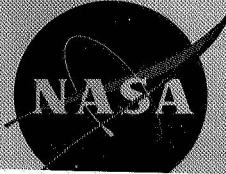


NASA CR-72368
ADL 69013-00



GPO PRICE \$ _____
CFSTI PRICE(S) \$ _____
Hard copy (HC) _____
Microfiche (MF) _____
ff 653 July 65

ADVANCED STUDIES ON MULTI-LAYER INSULATION SYSTEMS

prepared for

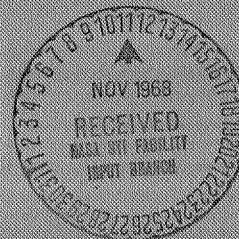
NATIONAL AERONAUTICS & SPACE ADMINISTRATION

CONTRACT NAS 3-7974

N 68-37852

FACILITY FORM 602

(ACCESSION NUMBER)	(THRU)
171	1
(PAGES)	(CODE)
CR-72368	B3
(NASA CR OR TMX OR AD NUMBER)	(CATEGORY)



Arthur D. Little, Inc.

NOTICE

This report was prepared as an account of Government sponsored work. Neither the United States, nor the National Aeronautics and Space Administration (NASA), nor any person acting on behalf of NASA:

- A.) Makes any warranty or representation, expressed or implied, with respect to the accuracy, completeness, or usefulness of the information contained in this report, or that the use of any information, apparatus, method, or process disclosed in this report may not infringe privately owned rights; or
- B.) Assumes any liabilities with respect to the use of, or for damages resulting from the use of any information, apparatus, method or process disclosed in this report.

As used above, "person acting on behalf of NASA" includes any employee or contractor of NASA, or employee of such contractor, to the extent that such employee or contractor of NASA, or employee of such contractor prepares, disseminates, or provides access to, any information pursuant to his employment or contract with NASA, or his employment with such contractor.

Requests for copies of this report should be referred to
National Aeronautics and Space Administration
Office of Scientific and Technical Information
Attention: AFSS-A
Washington, D.C. 20546

NASA CR-72368

ADL 69013-00

FINAL REPORT

ADVANCED STUDIES ON MULTILAYER
INSULATION SYSTEMS

prepared for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

31 January 1968

CONTRACT NAS3-7974

Technical Management
NASA Lewis Research Center
Cleveland, Ohio
Liquid Rocket Technology Branch
James Barber

Arthur D. Little, Inc.
Cambridge, Massachusetts

ADVANCED STUDIES ON
MULTILAYER INSULATION SYSTEMS

by
Arthur D. Little, Inc.

ABSTRACT

The heat flow contributed by a pipe penetration in a multilayer insulation was determined with analytical and experimental models. The penetration was a 17-inch-long, 6-inch-diameter stainless steel tube with a 0.025-inch wall thickness. This was connected to a LN_2 calorimeter tank with a 10-shield insulation system. Heat flows of about 15 and 46 Btu per hour were obtained with warm end temperatures of 540° and 750°R respectively. Heat flows that were about 35 per cent lower than these values were obtained by selectively lowering the emittance of the penetration interior. In a separate task, a heat flux of 0.33 Btu per hour was obtained on a 5-shield multilayer insulation, both before and after storage for one year.

Arthur D. Little, Inc.

TABLE OF CONTENTS

	<u>Page</u>
List of Figures	v - viii
List of Tables	ix - xi
I. SUMMARY	1
A. PURPOSE AND SCOPE	1
B. APPROACH	1
C. FINDINGS AND CONCLUSIONS	2
D. RECOMMENDATIONS	3
II. INTRODUCTION	4
III. EXPERIMENTAL FACILITIES	6
A. ARTHUR D. LITTLE, INC., TEST FACILITIES	6
B. NASA/PLUM BROOK STATION FACILITY	12
IV. THERMAL ANALYSIS OF PIPE PENETRATION	19
A. INTRODUCTION	19
B. GENERAL DISCUSSION	19
C. TECHNICAL DISCUSSION	23

TABLE OF CONTENTS (Continued)

	<u>Page</u>
V. EXPERIMENTAL SYSTEMS AND RESULTS - TASK I	46
A. INSULATION SYSTEM NO. 15	47
B. INSULATION SYSTEM NO. 16	65
C. INSULATION SYSTEM NO. 17	88
D. INSULATION SYSTEM NO. 18	103
VI. GOLD SHIELD SYSTEM RETEST - TASK II	114
A. INTRODUCTION	114
B. SYSTEM DESCRIPTION	114
C. TEST RESULTS AND DISCUSSION	116
D. CONCLUSIONS	120
REFERENCES	129
APPENDIX A DETERMINATION OF DIFFUSENESS OF SANDBLASTED METAL SURFACES	131
APPENDIX B EMISSOMETER DATA SUMMARY	147

LIST OF FIGURES

Figure No.		Page
1	CHAMBER AND CALORIMETER ASSEMBLY	13
2	FLOW SCHEMATIC, TANK INSULATION TEST FACILITY	14
3	CROSS-SECTION OF ADL MODEL -12 CALORIMETER	15
4	INSTRUMENTATION FOR ADL MODEL-12 CALORI- METER	16
5	EMISSOMETER	17
6	EMISSOMETER VACUUM SYSTEM	18
7	THERMAL PATHS IN PIPE PENETRATION	35
8	THERMAL CONDUCTIVITY OF 304 STAINLESS STEEL	36
9	ZONE SUBDIVISIONS IN THERMAL COMPUTER MODEL OF PIPE PENETRATION	37
10	PENETRATION WALL TEMPERATURE DISTRIBU- TION, EFFECT OF INSULATION SHIELDING FACTOR, μ	38
11	TEMPERATURE DISTRIBUTION IN WALL OF PIPE PENETRATION AS A FUNCTION OF INTERIOR EMITTANCE	39
12	TEMPERATURE DISTRIBUTION IN WALL OF PIPE PENETRATION AS A FUNCTION OF THERMAL CONDUCTIVITY OF WALL	40
13	COMPUTED LOCAL NET HEAT FLOWS FOR TEST IV-3C	41
14	EFFECT OF WALL CONDUCTANCE ON COMPUTED PENETRATION HEAT FLOW	42

LIST OF FIGURES (Continued)

Figure No.		Page
15	COMPUTED PENETRATION HEAT FLOW FOR VARYING AMOUNTS OF LOW-EMITTANCE TUBE WALL AT COLD END	43
16	COMPUTED PENETRATION HEAT FLOW FOR VARYING AMOUNTS OF LOW-EMITTANCE TUBE WALL AT WARM END	44
17	SCALED PENETRATION HEAT FLOWS FOR PIPE PENETRATIONS WITH $L/D = 2.8$	45
18	INSULATION SYSTEM 15	58
19	CALORIMETER TANK THERMOCOUPLE LOCA- TION	59
20	THERMAL CONDUCTIVITY APPARATUS, HEAT FLUX AS A FUNCTION OF COMPRESSIVE LOAD, SAMPLE 3093	60
21	INSULATION SYSTEM NO. 15, SHIELD TEMPERA- TURE DISTRIBUTION, TEST IV-2A	61
22	INSULATION SYSTEM NO. 15, SHIELD TEMPERA- TURE DISTRIBUTION, TEST IV-2B	62
23	INSULATION NO. 15, SHIELD TEMPERATURE DIS- TRIBUTION, TEST IV-2D	63
24	SHIELD TEMPERATURE DISTRIBUTION DURING TEST IV-2C	64
25	CUTOUT IN INSULATION SYSTEM NO. 15 FOR PENETRATION ATTACHMENT	77
26	MULTILAYER INSULATION 6" "A" CAP PIPE PENETRATION DETAIL	78

LIST OF FIGURES (Continued)

Figure No.		Page
27	UNINSULATED "A" CAP PIPE PENETRATION ATTACHED TO TANK CALORIMETER	79
28	INSULATED "A" CAP PIPE PENETRATION	80
29	INSULATED TANK CALORIMETER AND "A" CAP PIPE PENETRATION	81
30	SYSTEM NO. 16, PENETRATION HEAT FLOW VS END CAP TEMPERATURE, TEST SERIES IV-3	82
31	SYSTEM NO. 16, PENETRATION WALL TEM- PERATURES FOR TEST SERIES IV-3	83
32	SYSTEM NO. 16, PENETRATION SYSTEM TEMPERA- TURES, TEST IV-3C	84
33	SYSTEM NO. 16, PENETRATION SYSTEM TEM - PERATURES, TEST IV-3D	85
34	SYSTEM NO. 16, PENETRATION MLI, TEM- PERATURE DISTRIBUTION, TEST IV-3C	86
35	SYSTEM NO. 16, PENETRATION MLI, TEM- PERATURE DISTRIBUTION, TEST IV-3D	87
36	MULTILAYER INSULATION 6" "B" CAP PIPE PENETRATION DETAIL	98
37	PENETRATION "B" CAP AND CAP INSULATION SYSTEM	99
38	INSULATION SYSTEM NUMBER 17	100
39	INSULATION SYSTEM NUMBER 17 - PENETRA- TION WALL TEMPERATURES, TEST SERIES IV-4	101

LIST OF FIGURES (Continued)

Figure No.		Page
40	INSULATION SYSTEM NUMBER 17 - PENE- TRATION MLI TEMPERATURES, TEST IV-4G	102
41	PENETRATION WITH LOW - EMITTANCE ENDS	112
42	INSULATION SYSTEM NUMBER 18 - PENETRA- TION WALL TEMPERATURES, TEST SERIES IV-5	113
43	INSULATION SYSTEM NO. 14	126
44	INSULATION SYSTEM NUMBER 14 - SHIELD TEMPERATURE DISTRIBUTION, TEST SERIES IV-1	127
45	INSULATION SYSTEM NUMBER 14, SHIELD TEMPERATURE DISTRIBUTION, TEST SERIES III-7	128
A-1	REFLECTANCE COORDINATE SYSTEM	141
A-2	ALUMINUM, SANDBLASTED (SAMPLE NO. 372, SIDE A)	142
A-3	STAINLESS STEEL, SANDBLASTED (SAMPLE NO. 375, SIDE A)	142
A-4	COPPER, SANDBLASTED (SAMPLE NO. 376, SIDE A)	143
A-5	COPPER, GROUND FINISH (SAMPLE NO. 376, SIDE B)	143
A-6	SAND USED FOR BLASTING SAMPLES OF FIGURES A-2, A-3, AND A-4	144

LIST OF TABLES

Table No.		Page
1	HEAT FLOW SUMMARY FOR PARAMETRIC STUDY AND COMPARISON WITH TEST III-3A8	33
2	PENETRATION HEAT FLOW SUMMARY, EFFECT OF INTERIOR WALL EMITTANCE AND WALL THICKNESS	34
3	INSULATION SYSTEM NO. 15, SPECIFICATION	53
4	SPECIFICATION USED FOR PURCHASE OF ALUMINIZED MYLAR FOR SYSTEM NO. 15 RADIATION SHIELDS	54
5	EMITTANCE OF RADIATION SHIELDS IN INSULATION SYSTEM NO. 15	55
6	SYSTEM NO. 15, INSULATION SAMPLE 3093 EFFECT OF COMPRESSIVE LOADS ON HEAT FLOW	56
7	DATA SUMMARY, INSULATION SYSTEM NO. 15, TEST SERIES IV-2, CALORIMETER NO. 1	57
8	DATA SUMMARY, INSULATION SYSTEM NO. 16, TEST SERIES IV-3, CALORIMETER NO. 1	72
9	TEMPERATURE DISTRIBUTION, INSULATION SYSTEM NO. 16, TEST SERIES IV-3	73
10	SYSTEM NO. 16, COMPARISON OF COMPUTED AND EXPERIMENTAL RESULTS	74
11a	MATHEMATICAL MODEL RESULTS, INSULATION SYSTEM NO. 16	75
11b	MATHEMATICAL MODEL RESULTS, INSULATION SYSTEM NO. 16	76
12	DATA SUMMARY, INSULATION SYSTEM NO. 17, TEST SERIES IV-4, CALORIMETER NO. 1	93

LIST OF TABLES (Continued)

Table No.		Page
13	TEMPERATURE DISTRIBUTION, SYSTEM NO. 17, TEST SERIES IV-4	94
14a	MATHEMATICAL MODEL RESULTS, INSULATION SYSTEM NO. 17	95
14b	MATHEMATICAL MODEL RESULTS, INSULATION SYSTEM NO. 17	96
15	SYSTEM NO. 17, COMPARISON OF COMPUTED AND EXPERIMENTAL RESULTS	97
16	DATA SUMMARY, INSULATION SYSTEM NO. 18, TEST SERIES IV-5, CALORIMETER NO. 1	107
17	TEMPERATURE DISTRIBUTION, INSULATION SYSTEM NO. 18, TEST SERIES IV-5	108
18a	MATHEMATICAL MODEL RESULTS, INSULATION SYSTEM NO. 18	109
18b	MATHEMATICAL MODEL RESULTS, INSULATION SYSTEM NO. 18	110
19	SYSTEM NO. 18, COMPARISON OF PREDICTED AND EXPERIMENTAL RESULTS	111
20	INSULATION SYSTEM NO. 14, SPECIFICATION	121
21	INSULATION SYSTEM NO. 14, SHIELD EMITTANCE DATA SUMMARY, GOLD-COATED 1/4-MIL POLY-ESTER FILM	122
22	DATA SUMMARY, INSULATION SYSTEM NO. 14, TEST SERIES IV-1, CALORIMETER NO. 2	123
23	CALCULATED RADIATION HEAT FLUXES FOR RUNS IV-1B AND IV-1C	124

LIST OF TABLES (Continued)

Table No.		Page
24	TESTS OF A MULTILAYER INSULATION WITH GOLD COATED RADIATION SHIELDS	125
A-1	ROUGHNESS MEASUREMENTS OF SANDBLASTED SAMPLES	139
A-2	SPECTRAL REFLECTANCE AND PERCENT OF REFLECTANCE WHICH IS DIFFUSE	140
B-1	EMISSION METER DATA SUMMARY	148 -152

ADVANCED STUDIES ON MULTILAYER INSULATION SYSTEMS

I. SUMMARY

A. PURPOSE AND SCOPE

This report describes the studies performed on multilayer insulation (MLI) systems by Arthur D. Little, Inc., for the NASA/Lewis Research Center under Contract NAS3-7974. The effort consists of two tasks: Task I deals with the measurement and prediction of the heat flow in a multilayer insulation with a pipe-type penetration; Task II deals with the determination of the thermal performance of an MLI system that was warehoused for one year.

B. APPROACH

Task I was a study of the influence of a pipe penetration 6 inches in diameter and 17 inches long on the heat flow through a 10-shield evacuated multilayer insulation. Two environmental temperature conditions, three free-end temperature conditions, and two internal surface emittance configurations were considered. The MLI was placed on a 4-foot-diameter tank calorimeter, and tests were performed in a 5-foot-diameter space-simulation chamber. The two temperature environments investigated were 140° and 540°R, while the free-end penetration temperatures investigated were 140°, 400°, 540°, and 750°R. The emittance of interior surfaces of the penetration were generally on the order of 0.35; in one group of tests, portions of the interior surface were given an emittance of 0.025.

The experimental program was performed in conjunction with an analytical program for the prediction of the penetration heat flow performance and wall temperature distributions; this led to the establishment of a mathematical model to simulate the conductive and radiative heat transport mechanisms. The results are compared with the results obtained with the experimental model.

Task II was performed to determine the stability of the thermal performance of a five-shield MLI after a one-year warehouse storage period. The MLI consisted of five double-gold-coated Mylar shields and six spacers, each consisting of two layers of silk netting. It was fabricated onto a 4-foot-diameter tank calorimeter, and a heat flux of 0.33 Btu/hr-ft² was measured under NASA/LeRC Contract NAS3-6283. The thermal performance of this system was again measured under the current contract after one year of storage. MLI thermal measurements were made at both the Arthur D. Little, Inc., and NASA/Plum Brook Station facilities. Both liquid nitrogen and liquid hydrogen were used in the tank calorimeter during the performance of these measurements.

C. FINDINGS AND CONCLUSIONS

Task I

- (1) The major portion of the heat transport from the warm end of the penetration to the cold end is by the mechanisms of wall conduction and internal radiation. The heat flow from the environment to the pipe penetration through the MLI is of secondary importance. Penetration warm-end temperatures of 540° and 750°R result in typical heat flows of 15 and 46 Btu per hour (4 and 14 watts) respectively. The larger of these is equivalent to about 1000 pounds of liquid hydrogen boil-off in a six-month period.
- (2) The mathematical model produced excellent predictions of the penetration wall temperatures for all conditions investigated experimentally. The heat-flow predictions were excellent for all penetration-end temperatures except 750°R. In the latter case, the predicted heat flows were up to 25% less than the measured heat flows.
- (3) The differences between the measured and predicted heat flows obtained for penetration-end temperatures of 750°R are due in part to the uncertainties associated with the conductance and interior surface emittance of the penetration wall. Also, volatilization of MLI spacer material components at 750°R are considered as contributing to the differences noted.
- (4) The selective reduction of the penetration interior wall emittances produced substantial reduction in the penetration heat flow. The placement of 0.025 emittances at the warm and cold ends of the penetration lowered the heat flow obtained at a 750°R warm-end temperature from 44 to 27 Btu per hour (13 to 8 watts).

Task II

The results of this experimental study demonstrate that the thermal performance of a reasonably protected MLI system is not affected by storage in a shop or warehouse environment for a one-year period.

D. RECOMMENDATIONS

Task I

A conventional piping connection was simulated in the current investigation. However, the measured penetration heat flows must be improved by a factor of 10 to be practical for an extended spacecraft mission. The current study has demonstrated the validity of the mathematical model and at the same time our ability to construct an experimental model which is verifiable. It is recommended that this model be used to explore the heat flow and temperature distributions in other geometric piping configurations of practical value, including pipes with low-conductance walls, low-emittance surfaces, radiation baffles, elbows, exterior shielding factor variations, and ranges of length-to-diameter ratio. The thrust of these studies would be to achieve the lowest possible heat flow values consistent with propellant line use and available construction materials.

It is also recommended that selected configurations be tested experimentally for further confirmation of the mathematical model and verification of the pertinent thermal scaling laws.

II. INTRODUCTION

Multilayer insulations have reached a stage of development where experienced application engineers can make reasonable predictions of their thermal performance. Considerable information about MLI properties and their response to thermal and other environments is now available to make their application feasible at their demonstrated thermal performance levels on uninterrupted surfaces.

However, in practical applications, multilayer insulation systems must provide for the penetration of plumbing lines, structural supports, and electrical leads which are sources of a significant fraction of the total heat leak to the stored propellants. To date the work in these latter areas for accurately predicting thermal performance has not progressed at the same rate as that of basic multilayers. For example, insulating techniques for pipe penetrations are at an early stage of development, and the uncertainties associated with reproducibility and predictability are yet large. Because of the interdependence of MLI and penetration, predictions of the total system performance acquire the uncertainties associated with the latter.

The current effort has been directed exclusively to the piping connections at cryogenic propellant tanks, specifically to a 6-inch-diameter and 17-inch-long pipe penetration constructed of 304 stainless steel with a 0.027 inch wall thickness. This pipe penetrated a 10-shield MLI and was itself insulated with a 10-shield system. From this experimental model and a mathematical computer model, penetration heat flows and wall temperature distributions were obtained. Comparisons of the two sets of data led to refinement of the computer model and an understanding of the heat transport mechanisms.

The isolation of a pipe penetration to determine the added heat flow in an MLI due to the penetration's presence is experimentally difficult. For one thing, the heat flows are very small, ranging from close to zero to a few Btu per hour. Also, the techniques of integrating the penetration and tank multilayer insulations to produce the lowest degradation are still undergoing development. In addition, the measurements of the significant experimental parameters, such as the surface emittance and nature of the energy reflected from a surface, are costly to obtain and not yet highly accurate. There are added difficulties in establishing and controlling these parameters in the test article.

A parametric study was performed by Bonneville,⁽¹⁾ in which he investigated the effect of the pipe wall conductance and interior emittances on the heat flow for pipe length-to-diameter ratios of 3 and 9. On the basis of his findings, an experimental program⁽²⁾ was performed with a test article having a L/D of 3 placed in a multilayer insulation. L/D ratios of approximately 18 were investigated experimentally and analytically by Price et al.⁽³⁾ Experimental pipe

temperature distributions were in good agreement with predicted values. In the same study the use of low-conductance wall sections was also investigated. Sterbentz⁽⁴⁾ has conducted analytical and experimental studies of the pipe penetration problem in the design and development of a thermal protection system for a cryogenic spacecraft propulsion module.

The intent of the current work was to focus attention on a single pipe penetration configuration and to explore the heat transport mechanisms thoroughly with an experimental and a mathematical model. Associated with this was the further development of the insulating techniques for integrating the penetration and tank MLI's; isolating the penetration heat flow in the calorimeter system would have been impossible without a developed MLI system. The focus on a single system has permitted the verification of the mathematical model, which can now be widely used in parametric and design studies. This does not eliminate the need for experimental verification of the scaling laws or final design type tests, but it should reduce the extent of experimental work necessary in the future.

III. EXPERIMENTAL FACILITIES

The experimental program was conducted mainly at the Arthur D. Little, Inc. (ADL), facilities located at Cambridge, Massachusetts. These facilities were limited to the use of liquid nitrogen as the testing fluid. Tests with liquid hydrogen were conducted at the NASA/Plum Brook Station, J-3 facility.

A. ARTHUR D. LITTLE, INC., TEST FACILITIES

The thermal properties of the multilayer insulations studied in this program were measured through use of three kinds of apparatus. An emissometer was used to measure the total hemispherical emittance of shield materials. The heat flow performance of the multilayer insulations under compressive loads was measured with the use of a guarded flat plate thermal conductivity apparatus. The installed performance of MLI and penetration heat flows were measured with a 4-foot-diameter tank calorimeter.

1. Tank Calorimeter System

Test insulations were mounted on a tank supported by a guarded vent. This tank was used as a calorimeter. The heat flow through the insulation was determined from the rate at which liquid nitrogen boiled off from the calorimeter. The tank calorimeter was located in a chamber which simulated the vacuum and temperature environments required in the insulation studies. The environmental chamber and tank calorimeter are shown in Figure 1.

a. Tank Calorimeter

The tank calorimeter has a diameter of 48 inches, a depth of 27 inches, and a surface area of almost 40 ft². Its shape is that of a short vertical cylinder with torispherical top and bottom ends. It is made of high-thermal-conductivity copper, 1/4-inch thick to minimize temperature stratification in the tank wall and in the liquid -- i.e., to promote a uniform surface temperature that is independent of liquid level, so that the insulation heat flow is completely converted into vaporized liquid and/or rise in the temperature of the stored liquid.

The tank is suspended from a 24-inch-diameter mounting flange by a 5-inch, schedule 5 stainless steel pipe. The tank is thermally isolated through use of a cold guard placed at the center of this support. Liquid nitrogen circulated through this guard intercepts any heat conducted downward from the support flange, radiated downward in the interior of the support, or radiated to the outside of the support between the guard and the tank.

The tank calorimeter has provisions for attachment of a pipe section at the center of the bottom head. This attachment is used in the current program to study simulated pipe penetrations in a multilayer insulation system. The adapter piece is 6 inches in diameter by 17 inches long and has a wall section 0.027 inch thick. The free end of the pipe penetration is removable for access to the interior surfaces. This end cap also contains a heater which is used to establish the required end temperature during penetration tests.

b. Space Simulation Chamber

During the space simulation tests, the calorimeter and its insulation system were installed in the 5-foot-diameter, 7.5-foot-deep vacuum chamber in the position shown in Figure 1. Normal operating vacuum achieved during the tests was below 5.0×10^{-6} torr.

The chamber contains two baffles that completely enclose the calorimeter tank. The baffles are fabricated of copper and contain integrally mounted coils through which is circulated the temperature control fluid. The baffles can be maintained either at 140°R or 540°R by the circulation of liquid nitrogen or water respectively from external supply systems.

The chamber and its associated systems are shown in Figure 2. The calorimeter is filled from a 500-gallon liquid nitrogen supply tank. This supply tank also provides the feed for the cold guard and chamber baffles when they are operated at liquid nitrogen temperatures; this is accomplished through use of a gravity tank which is elevated above the chamber. The 4-inch column that supports the tank also serves as feed line for the baffles and cold guard. The baffles vent to the gravity tank ullage space, while the cold guard vents to atmosphere through a heat exchanger. The baffles can also be connected to the local water system when they are to be maintained at near room temperature.

The chamber is evacuated through use of a 6-inch oil diffusion pump and a mechanical fore pump. This system can achieve 10^{-6} torr pressure with a clean and out-gassed system in less than one day. A period of up to several days has been required to achieve this vacuum with new insulation systems. The insulation systems were generally tested with chamber operating vacuums in the range of 10^{-5} to 10^{-6} torr. Occasionally, on long-run tests, vacuums below 10^{-6} torr were achieved.

