering Aspects of Heat Transfer in Multilayer Reflective Insulation and Performance of NRC Insulation," Ibid., p. 199.

10. P. M. Riede and D. I-J. Wang, "Characteristics and Applications of Some Superinsulations," Ibid, p. 209.

11. Contract No. NAS 5-664, "Liquid Propellant Losses during Space Flight", NAS, Lewis Research Laboratories.

12. G. B. Wilkes, Heat Insulation, John Wiley & Sons, (New York, 1950).

13. M. M. Fulk, "Evacuated Powder Insulation for Low Temperatures," Progress in Cryogenics, K. Mendelsohn (ed.) Academic Press, (New York, 1959).

14. ASTM Standards 1961, Part 5, "Standard Method for Thermal Conductivity of Materials by Means of the Guarded Hot Plate," c177-45, p. 953.

15. J. F. Haskins and J. Hertz, "Thermal Conductivity of Plastic Foams from -423° to 75°F, Adv. in Cryogenic Engineering, K. D. Timmerhaus (ed.) Vol. 7, Plenum Press, Inc., (New York, 1962), p. 353.

16. D. Cline and R. H. Kropschot, "An Electrically Controlled Guarded Flat Plate Calorimeter," Ibid, p. 534.

17. I. A. Black, A. A. Fowle and P. E. Glaser, "Single-plate Apparatus for Tests of Low-Temperature Thermal Conductivity," Proceedings, Tenth International Congress of Refrigeration, International Institute of Refrigeration, Paris, 1959.

18. C. M. Pelanne and C. B. Bradley, "A Rapid Heat-Flow Meter Thermal Conductivity Apparatus," Materials Research & Standards, Vol. 2, 1962, p. 549.

19. D. I-J. Wang, "Multiple Layer Insulations," Proceedings Aerodynamically Heated Structures, P. E. Glaser (ed.) Prentice-Hall, Inc., (Englewood Cliffs, N. J. 1962) p. 95.

20. A. G. Emslie, "Radiation Transfer by Closely Spaced Shields," Contract No. NAS 5-664, Report No. 63270-04-02, May 1961.

21. A. G. Emslie, "Gas Conduction Problem with Multi-Layered Radiation Shields," Contract No. NAS 5-664, Report No. 63270-04-01, April 1961.

22. G. R. Smolak, R. H. Knoll and L. E. Wallner, "Analysis of Thermal Protection Systems for Space-Vehicle Cryogenic - Propellant Tanks," NASA Technical Report R-130, 1962.

23. M. A. Dubs and L. I. Dana, "Superinsulation for the Large Scale Storage and Transport of Liquefied Gasses," Annexe 1961-5, Supplement au Bulletin, International Institute of Refrigeration, Paris 1962, p. 71.

24. J. M. Bonneville, "Conductive Heat Leaks at Discontinuities in Multilayer Insulation," Contract No. NAS 5-664, Report No. 63270-00-05, June, 1962.

25. R. P. Mikesell and R. B. Scott, " Heat Conduction Through Insulating Supports in Very Low Temperature, " J. of Research of the National Bureau of Standards, Vol. 57, Research Paper 2726, 1956, p. 371.

26. "Leak Detection Procedures for S-II Liquid Hydrogen Tank", Agreement No. MAFPS-2-3000, North American Aviation, Inc., Space and Information Systems Division, Downey, California.

27. D. B. Chelton and D. B. Mann, Cryogenic Data Book, National Bureau of Standards, May 1956, p. 99.

28. R. M. Christiansen, M. Hollingsworth, Jr., and H. N. Marsh, Jr., "Low Temperature Insulating Systems," Adv. in Cryogenic Engineering, K. D. Timmerhaus(ed.), Vol. 5, 1960, p. 176.