The boil-off gases vented from the calorimeter were brought to room temperature in an air-warmed heat exchanger. They were measured volumetrically in wet-type, positive-displacement meters and vented to the atmosphere.

c. Instrumentation and Controls

The primary measurements made periodically during each test consisted of the following:

- (1) Calorimeter vent gas flow, temperature, and pressure;
- (2) Vacuum chamber pressure;
- (3) Local barometer and calorimeter pressure; and
- (4) Temperatures of selected foils in the insulation system, the chamber baffles, the calorimeter cold guard, the local environment, penetration wall, and others as required.

At both the Arthur D. Little, and Plum Brook facilities, all critical system operating temperatures and the barometer data were recorded on strip charts. In addition, these and all the other data were manually recorded at periodic intervals.

Temperatures in the calorimeter insulation and throughout the test facility were measured with copper-constantan thermocouples. These were made from selected wire, usually of 40-gauge size. In all cases the reference junctions were immersed in liquid nitrogen. Trend-type temperature points were continuously recorded. Data from each thermocouple were measured with a dc amplifier and digital voltmeter and recorded manually.

The calorimeter boil-off gas flow measurements were made with a wet-test meter and a dry-type meter in series to obtain the volumetric flow. The temperature and pressure of the metered stream were measured to establish the gas density. A water bubbling chamber was installed upstream of the wet-test meter to saturate the stream with water vapor prior to measurement.

An elevated liquid nitrogen gravity tank provides constant-pressure feed to keep the cold guard at liquid nitrogen temperature. When the chamber baffles are cooled with liquid nitrogen, this same system provides the necessary temperature control. When the baffles are operated at 540°R through the use of a water recirculation system, the temperature of the water is held constant by a thermostatically controlled water supply. Cooling is required because the recirculating pump introduces heat to the baffle water stream.

When the pipe penetration was being tested, the free end was usually maintained at temperatures between 400°R and 750°R. An electric heater was used to establish and maintain the desired temperature level. The power to the

heater was provided from a regulated power supply. No attempt was made to use this supply as a temperature controller; instead, the power level was manually adjusted until the equilibrium temperature was at the desired level. At equilibrium, the power input was maintained at a constant value.

d. Heat Flow Measurement

The thermal performance of the insulation systems had been measured over intervals of approximately 24 to 100 hours. During the measurement interval, the ullage pressure of the calorimeter varied directly as the barometric pressure, i.e., the calorimeter vented to the atmosphere through a low-pressure-drop system, and no attempt was made to control the ullage pressure.

The passage of heat through the insulation causes energy to be stored in the calorimeter system and vaporized liquid to be vented from the calorimeter. When no changes in the barometric pressure occur in the measuring interval, the energy transmitted through the insulation is evidenced primarily as vaporized liquid. An almost negligible amount of energy is stored in the calorimeter ullage.

When barometric pressure changes do occur during the interval, the system temperature increases or decreases directly with this pressure. An increase in system temperature causes heat to be stored in the metal of the tank calorimeter and in the liquid and gas masses. A decrease in temperature produces a heat release. Barometric changes also resulted in changes of other system energies.

A thermodynamic analysis⁽⁵⁾ of the calorimeter system indicates that for the conditions under which our measurements were made, only two effects need be considered when evaluating the insulation heat flow. The first and most significant factor is the mass of the vented gas; the second factor is the liquid mass heat storage effect.

The total heat rate is computed from the total vaporized liquid vented during the data interval and the latent heat of vaporization of the cryogen. The gas mass is computed from the integrated volume measurement of the calorimeter vent gas stream and its temperature and pressure at the metering point.

This measured heat flow is corrected for heat storage or heat release in the calorimeter in the following manner: At the beginning of each test the calorimeter is filled with liquid nitrogen, which provides a known liquid mass. Vaporized liquid vented from the calorimeter is continuously measured in a volumetric totalizing meter. The volumetric measurement provides a basis for establishing the vented gas mass and the mass of the liquid remaining in the system. This latter information is used to determine the average liquid mass present in the calorimeter tank during any data interval.

The saturation temperature of the bulk liquid at the end of the data interval was compared with that at the beginning. From the change in the bulk temperature and the specific heat of the liquid at saturation, the total heat energy stored or released during the data interval was computed and converted into an hourly rate. Any heat stored during the data interval was added to the heat flow determined from the calorimeter boil-off; a release of heat during the data interval was subtracted.

2. ADL Model-12 Calorimeter

To measure the heat flow passing through a multilayer insulation and to study the effect of different variables on the heat flow, we have designed and manufactured, under Contract NAS5-664, the flat-plate calorimeter shown schematically in Figure 3. Its principal components are a double-guarded cold plate, a warm plate, a sample chamber, and a vacuum jacket. A circulating fluid maintains the warm plate at the desired uniform temperature. A boiling cryogen in the measuring vessel and guard vessel maintains the cold plate at a uniform temperature. The enclosing vacuum jacket permits the apparatus and the test sample to be evacuated to pressures as low as 10^{-6} torr. The quantity of heat flowing through a 6-inch-diameter central section of a 12-inch-diameter test sample is determined by measuring the boil-off rate of the cryogen contained in the measuring vessel. The apparatus and related instrumentation (Figure 4) are described in detail in the final report under Contract NASw-615 and in the operating manual supplied under Contract NAS3-6283.

3. Emissometer

The heat flux of a multilayer insulation in the radiation heat transfer mode is directly dependent upon the surface emittance of its shields. The measurement of shield emittance is necessary for establishing the theoretical performance of any multilayer insulation system. An emissometer instrument⁽²⁾ developed by ADL and sponsored by NASA/Lewis Research Center under Contract NAS3-4181 was used to measure the total hemispherical emittance of shield materials studied in the current program.

The principal elements of the emissometer are contained in an evacuated steel chamber approximately 5 inches in diameter and 11 inches long. (See Figure 5.) A thin, circular, blackened metal disc, 2.0 inches in diameter, is placed closely adjacent and parallel to a circular sample piece, 2.5 inches in diameter, in an evacuated space. The back side of the blackened receiver disc is surrounded by a black cavity whose temperature is lower than that of the sample. The disc receives heat from the sample on one side and yields it to the black cavity on the other side, principally by radiative heat transfer. The geometry is such that the heat transfer between the sample and the receiver disc is essentially like that between two infinite parallel planes. Radiative transfer between the back side of the disc and the

black cavity is essentially that between two black bodies. Hence, a heat balance on the disc yields the equation:

$$e_s \sigma (T_s^4 - T_d^4) = \sigma (T_d^4 - T_c^4)$$

or

$$e_s = \frac{T_d^4 - T_c^4}{T_s^4 - T_d^4}$$

Thus, if the sample temperature, cavity temperature, and disc temperature are measured, the emissivity of the sample can be determined. In this apparatus, the sample temperature is maintained constant at slightly above room temperature. The cavity is maintained at the temperature of liquid nitrogen (140°R). The receiver disc temperature varies in relation to the sample emittance and is measured with a calibrated copper-constantan thermocouple.

In a typical test the sample is mounted on the sample holder with double-sided adhesive tape. The holder is placed in the apparatus, and the vacuum system (Figure 6) is then used to evacuate to an operating pressure of 10^{-6} torr. The cylindrical cavity and receiver disc mounting ring are cooled with liquid nitrogen. At temperature equilibrium the receiver disc and sample holder temperature are measured. These temperatures are used in the preceding equation for computing the emittance of the sample surface.

An error analysis indicates fixed and random errors of approximately 5% at the levels of emittance measured in this study. A calibration of the instrument indicated an instrument error of approximately 7%, indicating qualitative agreement with the results of the analysis.

The coating thickness of each vacuum-metallized emissometer sample was determined by measuring the electrical resistance of the coating. This method is widely used by commercial vacuum metallizers for the approximate determination of film thickness of production material. The accepted unit of measurement is the ohm per square. The resistance between the ends of a right prism is proportional to length and inversely proportional to its cross sectional area in accordance with the following relation:

$$R = \rho \frac{l}{A} = \frac{\rho l}{wt}$$

where

R = the electrical resistance;

ρ = the resistivity of the material;

l = the distance between the electrodes; and

A = the cross sectional area, product of w and t where w is the sample width and t the coating thickness.

When the length is equal to the width, the resistance or ohms per square is proportional to ρ/t . In this study all resistance measurements were made on a parallel bar device with a 2-inch measuring square. The resistance of the film between the two electrodes was measured with a resistance bridge and then converted to a thickness value using the above relation.

B. NASA/PLUM BROOK STATION FACILITY

The J-3 space simulation facilities at the NASA/Plum Brook Station in Sandusky, Ohio, were used to conduct tests with liquid hydrogen on the calorimeter containing the gold shields (System 14).

The vacuum chamber in the J-3 facility has approximately the same dimensions as the Arthur D. Little, Inc. chamber. Also, the upper baffle is similar. However, there is no baffle in the lower portion of the chamber; instead, the chamber is jacketed in this area. Both the baffle and jacket can be supplied independently with either liquid nitrogen or water for temperature control of the surfaces facing the calorimeter insulations.

The J-3 facility is serviced by a portable liquid hydrogen dewar and a stationary liquid nitrogen dewar, each having about 7000 gallons capacity. They are connected to the facility through a common piping system that supplies the required cryogen to the calorimeter tank and cold guard coil and to the upper baffle and lower jacket on the inside of the vacuum chamber. The latter are also served with cold or heated water so that the temperature surrounding the calorimeter may be held at 140°R or from 500°R to above 560°R. The vacuum system consists of a 20-inch CVC diffusion pump, equipped with a liquid nitrogen cold trap controlled by a Johns & Frane LN₂ flow controller and a Model 412H Stokes mechanical forepump having 300 cfm capacity; it is capable of attaining less than 10^{-7} torr absolute pressure.

The same experimental data, such as boil-off gas rate and shield temperatures, were obtained at the J-3 facility as with the Arthur D. Little, Inc. facility described in the previous section. However, the test in the J-3 facility was conducted from a remote control center, due to the hazards associated with the use of liquid hydrogen; the data were therefore transmitted and recorded in a different but equivalent manner.

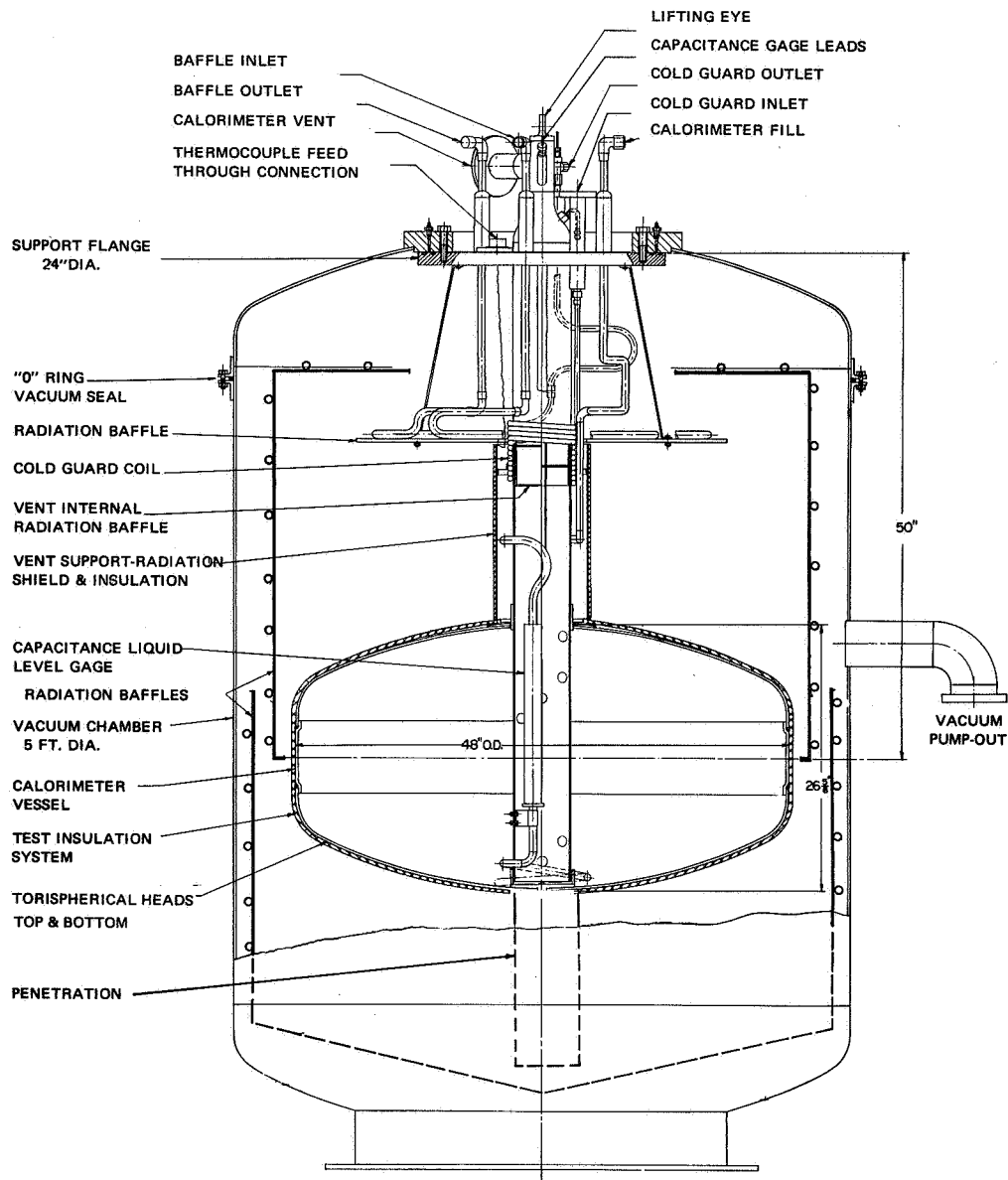


FIGURE 1 CHAMBER AND CALORIMETER ASSEMBLY

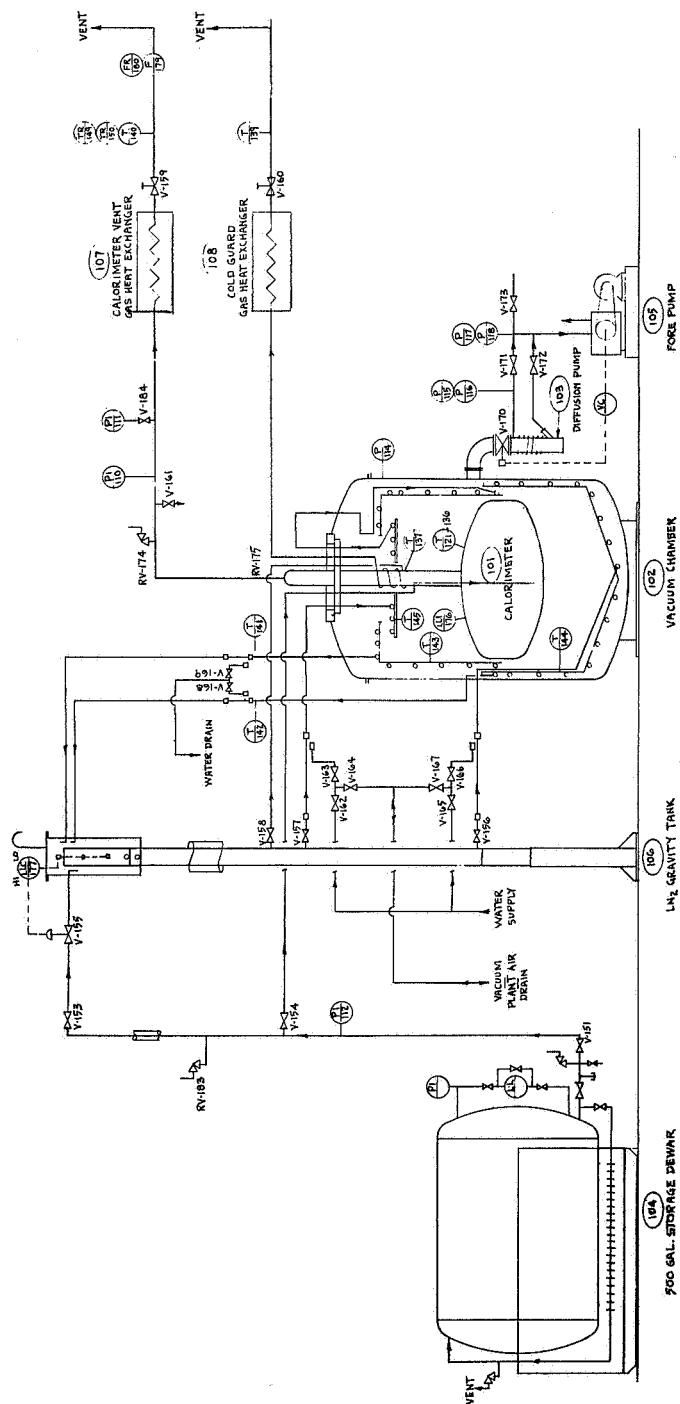


FIGURE 2 FLOW SCHEMATIC, TANK INSULATION TEST FACILITY

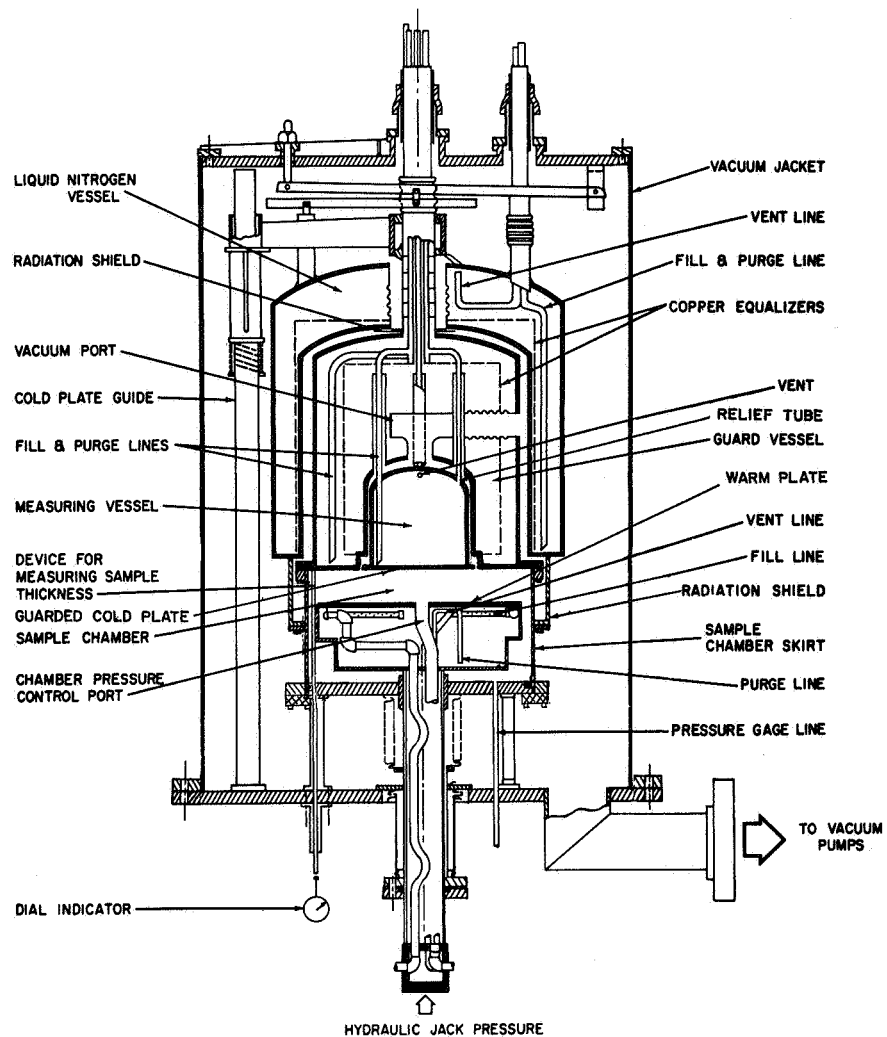


FIGURE 3 CROSS-SECTION OF ADL MODEL-12 CALORIMETER

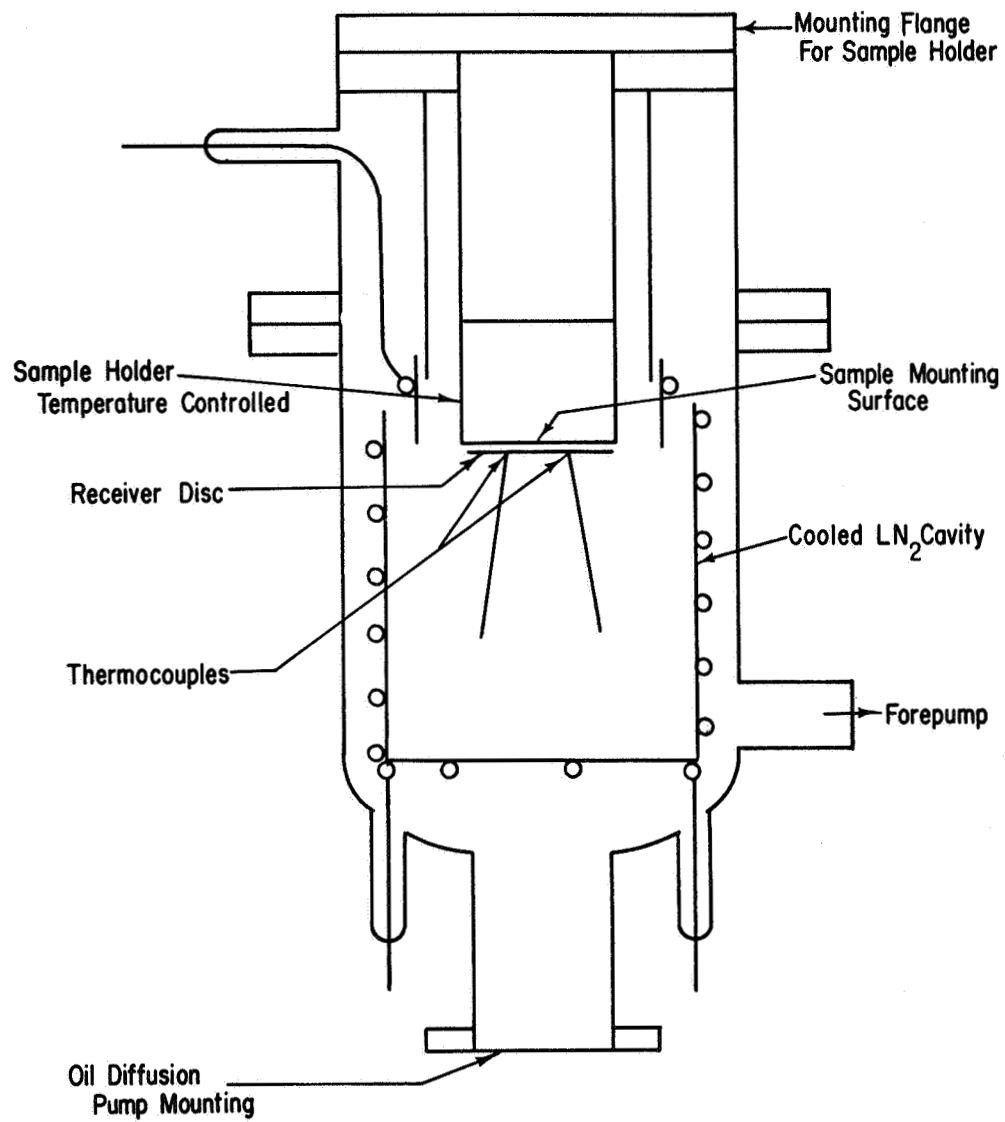
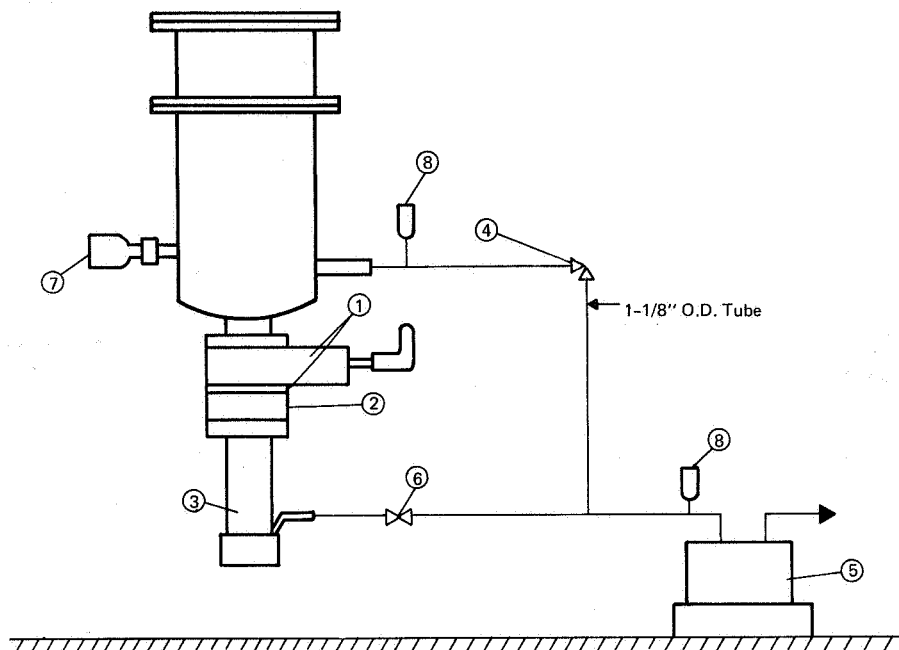


FIGURE 5 EMISSOMETER



- 1 High Vacuum Gate Valve
- 2 Chevron Baffle
- 3 2" Diffusion Pump
- 4 Roughing Valve – Veeco Bellows Angle Type
- 5 5 CFM Roughing and Fore Pump
- 6 Foreline Valve
- 7 Ion Pressure Gage
- 8 Thermocouple Pressure Gage

FIGURE 6 EMISSOMETER VACUUM SYSTEM

IV. THERMAL ANALYSIS OF PIPE PENETRATION

A. INTRODUCTION

There are three main reasons for developing a thermal-computer model of a physical piece of hardware:

- (1) It is a means of examining a large number of variations in a proposed thermal design.
- (2) An understanding of the heat transport mechanisms in the real hardware can be developed by comparing the predicted and measured values of temperatures and heat flows.
- (3) Once a good correspondence between the piece of hardware and its computer model has been established, a range of thermal boundary conditions can be examined to predict the thermal response of the hardware.

In the present study, one use of the thermal-computer model of the pipe penetration was to examine the significant parameters affecting the heat leak to the nitrogen-filled tank due to the pipe, and to predict how variations in each parameter would quantitatively affect the overall heat leak. This permitted an evaluation of how large a reduction in the measured heat leaks could be achieved by practical design changes. The computer model was also used to predict the temperature distribution and local heat flows for each test condition and to predict power settings for the heater so as to decrease the time required to reach thermal equilibrium for a new test condition.

B. GENERAL DISCUSSION

1. Thermal Description of Pipe Penetration

A physical description of the pipe penetrations used in the experimental models is given in Section V. The present discussion will therefore deal only with the thermal aspects of the pipe penetration. Figure 7 is a pictorial representation of the thermal paths for the heat flow to the cold end. The computed radiative heat exchange between the pipe and its external environment through the MLI blanket was small in all test cases; the dominant source of the heat leak was the electrical power dissipated in the end cap. The heater power leaves the end cap by 1) radiation from both the interior and exterior surfaces and 2) conduction into the tube wall. The energy, Q , emitted per unit area by the heater to the interior of the pipe, is directly proportional to the emittance of the heater surface, ϵ , and to the fourth power of its temperature; i.e., $Q = \epsilon \sigma T^4$. The energy conducted into the tube wall is proportional to the local thermal conductivity

of the pipe wall, the wall cross-sectional area, and the temperature gradient in the tube wall at the heated end.

2. Thermal Computer Model

If we consider steady-state conditions only, the thermal analysis effort can be described in simple terms as a mathematical simulation of the thermal interactions of the various parts of the pipe which permits one to compute the fraction of the power entering the system that is finally transmitted to the cryogen-filled tank. The computer model simulates the thermal characteristics of physical hardware by considering the pipe as being subdivided into a finite number of discrete elements, commonly called nodes or zones. An energy balance equation is written for each element, with the driving potentials being the temperature differences (or T^* for radiation) between the individual elements. The thermal coupling between a particular pair of elements is formed by the conductive and/or radiative heat flow paths. The thermal coupling for radiative energy exchange between two nodes is in the form of a view area (A_F), which can be considered as that part of the total emitting area (ϵA) of an element that is exchanging radiative energy with another. In this analysis, the view area matrix was symmetric; i.e., the view area coupling from zone n to zone $(n + 1)$ equaled the reverse radiative coupling from zone $(n + 1)$ to zone n . This reciprocity is valid when: (1) the angular distributions of the radiant fluxes leaving the participating surfaces are diffuse and (2) the magnitudes of the radiant fluxes do not vary across the respective surfaces. For these two conditions, the angle factors depend only on the geometrical orientation of the surfaces.

Because the interior surfaces of the pipe are not perfectly black, the energy diffusely emitted by zone n , and finally absorbed by zone $(n + 1)$, travels by a direct path as well as by a parallel path having one or more intermediate reflections. To simulate non-black surfaces in the thermal-computer model, an additional radiosity node, or fictive shield, was added to each interior surface.⁽⁶⁾ The exterior surface of each fictive shield was coupled to the true interior surface of the pipe by a radiative coupling $(A_F)_{fs}$ which is a function of the reflectance and emittance of the true surface.

$$(A_F)_{fs} = \frac{\epsilon}{\rho} A$$

where

ϵ = emittance of true surface

ρ = reflectance of true surface

and

$\rho = 1 - \epsilon$ for the assumed gray surfaces

The interior, facing surfaces of the fictive shields are considered to be perfectly black so that they exchange energy with one another only by direct radiation (i.e., no reflections). The radiative coupling between a node on the penetration and its fictive shield is a mathematically derived coupling whose value becomes infinite when the surface emittance reaches unity.

The conductive couplings between adjacent zones were determined by calculating the kA/l product for each zone and then using the weighting factors defined by the "Method of Zones."⁽⁷⁾ The latter is a technique which approximates the temperature distribution across each zone by a parabolic curve, i.e.:

$$T(x) = A + Bx + Cx^2$$

As Reference 6 contains a detailed explanation of the technique of the Method of Zones, the details will be omitted here.

3. Limitations of the Thermal Model

The basic, limiting assumptions in the thermal-computer model of the pipe penetration are as follows:

- (1) Each surface is treated as a gray, diffuse emitter and reflector whose emittance and absorptance are a function of the temperature of the surface.
- (2) The thermal conductivity of the stainless steel tube wall as a function of temperature was taken from published data. (See Figure 8.)
- (3) The emittances of the interior surfaces were derived from measurements made on small samples and did not include any degradation effects.
- (4) Joint interfacial resistances were assumed negligible.
- (5) Only radial, radiative heat transport was assumed through the MLI.

The assumption that all surfaces in the pipe were diffuse reflectors was effectively verified by goniometer measurements at ADL during the course of this study (see Appendix A). The assumption of diffuse emitters is a good approximation to an extremely complicated surface property. To use a model with other than diffuse emitters would first require an accurate determination of

the directional emission characteristics of each surface, followed by an evaluation of the radiation view-area couplings based on the non-Lambertian emission characteristics. This would represent a significant additional analytical and experimental effort and was not undertaken in this program.

It can be generally concluded, however, that for the conditions in which 3M black velvet paint was applied to the interior of the pipe, the computed heat flow to the cold end would have increased had non-gray surface properties been used in the thermal-computer model. This is due to the fact that the thermal radiation incident on the cold-end surfaces originated from surfaces at significantly higher temperatures, and the absorptance of 3M paint is dependent on the wavelength of the incident energy (increasing for shorter wavelengths). Parametric computer runs in which the absorptance (and emittance) of the cold-end surfaces was varied from 0.85 to 0.94 showed a total increase in the heat flow to the cold end of 0.2 Btu per hour (0.06 watt, 1.1%) for a 540°R end-cap temperature. (See Runs No. 7 and 9 in Table 1.)

The published data for the thermal conductivity of the stainless steel tube was used as presented in the literature (see Figure 8). Thermal conductivity was a significant parameter in the pipe penetration because of the importance of the "fin effect" in the tube wall at both the hot and cold end plates. The experimental determination of the thermal conductivity of the tube wall as a function of temperature was not in the scope of this program; however, an uncertainty analysis was made to determine the significance of variations in this parameter. (Over the range of temperatures measured in this investigation, the variation of the thermal conductivity of 304 stainless steel with temperature was approximately 0.096 Btu/hr ft °R per °R (0.0003 watt/cm-°K/°K).

A small sample of each surface material was given the same sand-blast treatment as used on the full pipe penetration, and the total hemispherical emittance of the sample was measured. These measured values were used in the computer model.

During the course of the experimental testing, the surface finish on the test hardware showed signs of degradation as well as other departures from the ideal (e.g., solvent residue, silver solder at the joints, and a gray, loose powder on the sandblasted stainless surface). The exact effect of these surface imperfections is not easily determined and imparts some uncertainty to the analysis. It can be generally stated that any change in a surface finish which tends to increase the surface emittance would also tend to increase the heat leak to the cold end.

The joint resistances between tube wall and both end plates (see Figure 25) and across the threaded interface at the free end were assumed small. This assumption was considered justified because: 1) the silver solder material

in the brazed joints would decrease any interfacial resistance and 2) the soft material (aluminum and copper) used in the threaded joints would also tend to produce a joint of low contact resistance.

The assumption of only radial, radiative heat transport in the MLI was necessitated by a lack of any justifiable model to predict a radiative or conductive heat transfer along the layers of the insulation. (Anomalies in the experimental data suggested a conductive heat transport mechanism when the end-cap temperature was near 750°R; see Section V-B for further discussion.)

C. TECHNICAL DISCUSSION

For the cap "A" configuration (Figure 25) the pipe penetration, with its associated fictive shields, was subdivided into 28 zones as shown in Figure 9. Counting the additional joining equations used by the Method of Zones, 34 temperatures were computed for each set of thermal boundary conditions. There were three fixed temperature zones namely: 1) the LN₂-filled tank, 2) the hot-end plate, and 3) the environmental shroud. The tube wall was divided into zones of smaller areas near the cold end so that the larger gradients that occur there could be predicted more accurately.

An ADL view-area program was used to compute the black-body radiation view area couplings. Calculation of the conduction couplings between adjacent zones was based on the published data for the thermal conductivity of stainless steel and a measured mean value of 0.027 inch for the tube wall thickness.

1. Parametric Studies

In the initial phase of this analysis (before the current series of tests had begun), the measured temperatures and heat flows from Test III-3A8,* performed under Contract NAS3-6283,⁽²⁾ were used as a guideline to determine the significance of the following input parameters: 1) effectiveness of the MLI, 2) wall emittance, and 3) thermal conductivity of stainless steel. Figures 10, 11, and 12 compare the predicted temperature profiles with the measured values for various values of the three listed parameters, and Table 1 summarizes the net heat flow to the LN₂-filled tank for the same conditions.

The conclusions from this parametric study were:

- (1) The heat leak to the LN₂ tank is sensitive to the thermal conductance, $k\delta$, of the tube wall (δ is the wall thickness). Table 1 shows that, for the conditions of Test III-3A8, the computer model

*Test III-3A8 was conducted with a penetration configuration identical to that shown in Figure 25. The interior walls of the penetration were coated with 3M Velvet black.

predicts that three to four times more energy will be conducted into the cold-end plate from the adjacent tube wall as will be radiated to the cold plate from the length of the pipe. The first three conditions in Table 1 show the significant effect of a variation in the tube wall conductivity at the warm end of the pipe. (The temperature distributions for these conductivities are shown in Figure 12.) This strong dependence is due to the fin effect of the tube wall at both ends of the pipe. The heat conducted from the hot cap into the tube wall and then radiated into the interior was, therefore, a significant source of entering energy. The fin effect is discussed in greater detail later in this section.

- (2) A shielding factor, μ , greater than 200 ($\mu = 1/\epsilon_{\text{eff}}$) for the MLI on the exterior of the pipe yielded essentially the same thermal results as an adiabatic exterior surface. A shielding factor of 500 ($\epsilon_{\text{eff}} = 0.002$) was assumed for the computer model of the Cap A and Cap B hardware,* and in all computer runs the heat flow through the MLI was small in comparison to the heat transferred down the pipe. (See, for example, Figure 13.)
- (3) Changing the emittance of the interior surface from 0.9 to 0.035 reduce the computed heat flow to the cold end from 18.2 Btu/hr (5.33 watts) (Run No. 9) to 9.20 Btu/hr (2.70 watts) (Run No. 11), or by approximately 50%.

A low-emittance interior surface has two significant effects on the thermal behavior of the pipe penetration. First, it decreases the amount of radiant energy emitted by the hot cap at a given temperature and, similarly, decreases the amount of energy absorbed by the cold-end cap for a given incident flux. Second, it reduces the fin effect of the tube wall near the hot- and cold-end plates.

To clarify the thermal behavior of the fins, the significant parameters of a fin having one adiabatic surface (exterior of pipe) will now be examined. The governing differential equation for a fin of unit width is:

$$-\frac{d^2 T(x)}{dx^2} + \frac{\epsilon \sigma}{k\delta} [T^4(x) - T_e^4] = 0 \quad (1)$$

where

$T(x)$ - local tube wall temperature

*See Section V-C for description of "B" Cap.

- T_e - environment "temperature"
 ϵ - emittance of tube wall
 $k\delta$ - thermal conductance of tube wall
 σ - Stefan Boltzmann constant

and $x = 0$ is the base of the fin where it joins an end plate. If one multiplies equation (1) through by $2 (dT/dx)$ and then integrates from 0 to ∞ , making use of the assumption that $(dT/dx)_\infty = 0$ and $T(\infty) = T_e$, the equation takes the following form:

$$-\left(\frac{dT}{dx}\right)_0^2 = \frac{\epsilon \sigma}{k\delta} \left[\frac{2}{5} (T_e^5 - T_0^5) - 2T_e^4 (T_e - T_0) \right]$$

For the heat leak through a unit width of the base of the fin, q_f , this gives:

$$q_f = \left[\epsilon k\delta \left(\frac{8}{5} \sigma T_e^5 + \frac{2}{5} \sigma T_0^5 - 2\sigma T_e^4 T_0 \right) \right]^{\frac{1}{2}} \quad (2)$$

The term in equation (2) that contains the tube wall properties is $\sqrt{\epsilon k\delta}$. Therefore, the heat entering the pipe penetration by way of the tube wall fin at the hot end (and also being conducted into the cold end) can be expressed more generally as:

$$q_f = (\epsilon k\delta)^{0.5} f(T_0, T_e)$$

The maximum variation in the entering conductive heat flow due to uncertainties in the tube wall properties can now be expressed as:

$$\Delta q_f = \left| \left(\frac{\partial q_f}{\partial \epsilon} \right) \Delta \epsilon \right| + \left| \left(\frac{\partial q_f}{\partial k} \right) \Delta k \right| + \left| \left(\frac{\partial q_f}{\partial \delta} \right) \Delta \delta \right| \quad (3)$$

where

$\Delta \epsilon$ - uncertainty in wall emittance

Δk - uncertainty in thermal conductivity of wall

$\Delta \delta$ - uncertainty in thickness of wall .

The three influence coefficients are expanded below.

$$\frac{\partial q_f}{\partial \epsilon} = \frac{1}{2} \frac{k\delta}{(\epsilon k\delta)^{0.5}} f(T_o, T_e)$$

$$\frac{\partial q_f}{\partial k} = \frac{1}{2} \frac{\epsilon\delta}{(\epsilon k\delta)^{0.5}} f(T_o, T_e)$$

and

$$\frac{\partial q_f}{\partial \delta} = \frac{1}{2} \frac{\epsilon k}{(\epsilon k\delta)^{0.5}} f(T_o, T_e)$$

For the condition of a 750°R end-cap temperature, with Cap A pipe configuration (see Section V), the following values were used in the uncertainty analysis:

$$\epsilon = 0.41$$

$$k = 8.7 \text{ Btu/hr ft } ^\circ\text{R} \text{ (0.150 watt/cm- } ^\circ\text{K)}$$

$$\delta = 0.0275 \text{ in. (0.0699 cm)}$$

$$\Delta\epsilon = 0.041 \text{ (10\%)}$$

$$\Delta k = 0.58 \text{ Btu/hr ft } ^\circ\text{R} \text{ (0.01 watt/cm- } ^\circ\text{K)}$$

$$\Delta\delta = 0.006 \text{ in. (0.5 x 10}^{-3} \text{ ft, 0.15 cm)}$$

The environmental temperature, T_e , to be used in the uncertainty analysis is not an easily defined quantity, as each fin is exposed to a range of environmental temperatures which are dominated by the local end-cap temperature. For the 750°R end-cap Test IV-3C, a mean environmental temperature of 666°R in equation (2) makes the analytical fin model predict a total conductive heat loss from the warm cap of 18.6 Btu/hr (5.46 watts), which is equal to that predicted by the full computer model. (See Figure 13.) This temperature is reasonably close to 630.5°R, which is the mean σT^4 between the hot (750°R) and cold (140°R) end-plate temperatures.

If

$$T_e = 666^\circ\text{R}$$

then

$$f(T_o, T_e) = 9.57 \times 10^3 \text{ Btu-}^\circ\text{R/hr ft}^2 \text{ (1.68 watt-}^\circ\text{K/cm}^2\text{)}$$

The three influence coefficients in equation (3) can now be evaluated.

$$\frac{\partial q_f}{\partial \epsilon} = 13.8 \text{ Btu/hr ft (0.133 watt/cm)}$$

$$\frac{\partial q_f}{\partial k} = 0.66^\circ\text{R (0.367}^\circ\text{K)}$$

$$\frac{\partial q_f}{\partial \delta} = 2500 \text{ Btu/hr ft}^2 \text{ (0.788 watt/cm}^2\text{)}$$

The maximum uncertainty in the entering conductive heat flow per unit width of the fin is:

$$\Delta q_f \text{ (English units)} = |(13.8)(0.041)| + |(0.66)(0.58)| + |(2500)(0.5 \times 10^{-3})|$$

$$= 0.56 + 0.38 + 1.25 = 2.2 \text{ Btu/hr ft (0.021 watt/cm)}.$$

For the 6-inch-diameter pipe, the uncertainty in the total conductive entering heat flow, $(\Delta q_f)_T$, is:

$$(\Delta q_f)_T \text{ (English units)} = 2.2 \pi D = 3.4 \text{ Btu/hr}$$

$$(\Delta q_f)_T \text{ (metric units)} = 0.021 \pi D = 1.0 \text{ watts}$$

This value is approximately 18% of the computed conductive heat flow at the hot end and approximately 10% of the 35 Btu/hr (10.2-watt) computed heat flow to the nitrogen tank.

2. Pipe Penetration Computer Model Results

The temperature distributions and heat flows predicted by the computer model were in good agreement with the measured test results for all conditions except the heat flow to the nitrogen tank for a 750°R end-cap temperature. The experimentally measured temperature distributions in the MLI for the 750°R test conditions did show indications of a conductive mode of heat transport in the exterior insulation wrap. However, it is difficult to postulate a reasonable heat transport mode in the MLI that would have increased the computed heat flow to the nitrogen tank from 34 to 44 Btu/hr (10.1 to 12.8 watts). As a further check on the uncertainties in the tube wall properties for the 750°R test conditions, several parametric runs were made with the computer model in which the tube wall thickness was increased from 0.027 inch to 0.033 inch. These results are summarized in Table 2.

The thicker wall increased the computed heat leak to the calorimeter tank for both conditions; it therefore increased the difference between the computed and measured heat flow for the IV-3A5 test but decreased the difference for the IV-3C test. The same tube wall was used in both tests, so the 8.5 Btu/hr (2.5-watt) difference in Test IV-3C cannot be satisfactorily explained by an uncertainty in the tube wall thickness.

3. Pipe Penetration Thermal Design Guidelines

The results of the thermal analysis indicate several ways to reduce the heat flow in a given pipe penetration. To reduce the fin-effect heat flow, the tube wall conductance, $k\delta$, should be low. The use of a thin tube wall, made from a material of low thermal conductivity, would be a good thermal design criterion. This criterion would, of course, have to be compromised with the requirements for structural strength in the choice of a final design for a particular installation. Figure 14 is a plot of computed total heat leak as a function of wall thickness. At an end-cap temperature of 750°R, there was a maximum (computed) reduction of 18.5 Btu/hr (5.42 watts, 53%) when the wall thickness was reduced from 0.027 to 0.001 inch.

A second criterion for a low heat leak would be a low-emittance, diffuse interior surface. For the general category of pipe penetrations with varying over-all lengths, the cold end would be a fixed location that could be treated to have a low emittance. (A specular component to the tube wall reflectance would increase the transmission down the pipe and, therefore, increase the heat leak.)

Several parametric runs were made with the thermal computer model of the Cap B pipe penetration to determine the reduced heat leak to the cold end as a function of the width of a band of low-emittance material ($\epsilon = 0.04$) placed on the tube wall at the cold end. For these runs, the cold end was given an emittance of 0.04. The results are plotted in Figure 15 for two wall thicknesses. For a 0.027-inch tube wall and a 750°R end-cap temperature, there is a 30% (computed) reduction in the heat leak when the full length of the tube is assumed to have an emittance of 0.04. If the tube wall thickness were then reduced from 0.027 inch to 0.010 inch, a reduction of approximately 60% in heat flow could be realized when compared with the uncoated 0.027-inch wall model.

A second approach to the application of low- ϵ internal surfaces would be to have one or more baffles in the pipe to block the radiation coming from the warm end. A single-baffle design was represented in the computer model by assuming a low- ϵ surface for the hot-end cap. Parametric runs were then made to determine the heat leak to the cold end as a function of the width of a low-emittance band on the tube wall which started at the warm end, next to the baffle. (See Figure 16*.) A comparison of Figures 15 and 16 shows that, for a given area of low-emittance material, a greater reduction in heat flow can be achieved by applying the low- ϵ coating to the warm-end region rather than to the cold end; this reduces the amount of radiant energy entering the system per unit time rather than reducing the amount of radiant energy being absorbed at the cold end. A further reduction in heat leak could be achieved by adding more low-emittance radiation baffles in the pipe penetration. The design of the baffles would, of course, have to permit the flow of liquid through the pipe.

For long-term storage requirements, it would be necessary to reduce the heat leak to the cold tank below the values measured and computed in this work. As an example, the minimum computed heat leak of 10 Btu/hr (2.9 watts) (750°R end cap, 0.010-inch wall, all low- ϵ surfaces) would boil off approximately 36 pounds of (para) liquid hydrogen over a period of one month. This boil-off rate would justify additional weight for thermal control devices such as radiation shields. Further analysis would then be needed to define an optimum thermal design.

4. Scaling of Pipe Penetration Heat Flows

The techniques of thermal scale modeling can be used to generalize the heat leaks computed in the present analysis to pipes of different dimensions but having the same L/D ratio. The modeling approach used will be one in which the temperature distributions will be maintained. This is important, as the thermal conductivity of the tube wall is temperature-dependent. If the (dimensionless) temperature distribution in the pipe penetration (the prototype) and a

*Note that the warm end is on the left side of this graph, opposite to Figure 15.

scaled model were different, the (dimensionless) distribution of thermal conductivity would also be different and the scaling techniques could not be easily applied.

For the general case of a two-dimensional heat transfer problem, with conduction and radiation, the following two dimensionless groups define the scaling ratios which are to be maintained:

$$\left(\frac{qL^2}{k\delta T} \right)_{\text{model}} = \left(\frac{qL^2}{k\delta T} \right)_{\text{prototype}} \quad (4a)$$

$$\left(\frac{\epsilon \sigma T^3 L^2}{k\delta} \right)_{\text{model}} = \left(\frac{\epsilon \sigma T^3 L^2}{k\delta} \right)_{\text{prototype}} \quad (5a)$$

where

- q - power generation per unit area
- L - characteristic length
- k - thermal conductivity
- δ - characteristic thickness (tube wall)
- T - characteristic temperature
- ϵ - surface emittance
- σ - Stefan-Boltzmann constant

The scaled versions are assumed to be of the same material as the prototype, with the same emittances; then, since the temperatures are being maintained, the scaling ratios simplify to:

$$\left(\frac{qL^2}{\delta} \right)_{\text{model}} = \left(\frac{qL^2}{\delta} \right)_{\text{prototype}} \quad (4b)$$

$$\left(\frac{L^2}{\delta}\right)_{\text{model}} = \left(\frac{L^2}{\delta}\right)_{\text{prototype}} \quad (5b)$$

For a fixed L/D ratio, which maintains a fixed geometry for the radiation cross fire inside the pipe, the ratio of the tube wall thicknesses of the model and the prototype is now a function of a ratio of the characteristic lengths of the model and prototype; i.e.,

$$\frac{\delta_m}{\delta_p} = \left(\frac{L_m}{L_p}\right)^2 \quad (6a)$$

For this condition, q , the power dissipation per unit area, is the same in model and prototype. The total heat leak is then scaled by the ratio of the areas or the square of the ratio of a characteristic length:

$$\frac{Q_m}{Q_p} = \left(\frac{L_m}{L_p}\right)^2 \quad (6b)$$

Given the heat leak for the prototype, it is now possible to compute a heat leak as a function of the pipe length, L . In summary, scaled heat leaks can be computed for pipes of different sizes when the following restrictions are satisfied:

- (1) The L/D ratio is maintained,
- (2) Materials and surface emittances are maintained,
- (3) $\delta_m/\delta_p = (L_m/L_p)^2$, and
- (4) The boundary temperatures are fixed.

Figure 17 is a plot of scaled heat leak against pipe length for the specific case of a 750°R hot-end cap temperature and a 140°R cold-end and environment temperature. The solid curves on Figure 17 show the scaled heat leak for pipes of different lengths but having a fixed (dimensionless) temperature distribution. Note that as the scaled length of the pipe increases, so does its scaled tube wall thickness. Cross-plotted on the same graph are curves of constant tube wall thickness. Using the graph as a design tool, one can enter it with a given pipe

length (fixed by pipe run geometry) and a given tube wall thickness (determined from pressure requirements) and read out the heat leak due to the pipe. Since an individual graph is restricted to a given L/D ratio with given pipe materials, surface emittances, and boundary temperatures, a family of graphs would be required to cover a range of L/D ratios and materials.

TABLE 1
HEAT FLOW SUMMARY FOR PARAMETRIC STUDY AND COMPARISON WITH TEST III-3A8

Computer Run	Wall Conductivity, Btu/hr ft °F (watts/cm °K)	MLI Shielding Factor	Emittance		Heat Flow to Cold End, Btu/hr (watts)		
			Warm End	Cold End	Radiation	Conduction	Total
1	8.7(0.150)	∞	0.90	0.90	3.34(0.98)	14.9 (4.36)	18.2(5.34)
2	4.9(0.085)	∞	0.90	0.90	3.88(1.14)	5.70(1.67)	9.6(2.81)
3	8.7→4.9(0.150)→(0.085)	∞	0.90	0.90	3.85(1.13)	14.0 (4.10)	17.8(5.23)
4	8.7→4.9(0.150)→(0.085)	50	0.90	0.90	4.26(1.25)	15.9 (4.66)	20.2(5.91)
5	8.7→4.9(0.150)→(0.085)	100	0.90	0.90	3.92(1.15)	14.6 (4.29)	18.5(5.44)
6	8.7→4.9(0.150)→(0.085)	200	0.90	0.90	3.78(1.11)	14.0 (4.11)	17.8(5.22)
7	8.7→4.9(0.150)→(0.085)	500	0.94	0.94	4.06(1.19)	14.3 (4.20)	18.4(5.39)
8	8.7→4.9(0.150)→(0.085)	500	0.86	0.86	3.75(1.10)	14.2 (4.15)	20.0(5.25)
9	8.7→4.9(0.150)→(0.085)	500	0.94	0.94→0.86	3.81(1.12)	14.4 (4.21)	18.2(5.33)
10	8.7→4.9(0.150)→(0.085)	500	0.35	0.35	1.94(0.57)	12.6 (3.70)	14.5(4.27)
11	8.7→4.9(0.150)→(0.085)	500	0.035	0.035	0.29(0.10)	8.9 (2.60)	9.2(2.70)
EXPERIMENTAL HEAT FLOW TO LN ₂ TANK OBTAINED WITH TEST III-3A8							15.3(4.48)

NOTE: Cold end of pipe was held at 140°R (77°K) in computer model, and hot end was held at 540°R shroud temperature 540°R.

TABLE 2

PENETRATION HEAT FLOW SUMMARY,
EFFECT OF INTERIOR WALL EMITTANCE AND WALL THICKNESS

<u>Test No. (1)</u>	<u>Wall Finish</u>		<u>Wall Thickness</u> <u>in.</u>	<u>Q_{Computed},</u> <u>Btu/hr</u> <u>(watts)</u>	<u>Q_{Measured},</u> <u>Btu/hr</u> <u>(watts)</u>
IV-3A5	3M Black: $\epsilon = 0.94$		0.027	45.0(13.2)	43.0(12.6)
			0.033	47.5(13.9)	43.0(12.6)
IV-3C	Sandblasted: $\epsilon = 0.41$		0.027	35.8(10.5)	44.3(13.0)
			0.033	37.5(11.0)	44.3(13.0)

(1) These test numbers refer to tests conducted in this program. The description of the test and thermal boundary conditions is given in Section V-B.

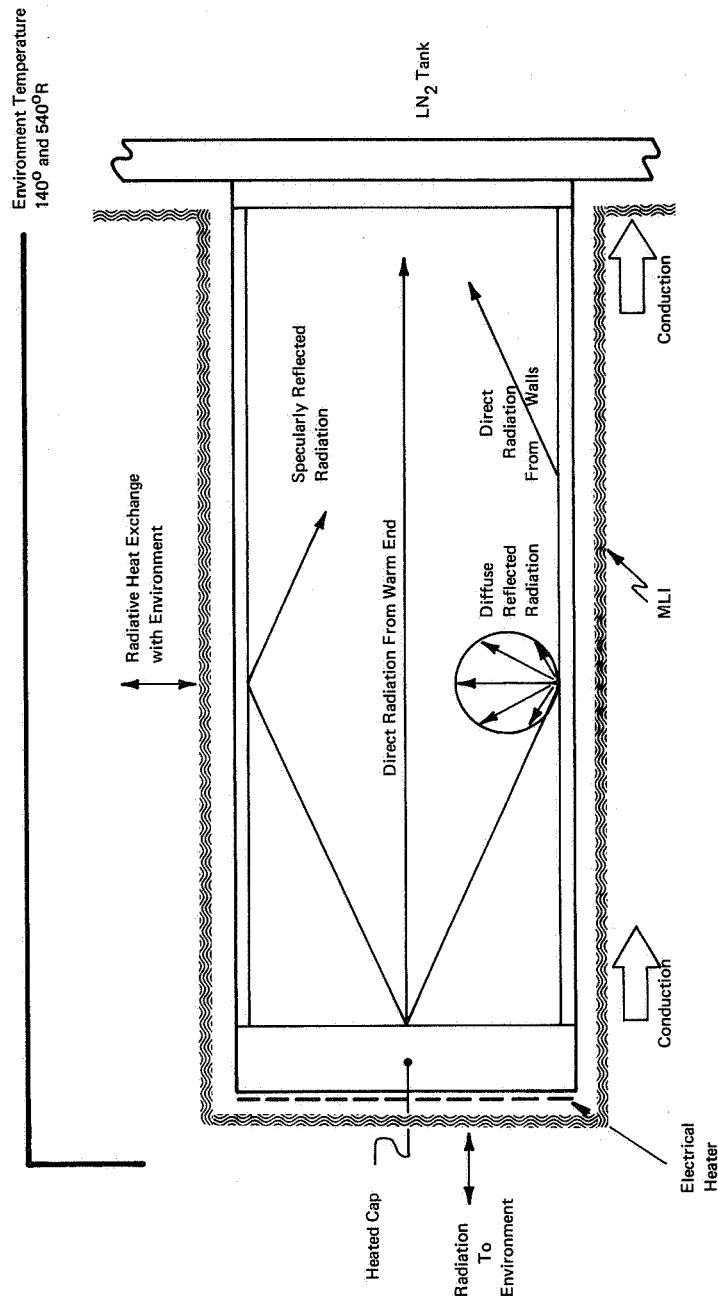


FIGURE 7 THERMAL PATHS IN PIPE PENETRATION

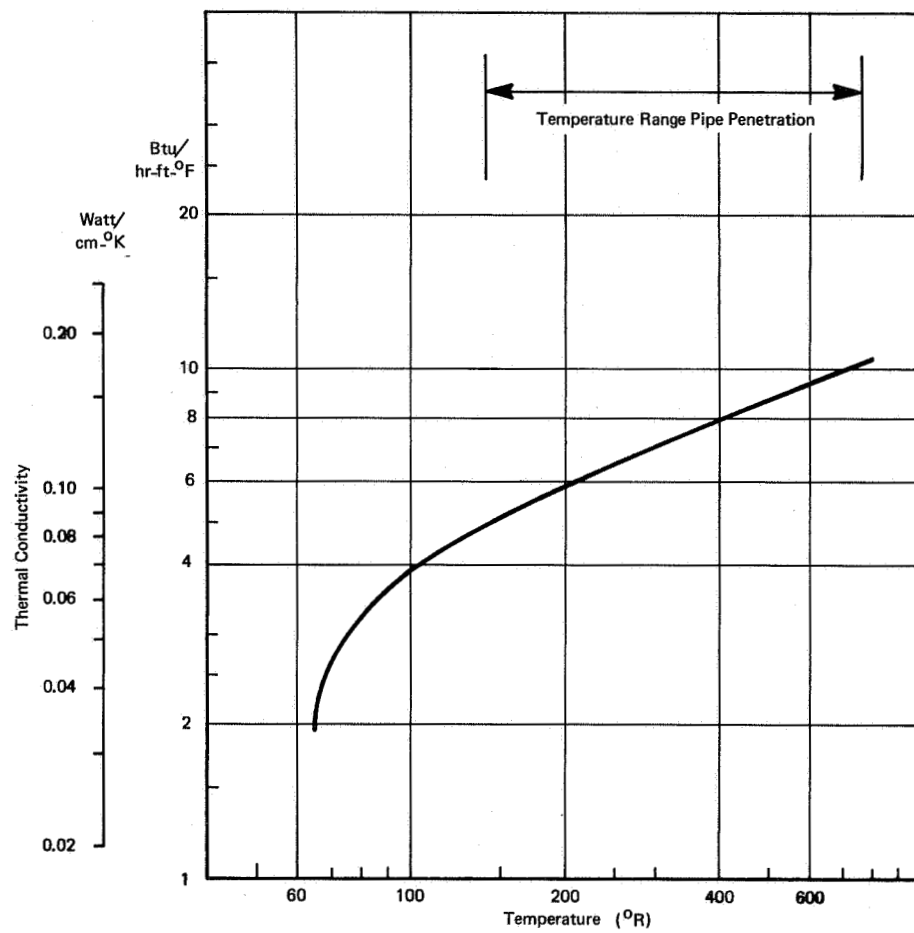


FIGURE 8 THERMAL CONDUCTIVITY OF 304 STAINLESS STEEL

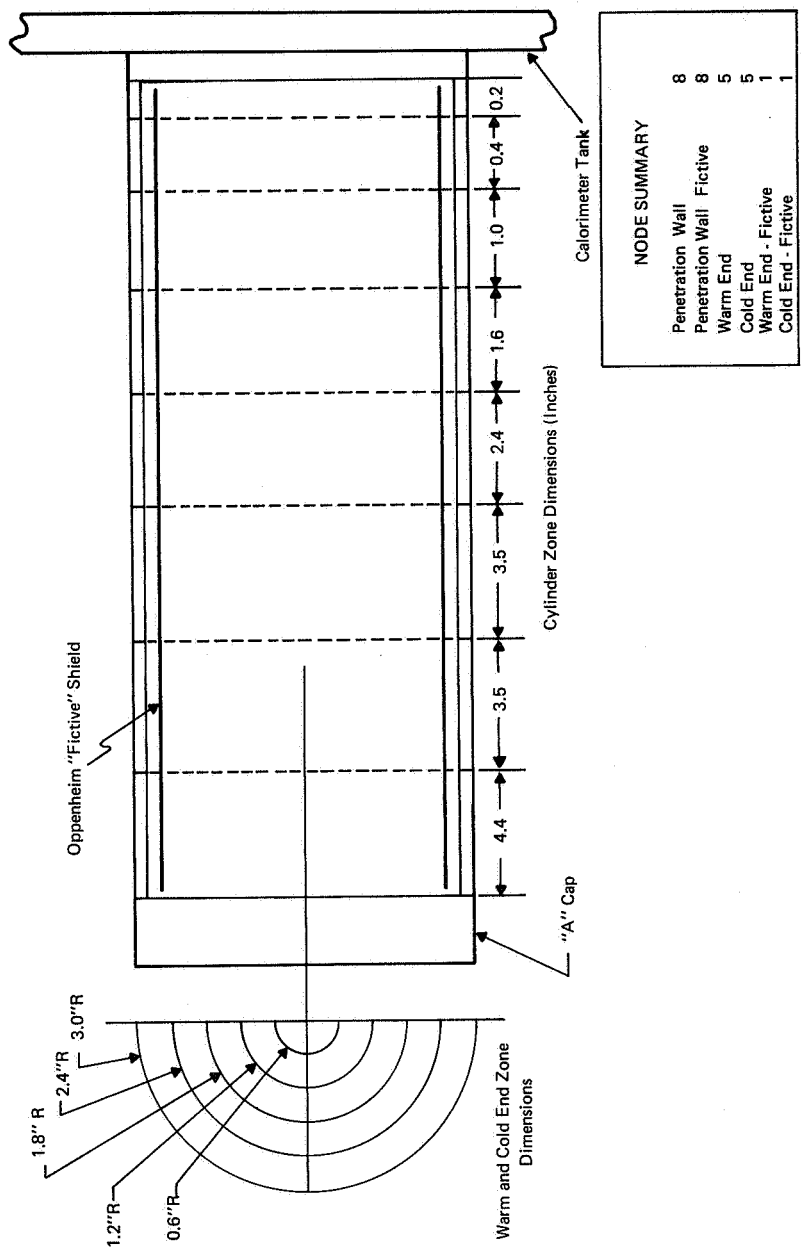


FIGURE 9 ZONE SUBDIVISIONS IN THERMAL COMPUTER MODEL OF PIPE PENETRATION

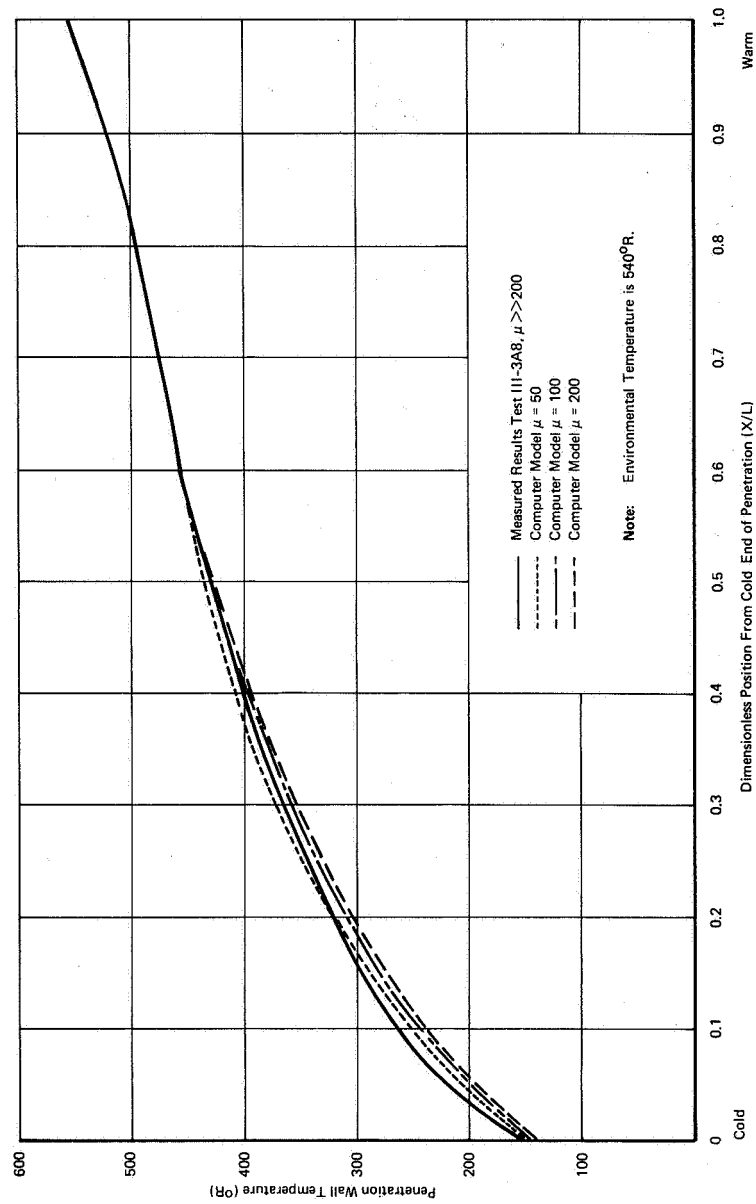


FIGURE 10 PENETRATION WALL TEMPERATURE DISTRIBUTION,
EFFECT OF INSULATION SHIELDING FACTOR, μ

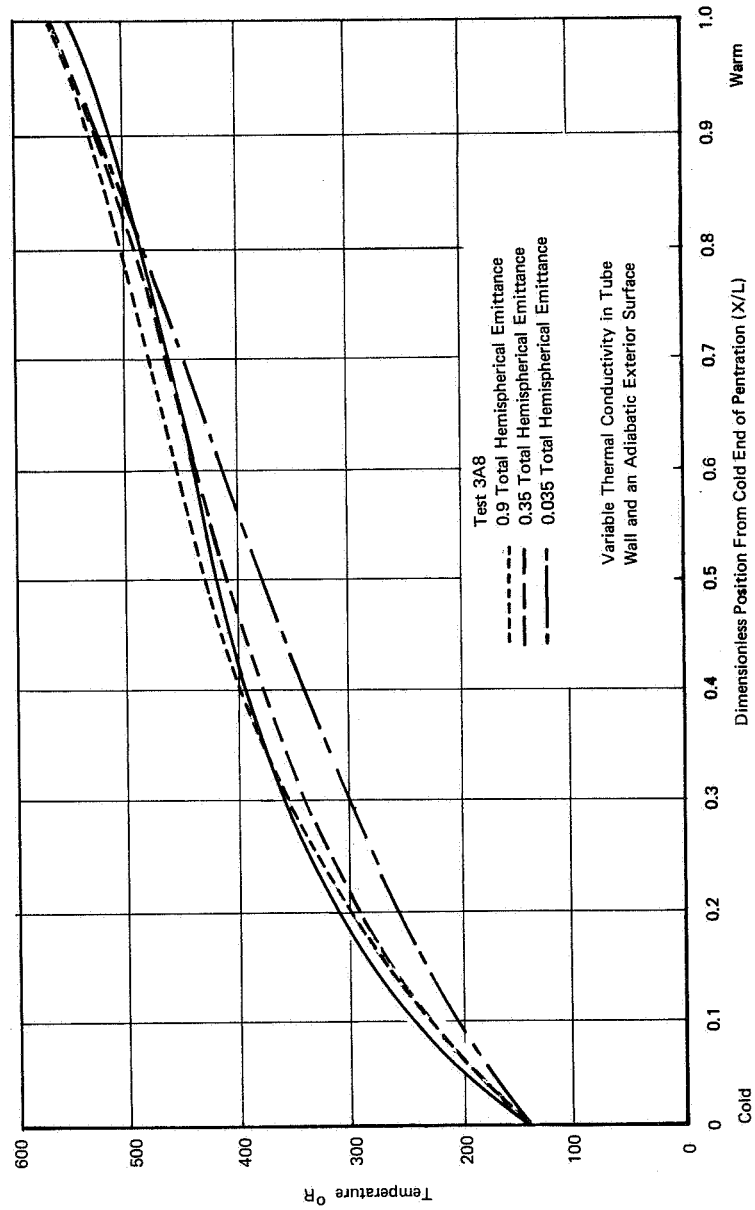


FIGURE 11 TEMPERATURE DISTRIBUTION IN WALL OF PIPE PENETRATION AS A FUNCTION OF INTERIOR EMITTANCE

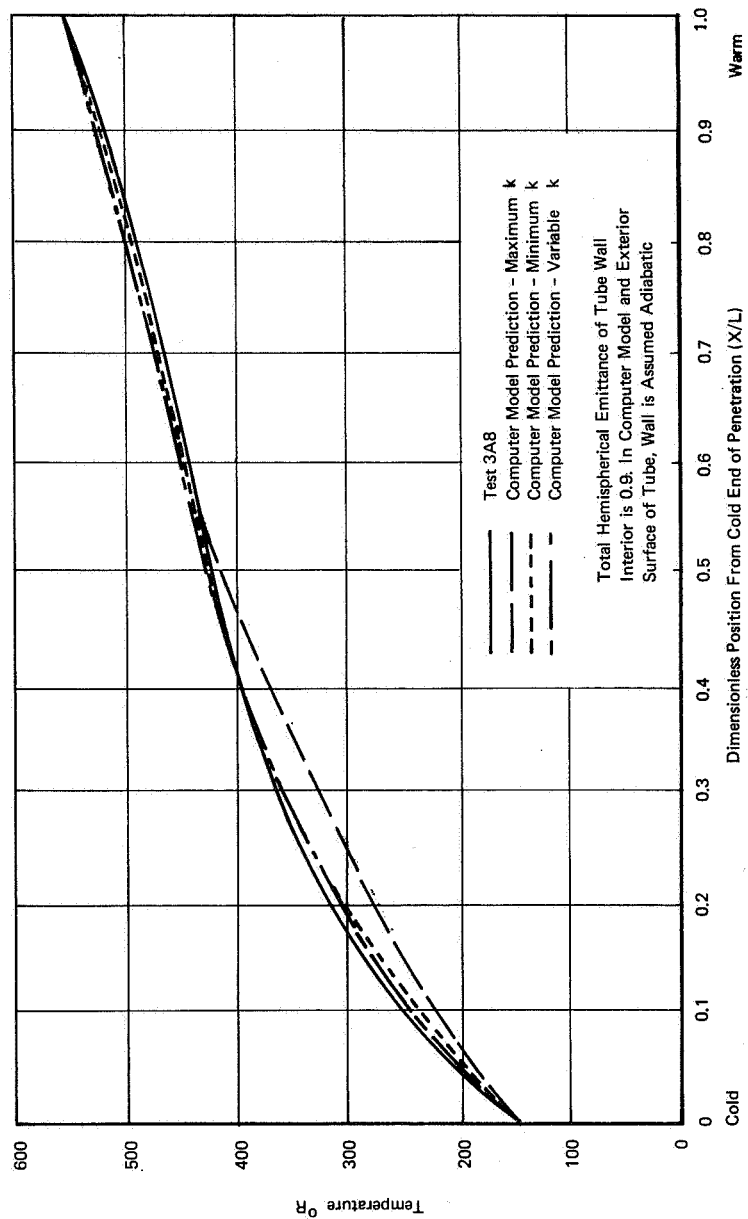


FIGURE 12 TEMPERATURE DISTRIBUTION IN WALL OF PIPE PENETRATION AS A FUNCTION OF THERMAL CONDUCTIVITY OF WALL

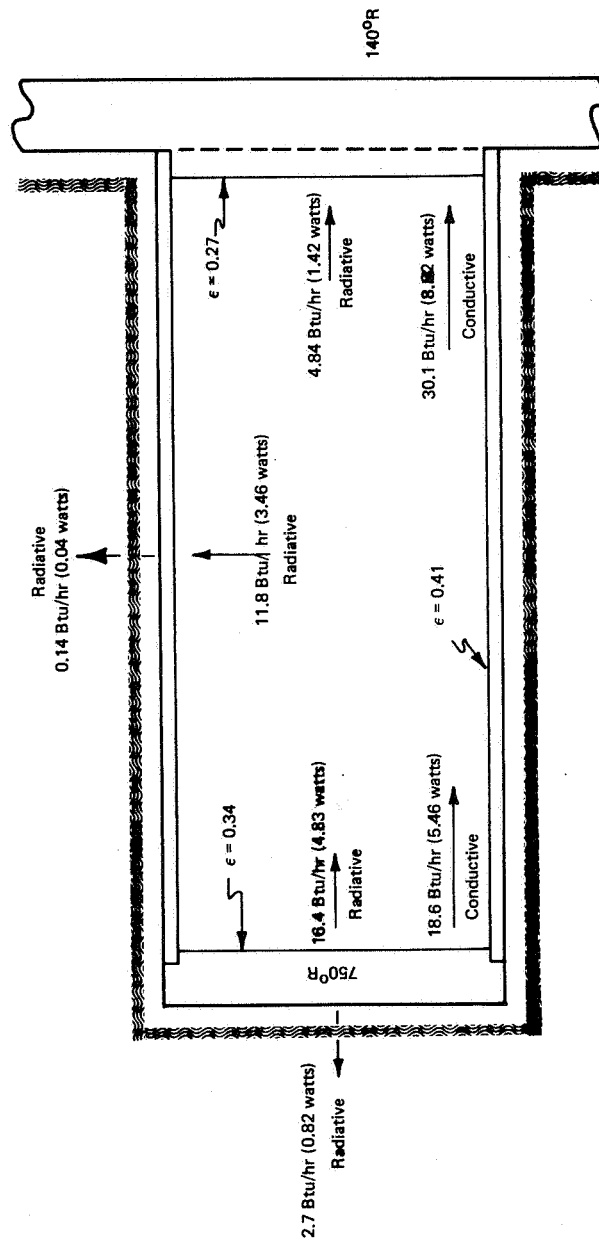


FIGURE 13 COMPUTED LOCAL NET HEAT FLOWS FOR TEST IV-3C

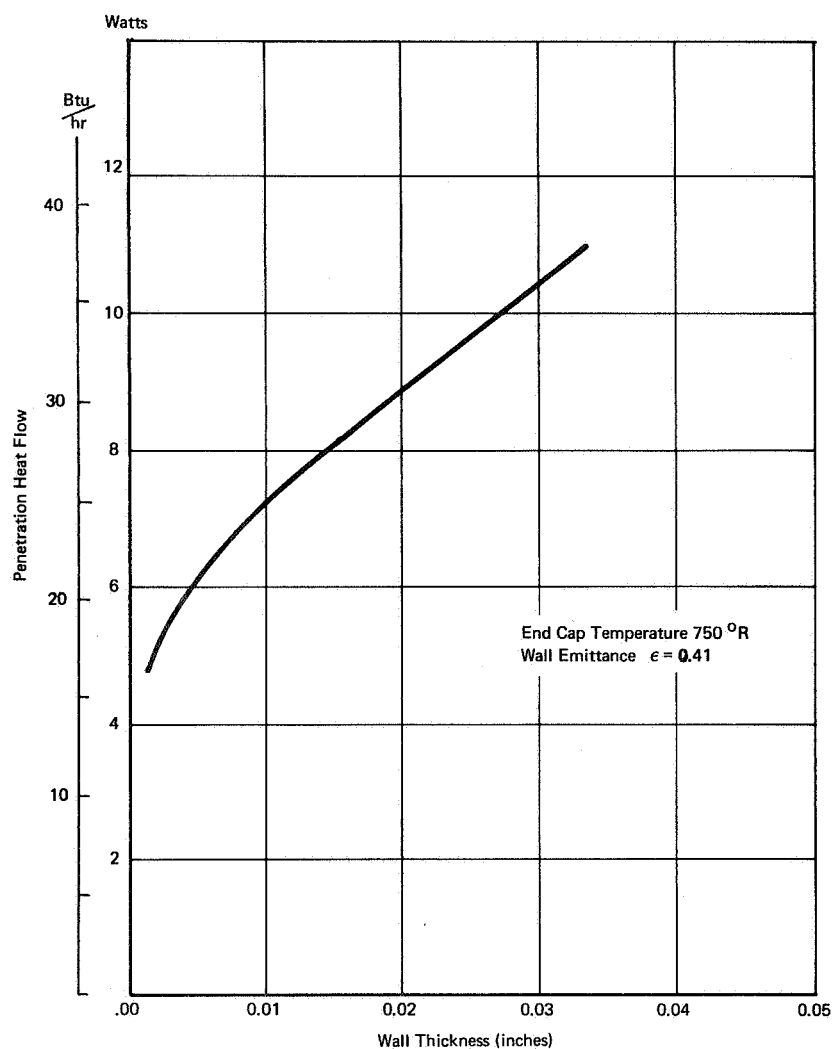
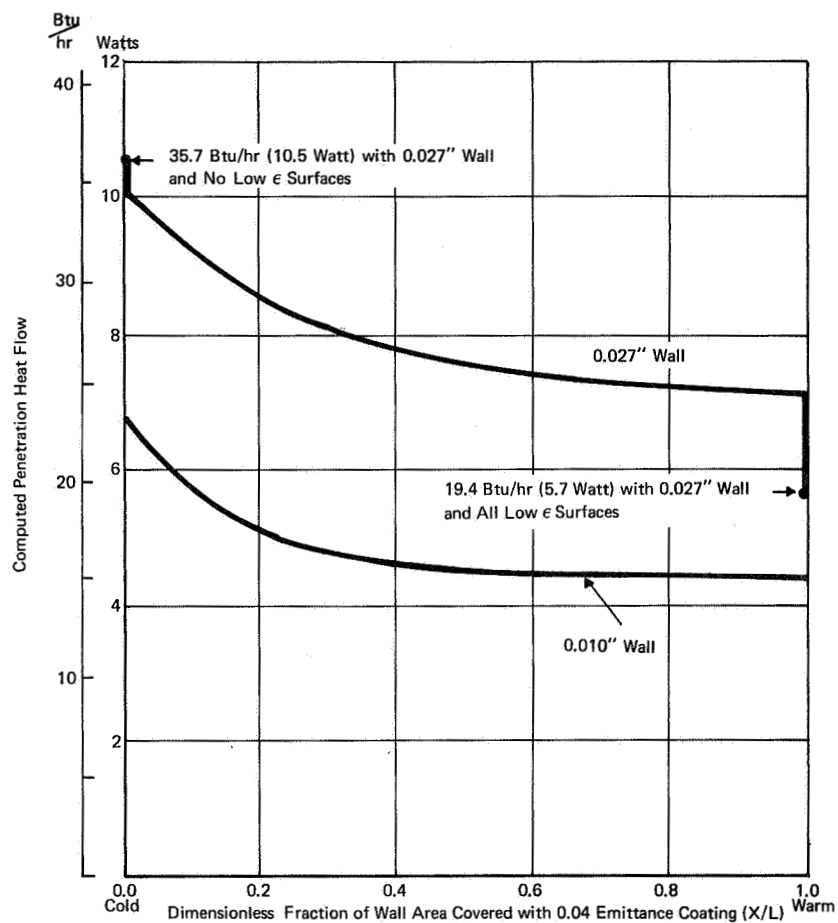


FIGURE 14 EFFECT OF WALL CONDUCTANCE ON
COMPUTED PENETRATION HEAT FLOW



Notes:

- 1) Cold-End Emittance = 0.04
- 2) 750°R Hot-End Cap Temperature

FIGURE 15 COMPUTED PENETRATION HEAT FLOW FOR VARYING AMOUNTS OF LOW-EMITTANCE TUBE WALL AT COLD END

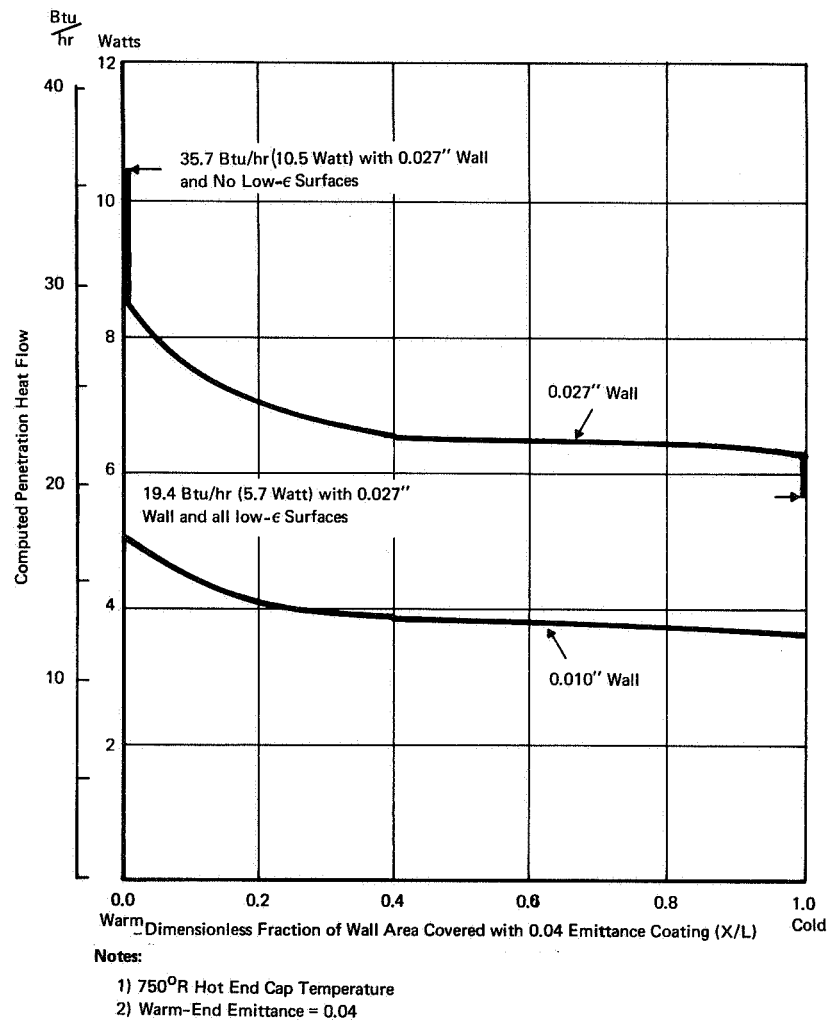


FIGURE 16 COMPUTED PENETRATION HEAT FLOW FOR VARYING AMOUNTS OF LOW-EMITTANCE TUBE WALL AT WARM END

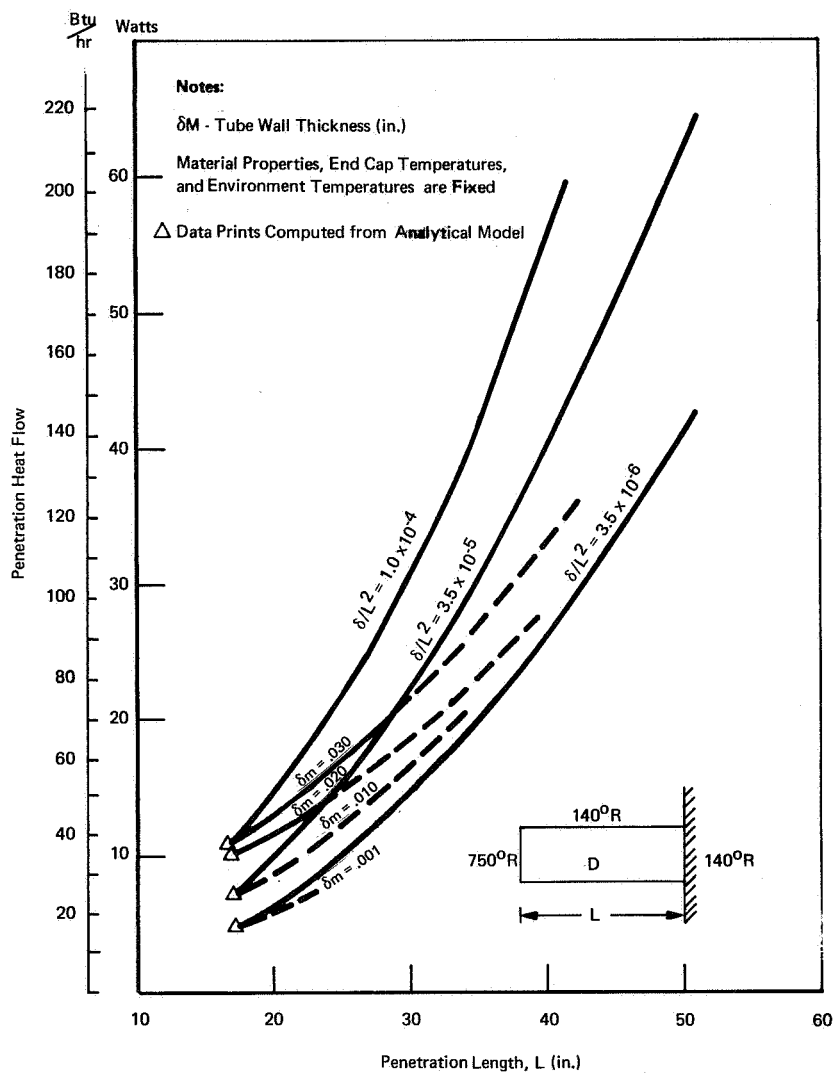


FIGURE 17 SCALED PENETRATION HEAT FLOWS FOR PIPE PENETRATIONS WITH $L/D = 2.8$

V. EXPERIMENTAL SYSTEMS AND RESULTS - TASK I

This task was undertaken for two purposes: the first was to obtain information that could be used to develop an understanding of the relevant mechanisms associated with the transport of heat in an insulated pipe connected to a cryogenic tank; the second was to develop techniques for insulating both the tank and penetration with negligible degradation in the thermal performance of either insulation.

The required penetration information was obtained both from analytical and experimental models used to simulate the piping connected to the cryogenic propellant tanks of a space vehicle. The experimental model consisted of the 48-inch-diameter tank calorimeter, described previously, and a 6-inch-diameter by 17-inch-long pipe penetration connected to the bottom center of the tank. The experimental data consisted of temperatures measured in the multilayer insulation and penetration wall and the measured heat flow to the tank resulting from the penetration's presence. The analytical model was that described in Section IV. The parameters used in this model were varied until an acceptable correlation was obtained among the predicted and experimental results. From the development of this correlation, an understanding of the relevant heat transport mechanism has developed.

Four experimental systems were investigated under Task I of the current program using the tank calorimeter. These systems are numbered 15 through 18*. System No. 15 was fabricated as a 10-shield system with no penetrations, so that it could be used in establishing the heat flux performance of the MLI alone. This insulation was subsequently modified, and the pipe penetration was attached to the tank calorimeter to make System No. 16. To accomplish this, the insulation of System 15 was cut out at the penetration location and modified to mate with the 10-shield MLI placed on the penetration exterior at the sides and free end. Systems No. 17 and 18 were formed by modifying the penetration in two subsequent consecutive steps, which are discussed in detail in subsequent sections.

Twenty-two tests were performed with these four systems. In the study of each, prior supporting experimental and analytical information was developed to predict the heat flow and temperature performance of the vital areas of the MLI and penetrations under consideration. In this section, the test articles and experimental conditions are identified in detail, and the experimental and analytical results are presented and compared.

*System numbers are consecutive with those applied to systems tested under Contracts NAS3-4181 and NAS3-6283.

A. INSULATION SYSTEM NO. 15

All insulation systems produced by Arthur D. Little, Inc., for NASA/LeRC in previous programs, with one exception, contained five radiation shields. The use of 10 shields for the current test series was based upon a contract requirement which aimed at increasing the amount of information available on the thermal performance of systems with an increased number of shields. This information is important at this time to help resolve the doubts held in some quarters on the validity of the predicted thermal performance of MLI systems with many shields as extrapolated from the measured performance of MLI systems containing few shields.

Aluminized Mylar was adopted as the shield material because of its availability and cost, although gold-metallized Mylar would have produced a superior insulation. This latter factor was not considered detrimental to the aims of the program.

1. Description

Insulation System No. 15 consisted of 10 radiation shields and 11 spacers. The insulation applied to the tank calorimeter had an area of 38.14 ft² and an approximate weight of 1.7 pounds. The completed system is shown in Figure 18, and detailed characteristics are presented in Table 3.

a. Shields

The radiation shields consisted of $\frac{1}{4}$ -mil polyester film aluminized on both surfaces. This material was purchased from Hastings and Company, Inc., of Philadelphia on the specification presented in Table 4. A coating thickness in the range of 600 to 1000 Å had been requested; actual coating thickness ranged between about 400 and 600 Å, and the coating of one surface was generally thicker than the other. The average surface emittance of the shields was found by measurement with the emissometer to be 0.026.

The thicknesses and emittances of the aluminum coatings on the shields used in the construction of System No. 15 are summarized in Table 5. Two samples were obtained from each side of each shield of the system. The location of the sample taken from the roll material was varied, as indicated. The thickness of the aluminum coating on each sample was determined from measurements of its electrical resistance. Emittance measurements were made on approximately 50% of the samples.

The emittance and thickness data presented in Table 5 are summarized below in a more meaningful manner.

<u>Side A Coatings</u>		
	<u>Thickness (Å)</u>	<u>Emittance</u>
No. of Data	20	10
Average Value	513	0.0230
Average Deviation	43	0.0016
Standard Deviation	56	0.0018

<u>Side B Coatings</u>		
	<u>Thickness (Å)</u>	<u>Emittance</u>
No. of Data	20	11
Average Value	456	0.0293
Average Deviation	39	0.0035
Standard Deviation	41	0.0042

As noted, the average emittances of Sides A and B were 0.023 and 0.029, respectively. The coating on Side A had an emittance value comparable to that of the best aluminum coatings measured in the previous program.⁽²⁾ By these standards, the Side B coating was poor but acceptable. The difference in the emittance values was larger than expected on the basis of coating thickness. Also, the larger variations in the emittance of the Side B coating were indicative of poor quality. This condition may be attributed to shortcomings in the production process, such as low coating deposition rate and poor coater vacuum.

A shield emittance of 0.026 was used in computing the theoretical thermal performance of System No. 15; this value was obtained by averaging the measured emittance values for each coating surface.

b. Spacers

Each spacer of the multilayer system was made up from two layers of silk netting. This material was purchased from the Jordan Marsh Company of Boston. It was applied to the calorimeter in the as-received condition, and no special treatments were given to it except for careful handling to prevent soiling and tearing. Additional pertinent information is given in Table 3.

c. Application

Each of the two netting layers in each spacer was applied separately. Also, each layer consisted of three segments, namely, top and bottom ends and a cylindrical section. The ends were circles of material about 38 inches in diameter. The material for the cylindrical portion of the calorimeter was formed from 2-foot-wide material gored at 3-inch intervals. These gores permitted the

spacer layer to conform to the knuckle radii of the tank calorimeter. The cylindrical portion was also slit in the transverse direction to allow for thermal contraction of the netting layer. The seams in each layer were butt-jointed to prevent excessive layup of material; this was accomplished by cutting the material to accurate dimensions and then sewing the segments together. The slits and gores on adjacent layers of netting were staggered to eliminate areas in the spacers where the shields could become shorted together.

Like the spacers, each radiation shield was made up of a top end, bottom end, and cylindrical segment. The ends were circles about 42 inches in diameter with one radial slit to permit conformance to the spherical segment of the tank ends. The cylindrical segment was gored on a 3-inch pitch to permit contouring the side sheet to the knuckle radii of the tank calorimeter. All seams were formed by overlapping the edges of the radiation shields approximately one inch so that the segments were in contact with each other. Small strips of Scotch tape were used to hold the radiation shield in place until covered by the outer netting spacer. The radiation shields were positioned and prevented from shifting during movement of the calorimeter by the light frictional forces produced by the spacer material.

d. Thermocouples

Copper-constantan thermocouples were attached to shields 1, 2, 4, 7, and 10 of the system at each of the locations indicated in Figure 19; all were in the same vertical plane. These thermocouples were made from 40-gage wire and attached to the shield with double-faced masking tape. Aluminized Mylar was adhered to the outer surface of the masking tape to eliminate alteration of the shield surface characteristics in the region of the thermocouple junction.

2. Predicted Heat Flux Based on Measured Shield Emittance

The emissometer measurements performed on shield materials used in the fabrication of System No. 15 were discussed in Section IV. Using the value of 0.026 obtained for the shield emittance, ϵ , and the relation $q/A = \delta (T_o^4 - T_i^4) \epsilon / 2n$ where T_o (540 °R) and T_i (140 °R) are the inner and outer boundary temperatures of the MLI, and n is the number of shields, the theoretical heat flux value for this MLI is 0.19 Btu/hr-ft² (0.060 mw/cm²). Further, based upon results obtained with System No. 11 (a similar five-shield system), solid-conduction heat flux for a five-shield system was estimated to be 0.12 Btu/hr-ft² (0.038 mw/cm²). The solid-conduction value for the current system is estimated at half (0.06 Btu/hr-ft²) that of System No. 11, because the current system has twice as many spacers. It is assumed that the gas conduction flux is zero. Thus, the predicted system heat flux for the boundary temperatures identified above is 0.25 Btu/hr-ft² (0.079 mw/cm²).

3. Flat Plate Thermal Conductivity Apparatus Results

A 12-inch-diameter sample of 10-shield multilayer insulation composed of the same materials as insulation System No. 15 was tested in the ADL Model-12 Calorimeter. Each radiation shield of the sample was cut from the same sheet material which was used for the corresponding shield in Insulation System No.15.

The initial flux measurement was made with zero compression applied to the sample. The compressive load was then successively increased in four steps until a pressure of 15 psi was exerted on the sample boundaries. The compressive load was next decreased to zero in four successive steps. This pressure cycle was repeated in fewer steps a second time.

The heat flow data obtained with the conductivity apparatus are summarized in Table 6. The variation of heat flux as a function of sample compression is shown in Figure 20. The two zero-load values are 0.25 and 0.26 Btu/hr-ft² (0.079 and 0.082 mw/cm²). The heat fluxes at different degrees of compression, both in increasing and decreasing pressure schedule, appear to be reproduced and thus show little hysteresis except for a single test point. The heat flux values obtained in the second pressure cycle closely reproduce those obtained in the first cycle. The heat flux is degraded by a factor of 120 when the values obtained at 15 psi are compared with those obtained at zero psi.

There was excellent agreement between the predicted system heat flux and that obtained with the conductivity apparatus; 0.25 Btu/hr-ft² in both cases. The system has good flux-load characteristics; an increase in compressive load from 0.0003 to 0.01 psi, a factor of 30, results in a heat leak increase which is less than a factor of 2. Repeated loadings produce little variation in the heat flux results at each compression level. The thermal-compression properties obtained with this system are of the type that are very desirable in a MLI system.

4. Tank Calorimeter Test Results

Four heat flux tests were performed on System No. 15 with the space simulation chamber described previously. In each test the tank calorimeter was filled with liquid nitrogen. In the first test, the chamber baffles used to simulate external insulation environment were maintained at 540°R. In the third test these baffles were cooled with liquid nitrogen and maintained at 140°R. In the second and fourth tests, one baffle was maintained at 140°R and the other at 540°R.

The purpose of this test series was to obtain overall and sectional heat flux measurements for System No. 15 prior to the attachment of the pipe penetration to the calorimeter and insulation system. The sectional heat flow was obtained through use of the specially designed split-baffle system built into the space simulation chamber. The data for the four tests performed in this series are summarized in Table 7.

The heat flux measured with all warm baffles (540°R) in Test IV-2A was 0.27 Btu/hr-ft² (0.085 mw/cm²). This compares with the value of 0.25 Btu/hr-ft² (0.079 mw/cm²), predicted both from the emissometer and flat plate conductivity apparatus.

The shield temperature distribution during this test is shown in Figure 21. The temperature measured at the side ("b" location) and bottom ("c" location) of the insulation system show a close approach to the theoretical values. On the other hand, the temperatures at the top ("a" location) deviate significantly from the theoretical values; this is the result of two effects: the first is due to the proximity of the calorimeter neck, and the second is due to compression of the top portion of the insulation by the weight of the lower portions which it supports. The two tests performed with the split baffles (Tests IV-2B and IV-2D) indicate that compression effects play a strong role in determining the top shield temperature.

Test IV-2B, performed with a warm (540°R) lower baffle and a cold (140°R) upper baffle, showed a heat flux for the lower insulation half of 0.20 Btu/hr-ft² (0.063 mw/cm²). This value was almost identical to the 0.19 Btu/hr-ft² computed previously as the theoretical heat flux for the system. It is possible for the side and bottom of the MLI to give very close to theoretical performance, because the shields and spacers hang free of one another. Close approximation of the temperature distribution in the shields to the theoretical values (see Figure 21) at the side and bottom are also evidence that the shields reduced heat flux to near-theoretical rates.

The results obtained with Test IV-2D, performed with a warm (540°R) upper baffle and a cold (140°R) lower baffle, showed a heat flux in the upper half of the insulation 70% greater than the theoretical value. The temperature distributions in the top part of the insulation were identical for Tests IV-2A and IV-2D, indicating that the thermal performance of this portion of the insulation was also identical for the two tests.

Referring again to Table 7, the two tests using the warm-cold baffle temperature conditions (Tests IV-2B and -2D) produced heat flows of 3.8 (1.1 watt) and 6.1 Btu/hr (1.8 watt). In these tests, heat flows through each half of the insulation were measured separately. The total of these values should equal the heat flow obtained when both halves were measured together. The two results did, in fact, closely agree; the conditions prevailing in Tests IV-2B and -2D produced a total heat flow of 9.9 Btu/hr (2.9 watt), which was only 4% less than the 10.3 Btu/hr (3.0 watt) obtained for the calorimeter insulation with all baffles at 540°R (Test IV-2A).

The thermocouples at the side of the insulation system (location "b") did not show relevant temperature distributions in the split-baffle tests (Figures 22 and 23). This location on the insulation views both the warm and cold baffles simultaneously. The measured temperatures therefore lay between those obtained for the upper and lower temperature measurement positions. The exact location of the thermocouples relative to the location of the baffle split determined the value and distribution of the measured temperature.

The cold baffle test (IV-2C) was performed to detect any significant extraneous heat leaks to the calorimeter. It also served to establish the performance of the calorimeter at low heat leak values. The measured heat flow was 0.2 Btu/hr for a 28-hour period; this is no more than 5% below usual values measured with the calorimeter. However, from Figure 24 it is evident that at the time of measurement the insulation was still in the process of achieving temperature equilibrium, though the approach to that equilibrium was very close: average shield temperatures were about 12°R from liquid nitrogen at 140°R.

5. Conclusions

- (a) In a carefully constructed MLI system, thermal performance can be predicted from shield emittance and flat-plate thermal conductivity data. The three instruments used to establish the performance of System No. 15 were in excellent agreement. In addition, the results obtained with the current 10-shield system were in good agreement with the extrapolated results obtained with data from a five-shield system.
- (b) The heat flux in the upper half of System No. 15 tank insulation was 70% larger than that in the lower half, due in part to the larger compression load on the upper insulation half. This sort of degradation must be expected in all terrestrial tests performed with MLI systems. Heat fluxes through well constructed MLI systems will undoubtedly be lower in the absence of a gravitational field than when measured in ground tests.
- (c) Measurements of the temperature distribution in any MLI system produce excellent indication as to whether the thermal performance approaches the theoretical value or is degraded. This was confirmed by measurements performed with MLI systems in the previous contract⁽²⁾ and re-confirmed by shield temperature measurements made with the current system.

TABLE 3
INSULATION SYSTEM NO. 15, SPECIFICATION

<u>Description</u>	Ten radiation shields of polyester film metallized on both sides with aluminum and six spacers, each consisting of two layers of silk netting. The multilayer is applied to Calorimeter No. 2.
--------------------	--

Shield

Material	DuPont, Type S polyester film
Thickness	Film 0.00025 inch; aluminum coatings, 320 to 630 angstroms
Weight	0.0018 lb/cu ft
Emittance	0.026 average for all shields
Geometry	Sides: gores 3 inches wide Head: full circles
Joints	All seams overlapped and contacting except for gored elements, which are butted.

Spacer

Material	Two layers of silk netting, Jordan Marsh No. 5517
Thickness	0.007 inch (0.0035 inch per silk layer)
Weight	0.0024 lb/sq ft
Density	4.1 lb/cu ft
Open Area	84%
Geometry	Side, gored edges of side sheet; head, full circles
Joints	All joints butted

Boundary Emittance

Calorimeter Tank	0.86 at -320°F
Chamber Baffles	0.93 at 80°F

System Properties

Surface Area	38.14 sq ft
Weight	0.0444 lb/sq ft

TABLE 4
SPECIFICATION USED FOR PURCHASE OF ALUMINIZED MYLAR
FOR SYSTEM NO. 15 RADIATION SHIELDS

Aluminum of a 99.0% minimum purity is to be vapor deposited on each side of 1/4-mil DuPont Type "S" polyester film. Each aluminum coating is to have a thickness between 600 and 1000 angstroms and is to be applied in a single pass at the most rapid coating rate possible in the vendor's equipment to produce a uniformly thick coating with a bright luster. The coatings are to be free of windows (unaluminized areas) except for openings, such as pinholes present in the polyester film. The coated film is to have no creases, and the wrinkling of the film is to be no greater in kind and degree than that found in the original mill-wound roll. Until the coatings are complete, all wind-rewind operations are to be performed in the coating chamber at vacuum to prevent partial coatings due to dust accumulation on the polyester film. The coated film is to be 48 inches wide.

The coating thickness is to be determined by the electrical resistance method using a resistance head supplied by Arthur D. Little, Inc. The resistance head consists of a 2 x 3 inch Micarta block with copper knife edges attached to the long side of the Micarta block. The measurement is to be made on a 2 x 3 inch sample with the measuring head set across the sample to produce a 2 x 2 inch measuring square. The coating resistance is to be measured using an impedance bridge such as an ESI Model 250. The thickness in angstroms will be computed from the relation $285/R$ where R is the measured electrical resistance of the coating in ohms. The limits of the coating thickness are 600 angstroms (0.475 ohm) and 1000 angstroms (0.285 ohm).

TABLE 5

EMITTANCE OF RADIATION SHIELDS IN INSULATION SYSTEM NO. 15

System 15 Shield No.	Shield (1) Location	Roll (2) Side	Emittance Value	Resistance Ω /square	Thickness, $\frac{1}{8}$	Distance into the Roll, ft	Distance Left to Right, ft	Sample No.	Emittance Test No.
1	H	A	0.025	0.543	525	12	1	373	707
	H	B		0.682	417	12	2	374	
	S	A	0.020	0.610	467	25	1	379	712
2	S	B		0.595	478	25	2	380	
	H	A		0.626	455	34	1	381	
	H	B	0.028	0.671	425	34	2	382	715
3	S	A		0.482	590	25	3	383	
	S	B	0.023	0.547	522	25	4	384	718
	H	A		0.511	556	42	1	385	
4	H	B	0.029	0.609	469	42	2	386	719
	S	A	0.024	0.479	595	55	1	387	720
	S	B	0.031	0.519	550	55	2	388	721
5	H	A	0.026	0.565	505	65	1	395	722
	H	B		0.645	442	65	2	390	
	S	A		0.584	489	55	3	391	
6	S	B	0.031	0.605	471	55	4	392	723
	H	A		0.549	520	73	2	393	
	H	B		0.620	460	73	3	394	
7	S	A	0.021	0.546	522	86	1	395	724
	S	B	0.028	0.596	478	86	2	396	725
	H	A		0.567	503	96	1	397	
8	H	B		0.699	418	96	2	398	
	S	A	0.021	0.500	570	86	3	399	726
	S	B	0.034	0.556	513	86	4	400	727
9	H	A		0.552	516	104	3	402	
	H	B		0.720	396	104	1	403	
	S	A	0.023	0.447	637	118	2	404	728
10	S	B	0.021	0.627	455	118	1	405	729
	H	A		0.650	438	126	1	406	
	H	B		0.527	540	126	3	407	
11	S	A	0.024	0.690	413	118	4	408	730
	S	B	0.034	0.683	417	118	3	409	731
	H	A		0.589	484	126	1	410	
12	H	B		0.556	512	126	3	411	
	S	A	0.022	0.540	527	140	1	412	732
	S	B	0.033	0.732	390	140	2	413	733
13	H	A		0.642	444	148	3	414	
	H	B		0.647	440	148	1	415	
	S	A	0.024	0.553	515	140	3	416	734
14	S	B	0.030	0.630	452	140	4	417	735

(1) H - Sheet used for tank ends; S - Sheet used for tank side.

(2) A - Inside of roll; B - Outside of roll.

TABLE 6

SYSTEM NO. 15, INSULATION SAMPLE 3093
EFFECT OF COMPRESSIVE LOADS ON HEAT FLOW

Sample No. 3093 (Corresponds to tank insulation system No. 15)

Sample Description: Ten radiation shields of 0.00025" thick x 11" diameter polyester film, aluminum-coated on both sides and twenty-two 12" diameter, 0.003" thick silk netting.
Shields are taken from same roll used for fabrication of System No. 15.

Sample Weight: 0.031 lb

Sample Chamber Pressure: Below 8×10^{-5} torr

Warm and Cold Plate Emittance: 0.9

Warm Plate Temperature: $80 \pm 3^\circ\text{F}$

Cold Plate Temperature: -320°F

Mechanical Compression, psi	Heat Flux, Btu/hr-sq ft	Sample Thickness, in.	Test Date	Test No.
0	0.25	0.482	3/22/67	b
0.01	0.33	0.245	3/23/67	d
0.1	1.41	0.113	3/24/67	e
5.0	14.5	0.067	3/27/67	f
15	33.5	0.046	3/27/67	g
5.0	22.4	0.056	3/27/67	h
----	13.1	0.066	3/27/67	i
0.1	1.23	0.104	3/28/67	j
0.01	0.76	0.117	3/29/67	k
0	0.26	0.227	3/29/67	l
0.01	0.35	0.195	3/30/67	m
0.10	1.54	0.108	3/31/67	n
15	32.8	0.056	3/31/67	o
0.01	0.30	0.193	3/31/67	p

TABLE 7

DATA SUMMARY, INSULATION SYSTEM NO. 15, TEST SERIES IV-2, CALORIMETER NO. 1

Insulation System:		Ten radiation shields of polyester film aluminized on both sides and eleven spacers, each consisting of two layers of silk netting									
Test Facility:		Arthur D. Little, Inc.									
Penetrations or Gaps:		None									
Tank Surface Area:		38.14 sq ft									
Tank Liquid:		Liquid nitrogen									
Boundary Conditions:		Temperature as noted below; emittance as follows: 540°R surfaces are 0.93 and 140°R surfaces are 0.86									
Test No.	Date & Time	Date & Time	Test Duration hr	Avg. Vac. Torr	Guard Liq. Temp. °R	Upper Baffle, °R	Lower Baffle, °R	Total Heat Flow, Btu/hr (watt)	Measured Heat Flux, Btu/hr-ft ² (mw/cm ²)		
IV-2A	4-16-67/1430	4-20-67/0830	90	8 x 10 ⁻⁶	142	540	540	10.3(3.0)	0.27 (0.085)		
IV-2B	4-25-67/0830	4-26-67/1630	32	6 x 10 ⁻⁷	142	140	540	3.8(1.1)	0.2 (0.063)		
IV-2C	4-28-67/0800	4-29-67/1200	28	4 x 10 ⁻⁷	142	140	140	0.2(0.06)	0.005(0.0016)		
IV-2D	5-1-67/1430	5-3-67/1630	48	1.5 x 10 ⁻⁶	142	540	140	6.1(1.8)	0.32 (0.10)		

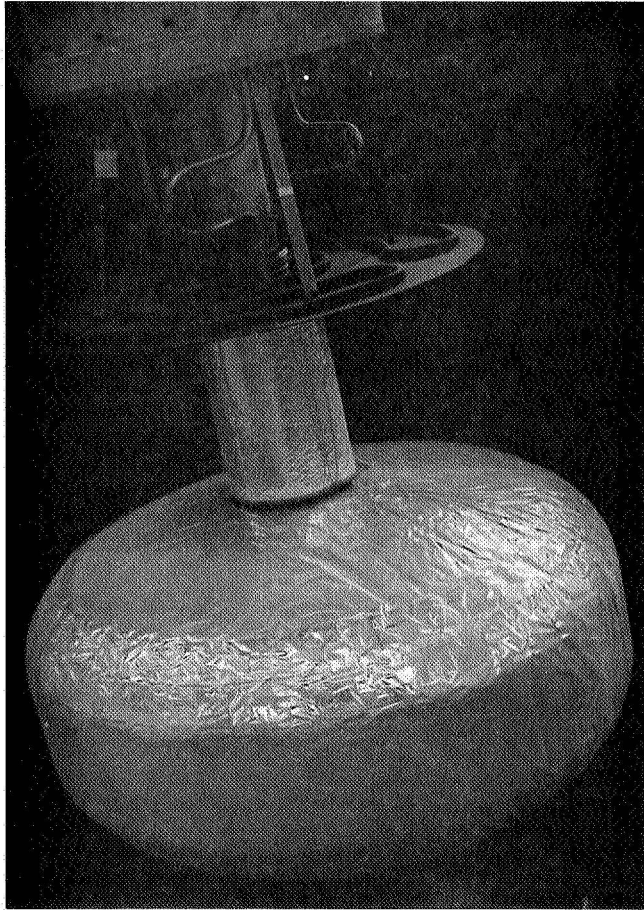
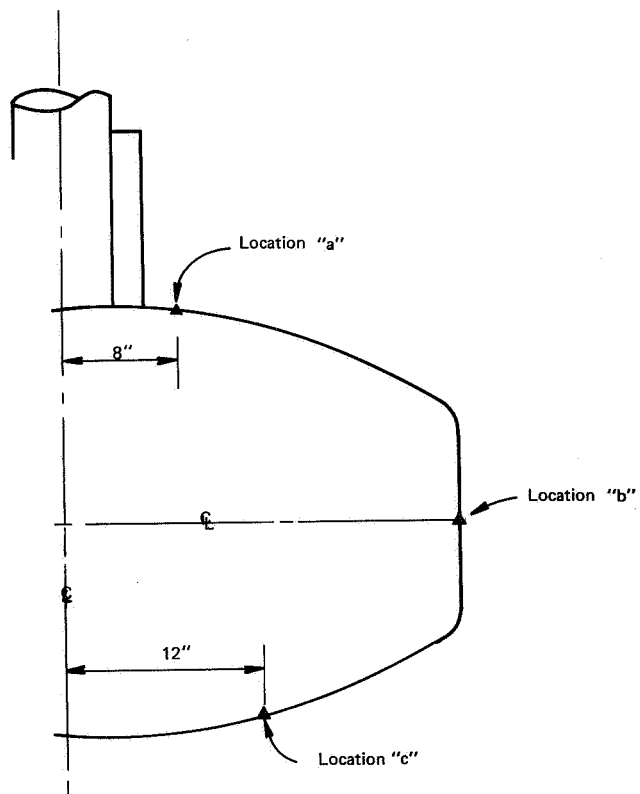


FIGURE 18 INSULATION SYSTEM 15



Thermocouples are located on
Shields 1, 2, 4, 7 and 10
(System No. 15, 16, 17 and 18)

FIGURE 19 CALORIMETER TANK THERMOCOUPLE LOCATION

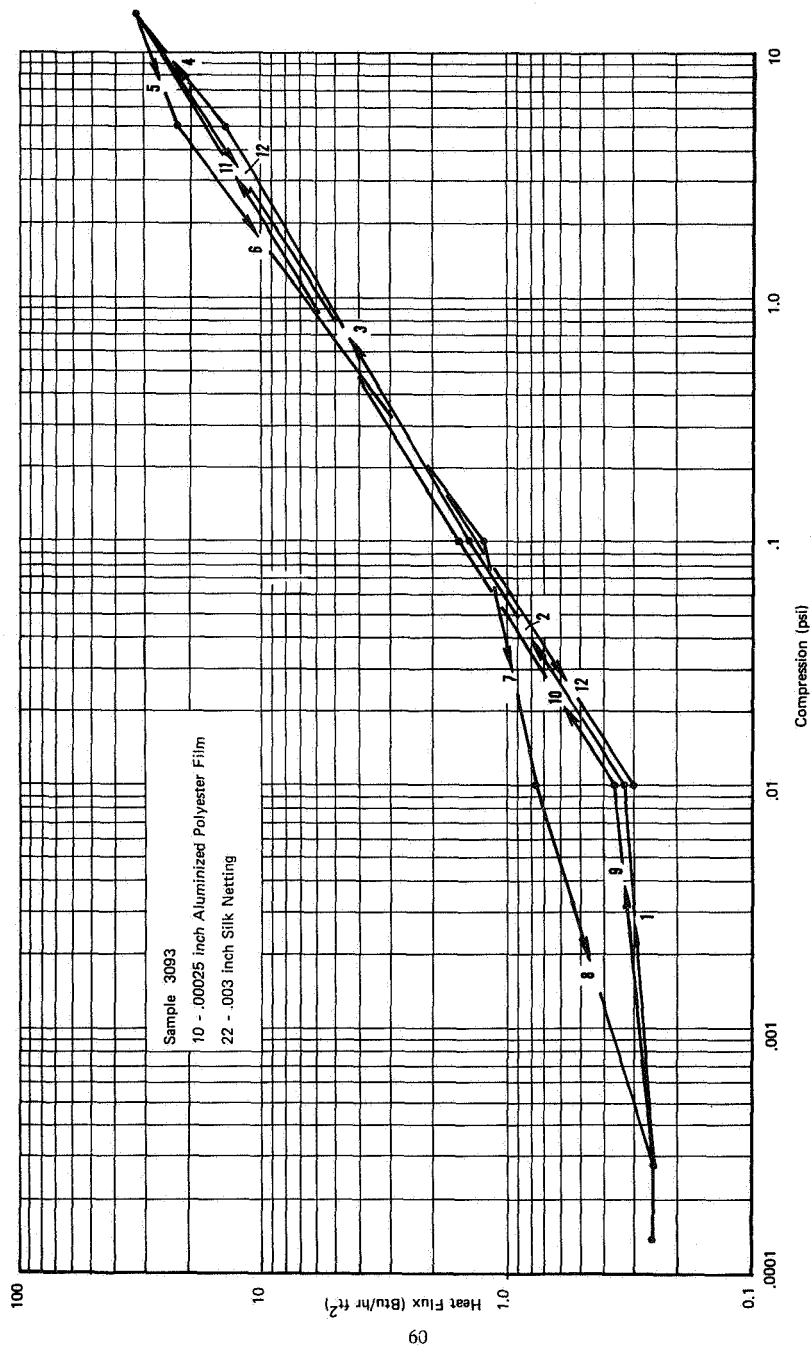


FIGURE 20 THERMAL CONDUCTIVITY APPARATUS, HEAT FLUX AS A FUNCTION OF COMPRESSIVE LOAD, SAMPLE 3093

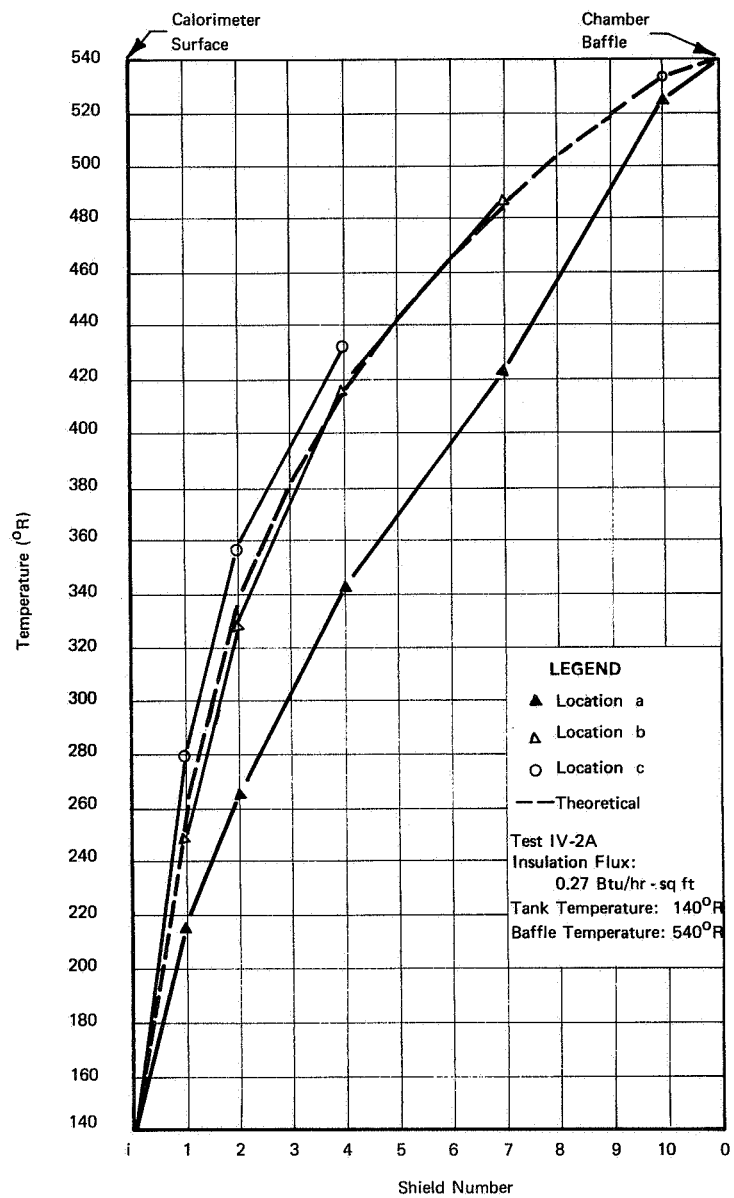


FIGURE 21 INSULATION SYSTEM NO. 15, SHIELD TEMPERATURE DISTRIBUTION, TEST IV-2A

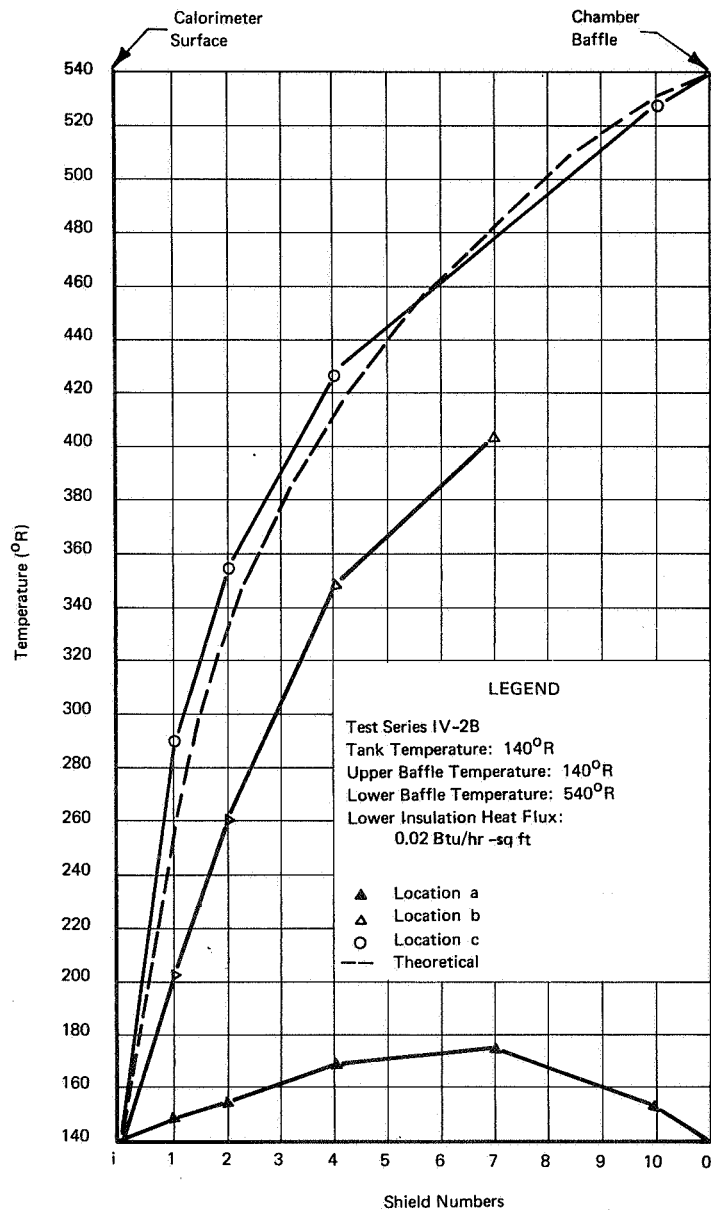


FIGURE 22 INSULATION SYSTEM NO. 15, SHIELD TEMPERATURE DISTRIBUTION, TEST IV-2B

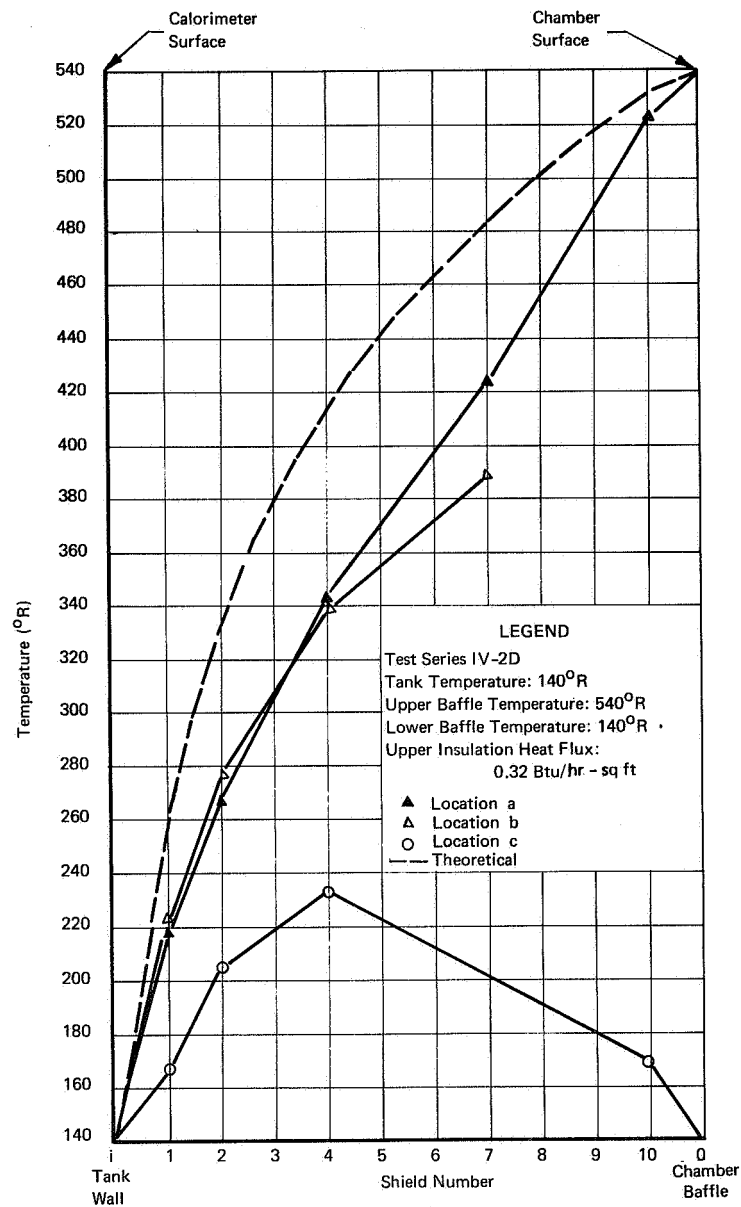


FIGURE 23 INSULATION NO. 15, SHIELD TEMPERATURE DISTRIBUTION, TEST IV-2D

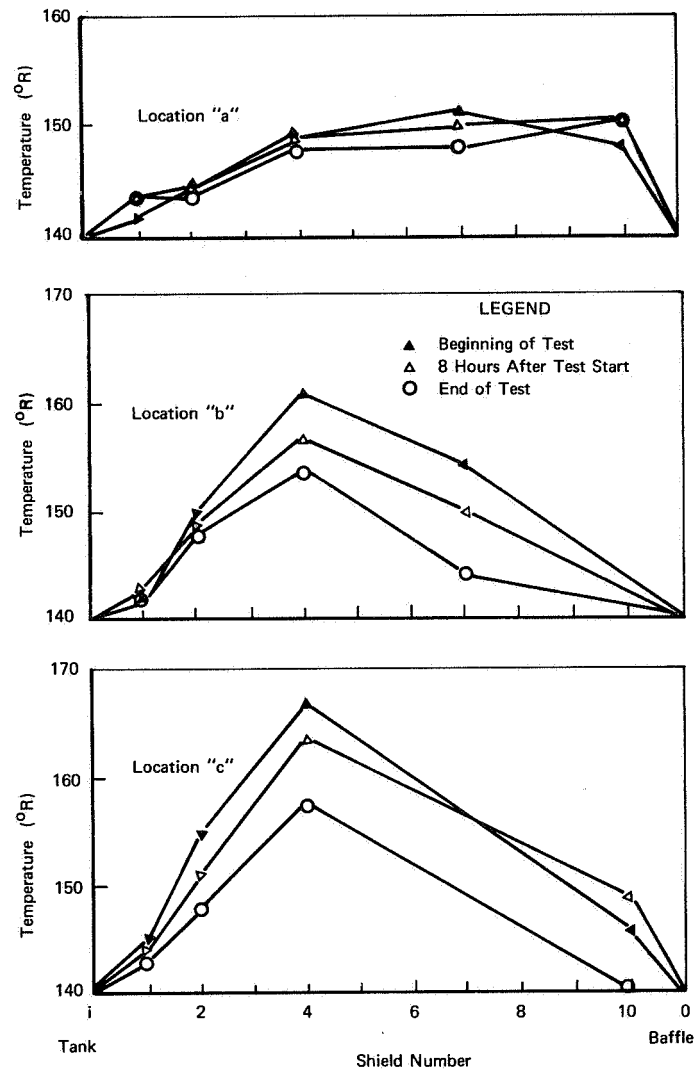


FIGURE 24 SHIELD TEMPERATURE DISTRIBUTION DURING TEST IV-2C

- (d) The technique of measuring the heat flow in a portion of an insulation system through the use of split baffles was confirmed. Separate measurements made of the upper and lower halves of System No. 15 MLI, when added together, are in good agreement with the results obtained from a single measurement of overall performance.

B. INSULATION SYSTEM NO. 16

Insulation System No. 16 consisted of a 10-shield MLI placed on the tank calorimeter and a pipe penetration, also completely insulated with MLI, placed at the bottom center of the tank. This system was obtained by modification of System No. 15, described in the preceding section, which was used to establish the thermal performance of the 10-shield MLI with no penetrations. A portion of the insulation at the bottom center of the tank was cut away, the penetration was installed, and the insulation was applied to the penetration and mated to the tank insulation.

Eight tests were performed on the system for various combinations of chamber baffle and penetration temperatures to obtain the temperature distribution in the penetration wall and the heat flow to the calorimeter due to the presence of the penetration.

1. Description of Penetration

a. "A" Cap Details

The penetration (see Figure 25) was a 17-inch stainless steel tube with a diameter of 6 inches and a wall thickness of 0.027 inch. The cold end of the penetration was formed with a copper disc $\frac{1}{4}$ -inch thick which was silver-soldered inside the end of the tube. This plate was used to attach the penetration to the outside of the tank bottom. A stainless steel stud was welded to the stainless steel section in the tank bottom, and was used for bolting and thermally connecting the penetration to the tank. A low thermal resistance between the penetration and tank was obtained through the use of copper and indium washers.

A threaded copper ring was silver-soldered to the inside of the stainless steel tube at its free end. An aluminum cap was threaded into the copper ring so that the interior surfaces of the penetration would be accessible for altering its emittance and reflectance characteristics. An electrical heater with a resistance of about 2000 ohms was bonded to the aluminum cap. During tests, this heater was used to maintain the temperature at the free end of the penetration at predetermined values.

b. Insulation

The insulation on the penetration consisted of ten radiation shields and ten foam spacers. The shields were aluminized Mylar and identical to those used on the tank calorimeter. A hole 12 inches in diameter was cut into the tank as shown in Figures 25 and 26, the penetration was bolted in place as shown in Figure 27, and the insulation was built up a layer at a time.

The cutout portions of the tank insulation were replaced by annular rings of shield and spacer which overlapped the tank insulation by 2 inches. Each layer of the penetration insulation was formed with a cylinder of shield and spacer material. The joint between the tank and penetration shield was formed by overlapping the legs of aluminum-foil angles so that no radiation holes could develop at the corner from slight displacements of the tank and/or penetration shields.

The shield and spacer cylinders on the penetration were extended beyond the free end of the penetration. The shield material was folded over the end of the penetration and interleaved with circles of shield material cut to the appropriate diameter. When the calorimeter was mounted in the space simulation chamber, the penetration was below the tank; there was thus a tendency for the insulation to slip down on the penetration. On System No. 16 this was controlled by a rigid disc placed at the outside of the insulation but tied to the aluminum cap with a stainless steel wire. This disc supported the insulation at the outside edges and prevented movement of the wall shields on the penetration. The completed insulation system at the penetration is shown in Figure 28, and the entire System No. 16 is shown in Figure 29.

c. Surface Emittance

Under the previous contract, experimental data were obtained with the simulated pipe penetration. At that time, the interior surfaces were coated with 3M black to produce a surface emittance of approximately 0.9. From our analysis it was determined that the transport of heat from the warm end to the cold end of the penetration by internal radiation could be significantly reduced by lowering the emittance of its interior surfaces. A surface emittance of 0.025 could be obtained by lining the penetration with aluminized Mylar; however, because of their mirror-like nature, such surfaces reflect the incident radiation in a specular manner as opposed to diffuse (Lambertian or cosine) reflection. The mathematical model used in conjunction with this program could solve the case involving the diffuse surface but not the specular. Thus, it was necessary to produce a surface which was low in emittance and diffuse in nature. This led to the technique of sandblasting the surfaces.

As indicated in Figure 25, the penetration had a stainless steel side wall, copper cold end, and predominately aluminum warm end. Samples 2.5 inches in diameter and 0.1 inch thick were prepared from each of these materials. These were then sandblasted like the penetration surfaces and placed in the emissometer, where the total hemispherical emittance of each was measured. The results obtained are as follows:

<u>Material</u>	<u>Surface Condition</u>	<u>Emittance</u>
Aluminum	Sandblasted	0.34
Stainless Steel	Sandblasted	0.41
Stainless Steel	Polished	0.104
Copper	Sandblasted	0.27
Copper	Polished	0.014

From the visual appearance of the sandblasted surfaces, the assumption was made that each would have the property of a diffuse reflector. As the experimental work progressed, certain anomalies arose between the experimental and analytical results which required confirmation of this property. Accordingly, a goniometer developed by Arthur D. Little, Inc., was used in a subsequent series of experiments to make this confirmation. The results, which are summarized in Appendix A, indicate that the surfaces are probably 80-90% diffuse.

d. Thermocouple Locations

Temperatures in the test penetration were measured with a 40-gauge copper-constantan thermocouples. These were placed on the wall of the penetration, on selected shields of the penetration insulation, and on the adjacent tank insulation at the locations shown in Figure 25. Each wall thermocouple junction was set into a hole drilled into the penetration and was held in place with an epoxy adhesive. All shield thermocouples were attached in the same manner as in Insulation System No. 15.

The leads from the penetration wall and shield thermocouples were carried along the penetration and run through the MLI in a bundle at the free end, as shown in Figure 28. The location of the exiting leads was selected to have minimum effect on the computed value of the penetration heat flow.

2. Method of Determining Penetration Heat Flow

The heat flow to the calorimeter produced by the presence of the penetration was determined experimentally from two boil-off test measurements. In

one of these, the effect of the penetration was not present; all the tests performed in Test Series IV-2 were of this type, as was Test IV-3H. In all the tests of the IV-3 series except Test 3H, the effect of the penetration was present. Thus, when the baffle conditions of the control test (i.e., without penetration effect) were matched to those of the penetration test, the difference in the calorimeter-measured heat flows obtained was due to the penetration alone.

3. Test Results

a. Tests and Conditions

Eight tests were performed with System No. 16 in the IV-3 test series. They were arranged in a sequence that resulted in the minimum expenditure of test time. The relevant test data were obtained after the calorimeter system had achieved quasi steady-state conditions; this equilibrium was achieved in a period ranging from one to ten days. The duration of the steady-state test periods ranged from 24 to 72 hours. The space simulation chamber vacuum at the time these measurements were taken was 5×10^{-7} torr or less. The baffles were maintained at their required temperature levels of 140°R and/or 540°R, generally within 2°F both before and during the measurement period.

The prevailing conditions of test and the heat flow determined from the calorimeter boil-off are presented in Table 8. A tabulation of the steady-state temperatures measured at the penetration wall and within the penetration and tank multilayer insulation are presented in Table 9.

(1) Comparison of System No. 15 and System No. 16. The first comparisons to be made between Systems 15 and 16 are for the same boundary conditions, but with no power applied to the penetration heater. With all baffles at 540°R, System No. 16 produced a heat flow of 11.9 Btu/hr (3.5 watt) (Test IV-3A); System No. 15 showed a heat flow of 10.3 Btu/hr (3.0 watt) (Test IV-2A). A portion of the difference is accounted for by the added area (2.55 ft²) of the penetration surface in System No. 16. From the average heat flux computed for the insulation in Test IV 2A of 0.27 Btu/hr-ft² (0.085 mw/cm²), an added heat flow 0.7 Btu/hr (0.2 watt) results.

The remaining heat flow difference of 0.9 Btu/hr (0.26 watt) between Tests IV-2A and IV-3A can be accounted for by the thermocouple lead wires and stainless steel wires used on the cap to hold the insulation in place. Ten copper-constantan thermocouples and four copper heater leads (all wires 0.0035 inch diameter) penetrated the MLI to the pipe penetration from the 540°R environment (see Figure 28). These wires caused an added heat flow to the system of 1.1 Btu/hr (0.32 watt), assuming that the wire temperature at the point where it left the penetration was 540°R. The stainless steel pins allowed a computed heat flow of 0.2 Btu/hr (0.06 watt). An added 0.14 Btu/hr (0.04 watt) was computed for the partial heat short represented by the three thermocouples attached

to Shield No. 2 on the penetration insulation. Because the thermocouple wire and stainless pin temperatures must have been less than 540°R at the exit point, the shorting effect must have been smaller than the computed results. Thus, it appears that 0.9 Btu/hr (0.26 watt) was the result of the shorting effect of the penetrating pins and lead wires.

The penetration area and thermocouple effect are associated only with the lower half of the calorimeter. Thus, measurements of the thermal performance of the upper halves of Systems 15 and 16 should have been unaffected by any modification performed on the lower half. This was confirmed by the results obtained in Tests IV-2D and IV-3H, in which the upper and lower chamber baffles were maintained at 540°R and 140°R, respectively: both tests produced a heat flow of 6.1 Btu/hr (1.8 watts).

The last comparison between these two systems is that in which all chamber baffles were maintained 140°R. System No. 15 produced a heat flow of 0.20 Btu/hr (0.06 watt) and System No. 16 a value of -0.27 Btu/hr (-0.08 watt). The negative value is possible but improbable. However, either value is quite small and within the uncertainty of the measurement.

(2) System No. 16, Heated Penetration Results.

(a) Conditions - Five tests were performed with a heated penetration. Power to the electric heater was adjusted in each test to maintain the cap temperature at a constant value. Nominal cap temperatures were 400, 540, and 750°R. One test was performed at each cap temperature with 140°R environmental temperature (chamber baffles cooled with liquid nitrogen). In addition, one test was performed at each of the two latter cap temperatures with a 540°R environment.

(b) Comparison of Measured and Predicted Heat Flows - The measured penetration heat flows are presented in Table 10. Those determined through use of the mathematical model are presented in Table 11. (Table 11a shows the predicted values in Btu/hr units, Table 11b shows the same in watts.) Measured and predicted values are compared in Table 10 and Figure 30. There is good agreement between the two at end-cap temperatures of 540°R and below. At the 750°R nominal temperature level, predicted values are about 23% lower than the measured values.

The magnitude of the heat flow at all end temperatures, particularly at the 750°R condition, is too large to permit storage of liquid hydrogen during space missions of a few months or longer. For example, a 540°R end cap would produce a liquid boil-off of 300 pounds over a period of six months; this is clearly excessive. An acceptable value for the penetration heat leak is probably one-tenth of the current values and can be achieved only by redesign of the penetration to reduce the conductance of the wall and decrease its surface emittance.

(c) Heat Flow at 750°R End-cap Temperature - The heat flow predicted for the conditions prevailing in Tests IV-3C and -3D for a 745°R end-cap temperature is 35 Btu/hr (10.2 watt). From Table 11, it is evident that the wall conduction and radiation heat flows predominate. For example, there is an equal split in the two at the warm end; at the cold end, wall conduction is six times larger than the radiation heat flow. The predicted heat loss from the penetration through the MLI is 0.75 Btu/hr (0.22 watt); compared with losses for the two other transport modes, this value is not significant.

The measured heat flows for these conditions are approximately 10 Btu/hr (3 watt) larger than the predicted values. The experimental data and conditions account for only a portion of this difference. In Section IV of this report, the values of influence coefficients developed to account for the uncertainty in wall conductivity, interior wall emittance and wall thickness predict a maximum uncertainty of 3.4 Btu per hr (1.0 watt). However, it is unlikely that the systematic experimental errors would have the same sign and their maximum magnitude.

In many instances, the penetration end temperatures used in the computer runs differ from the temperatures prevailing in the experimental runs. In addition, measurements of end temperatures have an associated error which may be as large as 5°R. From the temperature-power curves, presented in Figure 30, the approximate power rate in Btu/hr °R and watt/°R were computed. These are tabulated below for the four temperature levels of interest here.

End-cap Temperature (°R)	Experimental		Mathematical Model	
	(Btu/hr°R)	(watt/°R)	(Btu/hr°R)	(watt/°R)
200	0.017	(0.005)	0.017	(0.005)
400	0.034	(0.01)	0.034	(0.01)
540	0.058	(0.017)	0.068	(0.02)
750	0.205	(0.06)	0.170	(0.05)

A five-degree difference in end-cap temperatures at the 750°F level should result in about 1 Btu/hr (0.3 watt) difference in power level. Tests IV-3C and -3D were performed at 746°R and 745°R end-cap temperatures respectively, while corresponding computer runs 108 and 109 were performed at end-cap temperatures of 750°R. On the basis of the above results, the predicted electrical power should be about 1 Btu/hr (0.3 watt) above the experimental. However, this effect is in the opposite direction of the observed difference and would produce a wider discrepancy between measured and predicted heat flows than now exists.

b. Penetration Temperatures

(1) Wall Temperatures. The measured and predicted temperatures for the penetration wall are in good agreement, as shown in Figure 31, for

nominal end-cap temperatures of 400, 540, and 750°R. In each case electrical power was dissipated at the free end of the cap to develop a penetration heat flow. In Test IV-3G no power was dissipated at the free end. However, there was a temperature rise along the penetration, which was associated with the thermocouple leads and stainless steel support pins that penetrated the multilayer insulation, as discussed previously.

(2) Penetration MLI Temperatures. Temperatures were also measured in the MLI at the penetration and are shown in Figures 32 and 33 for a 750°R end-cap temperature and 540°R and 140°R environmental temperatures, respectively. Because some thermocouple leads failed, the graphing of the temperature field within the MLI is incomplete. From each of these graphs the temperature distribution was replotted as a function of shield number for various distances (X/L) along the penetration. This is shown in Figures 34 and 35.

(a) Gas Conduction Effects - The approximate temperature distributions in the penetration MLI for 540°R environment and 746°R end-cap temperature are shown in Figure 34. For comparison, the theoretical temperature distributions are shown that would prevail in the MLI for the given boundary temperatures, zero axial heat flow, and pure radiation heat flow in the perpendicular direction. At X/L = 1.0 there is a severe temperature gradient, particularly in the space between the wall and shield No. 2, which is grossly different from the theoretical distribution and which indicates the presence of conduction. The same effect is present at the same X/L and shield numbers in Figure 35, which was prepared from Test IV-3D data. This effect is believed to be caused by the foam spacer used in the penetration MLI. The foam manufacturer has indicated that two components of the foam could cause serious outgassing at 750°R.

The gas conduction effects were localized at the warm end. It is quite evident from the temperature distributions at X/L less than 0.9 that the heat transfer is mainly radiative. Thus, gases boiled off from the spacer very likely recondensed a short distance away on the cooler shield and spacer surfaces. Scott foam should not be used at temperatures above 550°R for MLI spacers.

(b) Heat Shunt Effect - The temperatures shown in Figure 35 at X/L = 0.7, 0.6, and 0.5 indicate an apparent channeling of heat in shields 1, 2, and 3. It is estimated that the maximum magnitude of this heat flow is of the order of 0.25 Btu/hr (0.073 watt), considering gas conduction in ten foam spacers and solid conduction in the aluminum coatings of ten shields. Even if all this heat is shunted between the warm and cold ends of the penetration, it is still a factor of ten less than the difference between the experimental and predicted results.

c. Summary of Results

We have found that the predominant mode of heat transfer from the warm to the cold end of the penetration is by the mechanisms of wall conduction

TABLE 8

DATA SUMMARY, INSULATION SYSTEM NO. 16, TEST SERIES IV-3, CALORIMETER NO. 1

Insulation System: Ten radiation shields of polyester film aluminized on both sides and eleven spacers, each consisting of two layers of silk netting (System No. 15)

Test Facility: Arthur D. Little, Inc.

Penetrations or Gaps: 6-inch diameter pipe penetration ("A" Cap)

Tank Surface Area: 40.6 ft² based on surface area of the calorimeter tank and penetration

Tank Liquid: Liquid nitrogen

Temperature of Neck Guard: 142°R

Test No.	-----Data Period----- Date & Time to Date & Time		Test Duration, hr	Avg. Vac., Torr	Temp. of Penetr., Warm, End °R		Upper Baffle, °R	Lower Baffle, °R	Total Heat Flow, Btu/hr (watts)	Power to Penetr. End, Btu/hr (watts)
IV-3A	5-26-67/1630	5-28-67/1500	46.5	5 x 10 ⁻⁷	208		540	540	11.9 (3.5)	0 (0)
IV-3B	6-2-67/1630	6-3-67/1230	28.0	4 x 10 ⁻⁷	534		540	540	24.1 (7.1)	13.6 (4.0)
IV-3C	7-5-67/1630	7-8-67/0830	64.0	2 x 10 ⁻⁷	746		540	540	57.3 (16.8)	44.3 (13.0)
IV-3D	6-6-67/2300	6-8-67/0830	33.5	2 x 10 ⁻⁷	745		140	140	43.3 (12.7)	44.6 (13.1)
IV-3E	6-10-67/2200	6-11-67/2200	24.0	1 x 10 ⁻⁷	539		140	140	14.3 (4.2)	15.4 (4.5)
IV-3F	6-13-67/1630	6-15-67/1430	46.0	1 x 10 ⁻⁷	409		140	140	6.5 (1.9)	6.8 (2.0)
IV-3G	6-22-67/0830	6-24-67/1330	53.0	1 x 10 ⁻⁷	145		140	140	-0.27 (-0.08)	0 (0)
IV-3H	6-27-67/0830	6-30-67/0830	72.0	2 x 10 ⁻⁷	142		540	140	6.1 (1.8)	0 (0)

TABLE 9
TEMPERATURE DISTRIBUTION, INSULATION SYSTEM NO. 16.

TEST SERIES IV-3

Temperature (°R)

<u>T.C. No.</u>	<u>IV-3A</u>	<u>IV-3B</u>	<u>IV-3C</u>	<u>IV-3D</u>	<u>IV-3E</u>	<u>IV-3F</u>	<u>IV-3G</u>	<u>IV-3H</u>
1b	220	220	221	141	140	140	140	220
2b	332	331	332	148	142	140	140	269
4b	422	422	419	157	151	145	147	334
7b	487	487	484	150	143	140	140	386
10b	534	534	531	148	143	140	141	419
0pi	140	146	158	159	141	140	140	139
1pi	141	154	174	174	149	140	140	139
2pi	---	---	---	---	---	---	---	---
3pi	151	220	302	301	210	171	140	139
4pi	160	277	398	398	264	207	140	139
5pi	171	339	483	484	324	---	---	139
6pi	186	402	559	560	292	302	145	140
7pi	199	452	615	614	446	345	145	140
8pi	---	---	---	---	---	---	---	---
9pi	208	534	745	745	536	409	145	141
3p2	326	369	430	363	254	208	145	152
6p2	332	450	572	549	392	303	144	144
8p2	376	522	676	637	470	359	146	144
3p7	---	---	---	---	---	---	---	---
6p7	494	512	543	431	311	249	147	152
8p7	501	530	587	500	371	289	147	147
2p10	---	---	---	---	---	---	---	---
6p10	534	536	536	250	198	174	148	152
8p10	---	---	---	---	---	---	---	---
9p10	534	538	541	276	203	172	140	142
10C1	536	535	532	166	156	150	148	158
10C2	535	535	532	166	156	150	148	159
10C3	536	536	534	164	153	145	146	157
10C4	535	535	532	163	154	148	148	161
10C5	534	534	532	165	163	162	159	166
10C6	534	534	531	168	170	166	164	172

TABLE 10
SYSTEM NO. 16, COMPARISON OF COMPUTED AND EXPERIMENTAL RESULTS

TEST NO.	Temperature, °R			Computed Heat Flow, Btu/hr (watts)			Experimental Heat Flow, Btu/hr (watts)					
	Cap	Upper Baffle	Lower Baffle	Heater Power	Penetr. Effect	Heater Power	Calor.		Calor. Insulation	Penetr.		
							Boil-off			Effect Total	Difference	
IV-3A	208	540	540	0	1.7 (0.5)	0	11.9 (3.5)	10.2 (3.0)	1.7 (0.5)	0		
3B	534	540	540	13.1 (3.8)	13.5 (4.0)	13.6 (4.0)	24.1 (7.1)	10.2 (3.0)	14.1 (4.1)	-4		
3C	746	540	540	37.9(11.1)	34.8(10.2)	44.3(13.0)	57.3(16.8)	10.2 (3.0)	47.1(13.8)	-26		
3D	745	140	140	39.3(11.5)	34.5(10.1)	44.6(13.1)	43.3(12.7)	0	43.3(12.7)	-20		
3E	539	140	140	14.5 (4.2)	13.3 (3.9)	15.4 (4.5)	14.3 (4.2)	0	14.3 (4.2)	-7		
3F	409	140	140	7.0 (2.0)	6.6 (1.9)	6.8 (2.0)	6.5 (1.9)	0	6.5 (1.9)	1		
3G	145	140	140	---	---	0	-0.27(-0.08)	0	0	---		
3H	142	540	140	---	---	0	6.1 (1.8)	6.1 (1.8)	0	---		

TABLE 11a
MATHEMATICAL MODEL RESULTS
INSULATION SYSTEM NO. 16
(Power in Btu/hr)

Computer Run No.	50	51	108	109	52	53
Test Simulation	IV-3A	IV-3B	IV-3C	IV-3D	IV-3E	IV-3F
Warm End Temp. (°R)	208	535	750	750	535	408
Cold End Temp. (°R)	140	140	141	140	140	140
Environment Temp. (°R)	540	540	540	140	140	140
Pen. Wall Thickness (in.)	.027	0.027	0.027	0.027	0.027	0.027
Warm End Emittance	.34	0.34	0.34	0.34	0.34	0.34
Cold End Emittance	.27	0.27	0.27	0.27	0.27	0.27
Wall Emittance	.41	0.41	0.41	0.41	0.41	0.41
Heat Flow (Btu/hr) (1)	-1.02	0.03	2.80	3.85	1.02	0.34
(2)	0.05	4.10	16.5	16.6	4.19	1.36
(3)	0.96	8.95	18.6	18.8	9.25	5.25
(4)	0.02	2.87	11.8	11.6	3.00	0.96
(5)	0.60	0.41	-0.14	-0.75	-0.19	0.06
(6)	1.60	12.30	30.1	29.85	12.1	6.18
(7)	0.04	1.23	4.84	4.81	1.19	0.41
(8)	1.64	13.50	34.8	34.5	13.3	6.61
Heater Power (Btu/hr)	0	13.08	37.9	39.25	14.46	6.95
Side Wall Balance (Btu/hr)	-0.02	-0.07	+0.16	-0.15	-0.04	+0.09
Cold End Balance (Btu/hr)	0	+0.03	+0.14	+0.16	-0.01	-0.02
Pen. Heat Balance (Btu/hr)	-0.02	-0.04	+0.16	+0.15	-0.05	+0.06

Heat Balance Equations

$$\begin{aligned}
 \text{Heater Power} &= (1)+(2)+(3) \\
 \text{Side Wall Balance} &= (3)+(4)+(5)-(6) \\
 \text{Cold End Balance} &= (6)+(7)-(8) \\
 \text{Pen. Heat Balance} &= \text{Heater Power} \\
 &\quad -(1)+(5)-(8)
 \end{aligned}$$

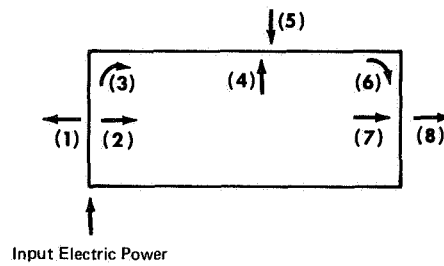


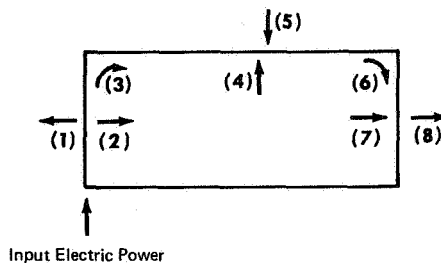
TABLE 11b

MATHEMATICAL MODEL RESULTS
INSULATION SYSTEM NO. 16

	(Power in Watts)					
Computer Run No.	50	51	108	109	52	53
Test Simulation	IV-3A	IV-3B	IV-3C	IV-3D	IV-3E	IV-3F
Warm End Temp. (°R)	208	535	750	750	535	408
Cold End Temp. (°R)	140	140	141	140	140	140
Environment Temp. (°R)	540	540	540	140	140	140
Pen. Wall Thickness (in.)	.027	0.027	0.027	0.027	0.027	0.027
Warm End Emittance	-.34	0.34	0.34	0.34	0.34	0.34
Cold End Emittance	-.27	0.27	0.27	0.27	0.27	0.27
Wall Emittance	.41	0.41	0.41	0.41	0.41	0.41
Heat Flow (watts) (1)	-0.30	0.01	0.82	1.13	0.30	0.10
(2)	0.02	1.20	4.83	4.86	1.23	0.40
(3)	0.28	2.62	5.46	5.51	2.71	1.54
(4)	0.01	0.84	3.45	3.40	0.88	0.28
(5)	0.17	0.12	-0.04	-0.22	-0.06	0.02
(6)	0.47	3.60	8.82	8.74	3.54	1.81
(7)	0.01	0.36	1.42	1.41	0.35	0.12
(8)	0.48	3.95	10.20	10.10	3.90	1.94
Heater Power (watts)	0	3.83	11.11	11.50	4.24	2.04
Side Wall Balance (watts)	-0.01	-0.02	+0.05	-0.05	-0.01	+0.03
Cold End Balance (watts)	0	+0.01	+0.04	+0.05	-0.01	-0.01
Pen. Heat Balance (watts)	-0.01	-0.01	+0.05	+0.05	-0.02	+0.02

Heat Balance Equations

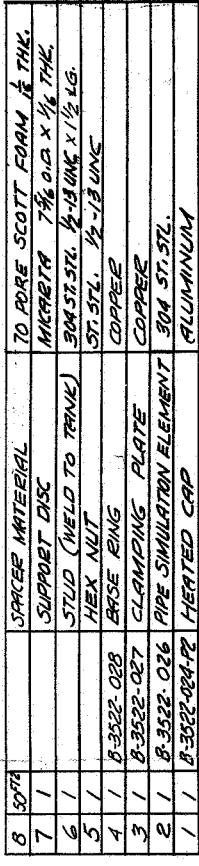
$$\begin{aligned}
 \text{Heater Power} &= (1)+(2)+(3) \\
 \text{Side Wall Balance} &= (3)+(4)+(5)-(6) \\
 \text{Cold End Balance} &= (6)+(7)-(8) \\
 \text{Pen. Heat Balance} &= \text{Heater Power} \\
 &\quad -(1)+(5)-(8)
 \end{aligned}$$





**FIGURE 25 CUTOUT IN INSULATION SYSTEM No. 15 FOR
PENETRATION ATTACHMENT**

FIGURE 26 MULTILAYER INSULATION 6" "A" CAP PIPE PENETRATION DETAIL

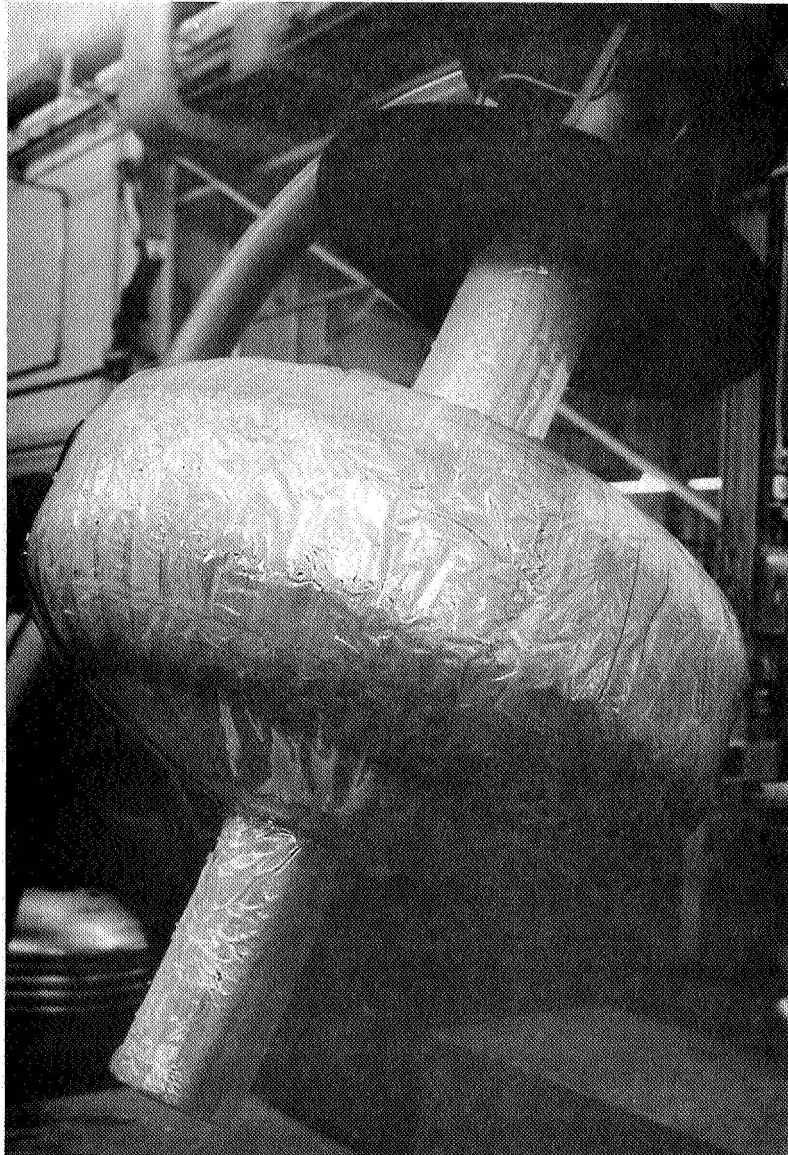




**FIGURE 27 UNINSULATED "A" CAP PIPE PENETRATION
ATTACHED TO TANK CALORIMETER**



FIGURE 28 INSULATED "A" CAP PIPE PENETRATION



**FIGURE 29 INSULATED TANK CALORIMETER AND "A" CAP
PIPE PENETRATION**

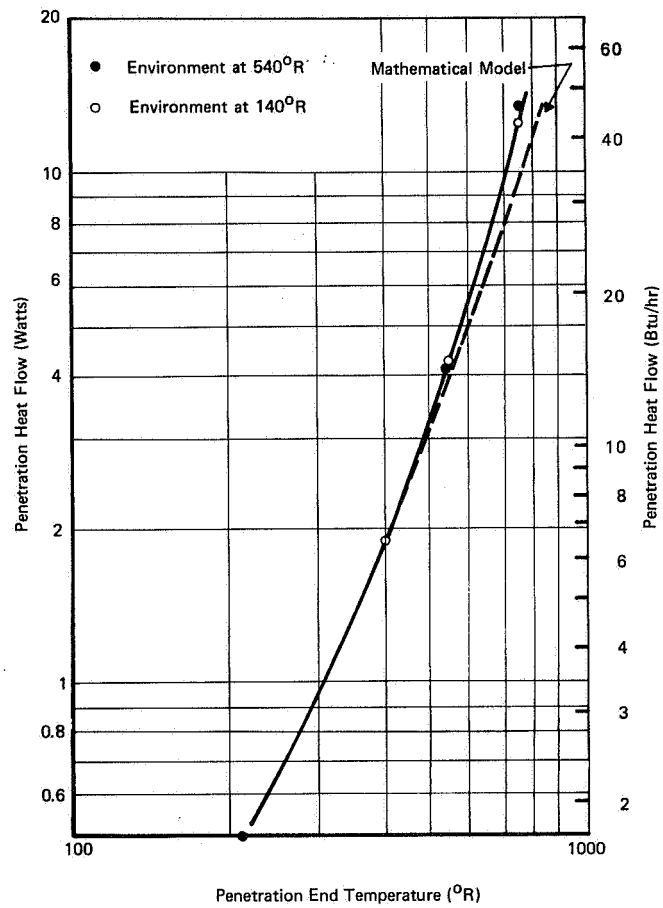


FIGURE 30 SYSTEM NO. 16, PENETRATION HEAT FLOW VS END CAP TEMPERATURE, TEST SERIES IV-3

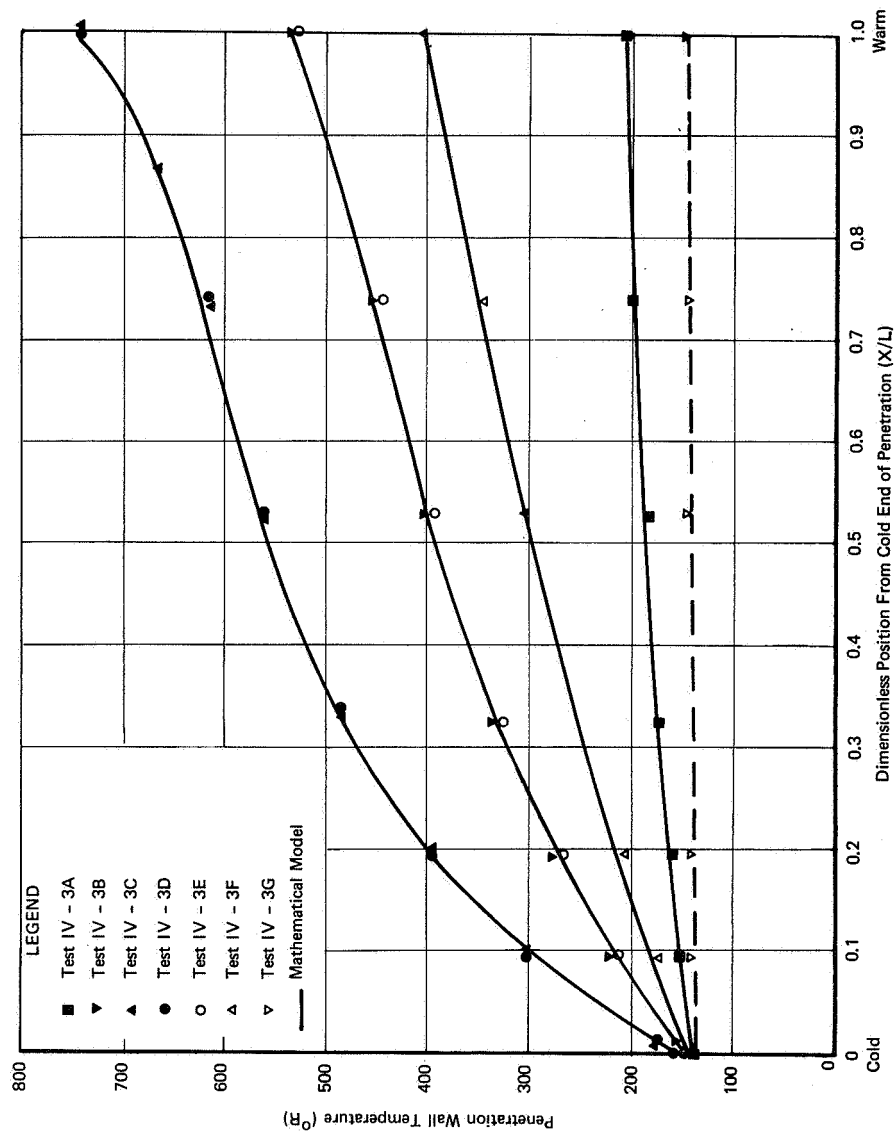


FIGURE 31 SYSTEM NO. 16, PENETRATION WALL TEMPERATURES FOR TEST SERIES IV-3

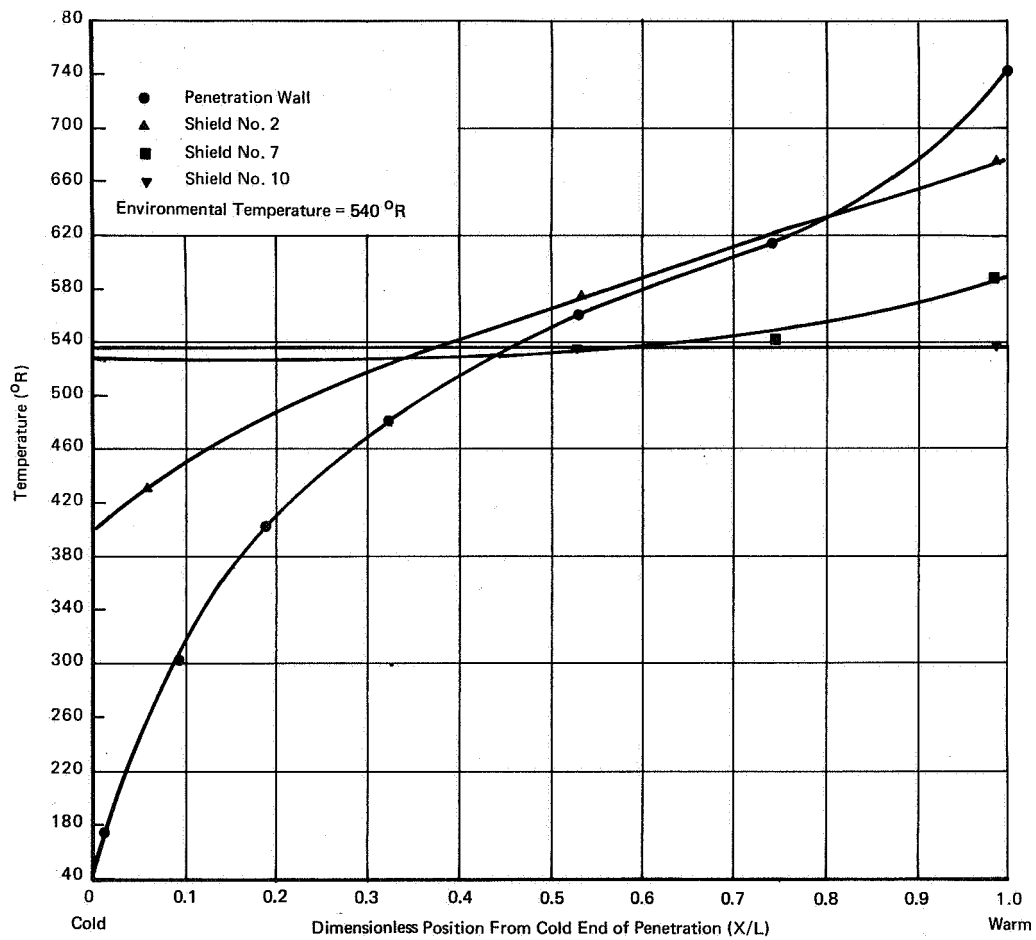


FIGURE 32 SYSTEM NO. 16, PENETRATION SYSTEM TEMPERATURES, TEST IV-3C

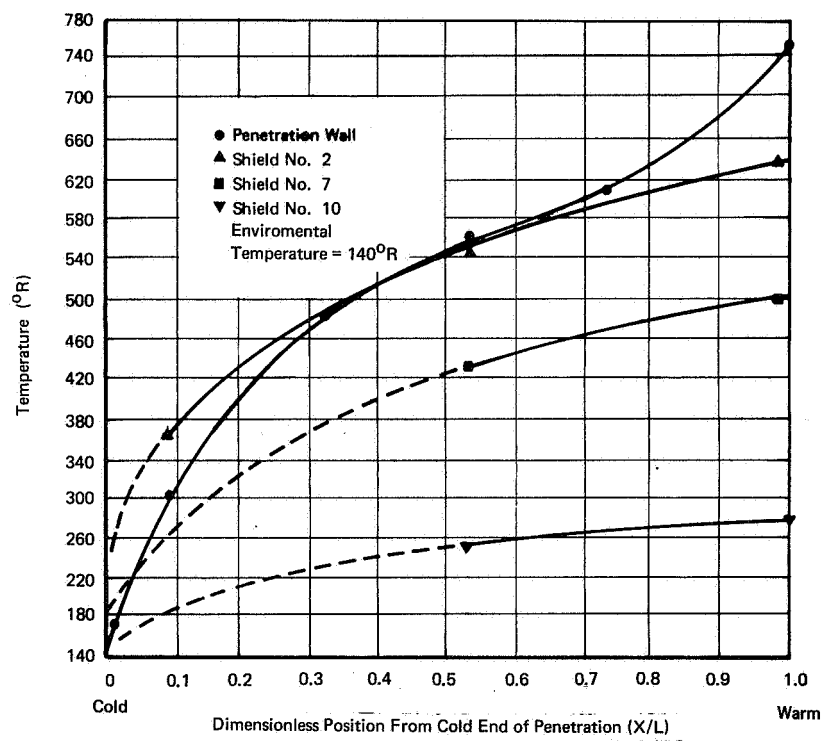


FIGURE 33 SYSTEM NO. 16, PENETRATION SYSTEM TEMPERATURES, TEST IV-3D

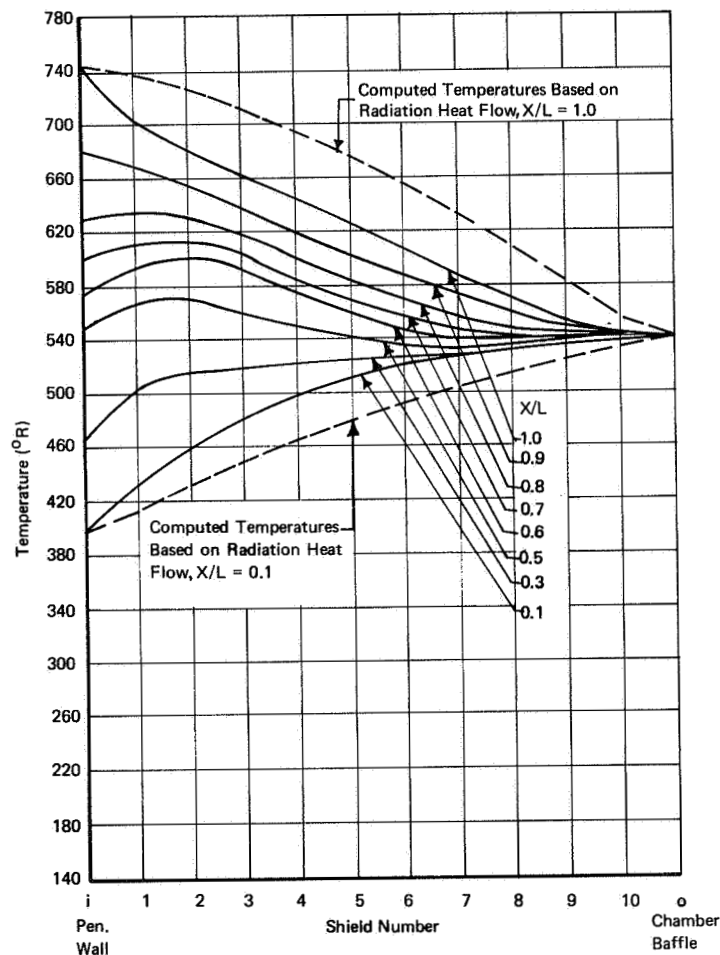


FIGURE 34 SYSTEM NO. 16, PENETRATION MLI, TEMPERATURE DISTRIBUTION, TEST IV-3C

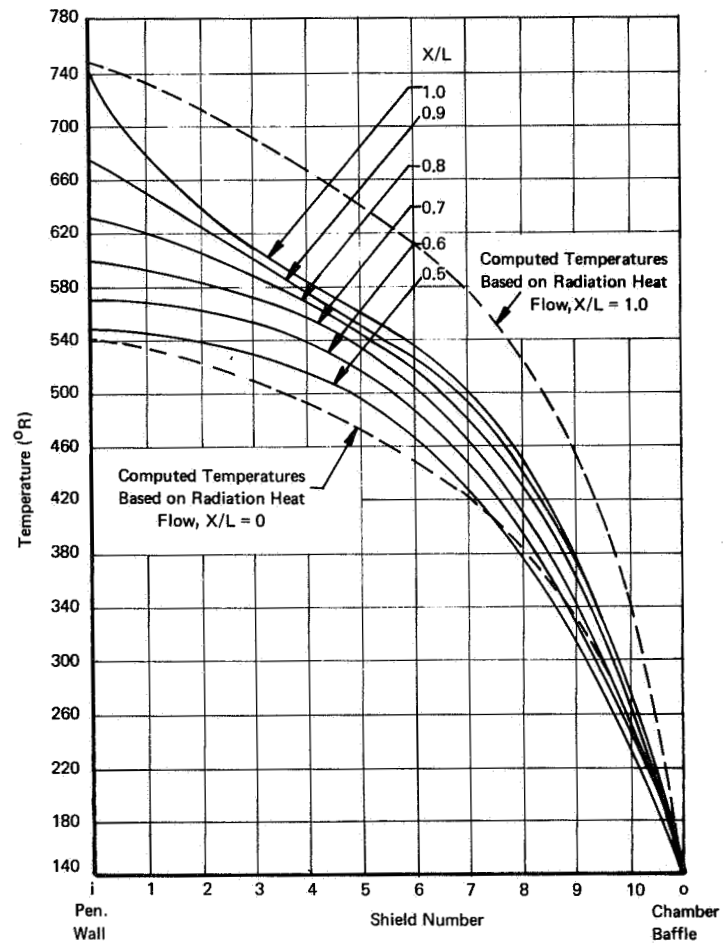


FIGURE 35 SYSTEM NO. 16, PENETRATION MLI, TEMPERATURE DISTRIBUTION, TEST IV-3D

and internal radiation. Also, the heat flows obtained represent significant values of liquid hydrogen boil-off on extended missions. At end-cap temperatures less than 540°R, the predicted values are about 5% less than those measured. At end-cap temperatures of 750°R, heat flow values of about 44 Btu/hr (13 watt) were measured, and the predicted values were smaller by 25%. Excellent agreement was demonstrated between the measured temperature distributions and the predicted temperatures. Finally, the foam spacer used in the penetration MLI appeared to outgas seriously at the penetration warm end during tests at 750°R.

C. INSULATION SYSTEM NO. 17

The penetration of System No. 16 was modified, and a second series of tests was performed with the insulated calorimeter and penetration now identified as System No. 17. The modification involved the removal of the "A" cap and the attachment of the "B" cap on the penetration. With this change it was possible to maintain the warm end of the penetration and the environment surrounding the penetration at the same temperature level while the tank calorimeter viewed a different environmental temperature level. Seven tests were performed with the system for various combinations of chamber baffle and penetration temperatures to obtain the temperature distribution in the penetration wall and the heat flowing through the penetration to the calorimeter.

1. Description of Penetration

a. "B" Cap Details

The portion of the "B" cap attached to the penetration was constructed similarly to the "A" cap and made of the same material (6061-T6 aluminum). The cap was threaded to fit the free end of the penetration. The portion of the cap facing the interior of the penetration was sandblasted to produce a diffuse reflecting surface with an emittance of about 0.34. A wafer-type electrical heater 6 inches in diameter was bonded to the outside of the cap at its center. The "B" cap differed from the "A" cap in that its exposed portion extended out to a diameter of 10 inches, where it connected to a 9-inch OD aluminum cylinder with $\frac{1}{4}$ -inch wall that extended back toward the tank from the free end of the penetration so as to completely surround the penetration. Details of this configuration are shown in Figure 36.

b. Insulation Details

The penetration insulation of System No. 16 was modified to accommodate the "B" cap as follows: the flat layers of insulation at the free end were removed one at a time, and the "A" cap was unscrewed from the penetration. Next, the cylindrical portion of the insulation was cut flush with the end of the penetration. A Dacron cover was then sewn over the edge of the MLI to prevent shorting of the shield to the "B" cap when the latter was in place and to maintain dimensional control of the edge section.

An MLI cover was also placed over the "B" cap to limit heat loss from the cap to the chamber baffles. This system was constructed of the same materials as the tank calorimeter insulation, with the exception that 25 shields were used in the cap system.

The construction details at the corner and at the mouth of the cap are shown in Figure 36. The cap and its insulation are shown separately in Figure 37. Finally, in Figure 38 the cap and its insulation are shown assembled to the penetration at the calorimeter tank.

c. Surface Emittances

The emittances of all interior surfaces remained unchanged from System No. 16. The inner surfaces of the "B" cap, except the threaded area, and the surface of the cap facing the penetration were coated with 3M velvet black. The outer surfaces of the cap were left in their original machined condition.

d. Temperature Sensors

While the penetration insulation was being modified, a number of thermocouple leads were broken. The sensors could not be easily replaced without dismantling the insulation.

Four new thermocouples were added to the temperature measuring system. All were placed on the exterior surface of the "B" cap at the locations indicated in Figure 36 (9 Pi, 9 P12, 9 P25, BC-3).

2. Test Results

a. Tests and Conditions

Seven tests were performed with System No. 17 in the IV-4 test series. As in other tests discussed previously, the relevant test data were obtained after the calorimeter system had achieved quasi steady-state conditions. The duration of the steady-state test period ranged from 47 to 112 hours. The space simulation chamber vacuum at the time these measurements were taken was 5×10^{-7} torr or less. The baffles were maintained at their required temperature levels of 140°R and/or 540°R, generally within 2°F both before and during the measurement period.

The prevailing conditions of test and the heat flow determined from the calorimeter boil-off are presented in Table 12. The average temperatures measured at the penetration wall and within the penetration and tank MLI are presented in Table 13.

b. Heat Flow

(1) Comparison of Systems 15 and 17. Two tests were performed with System No. 17 that are comparable to those performed with System No. 15. The additional heat flow due to the penetration was avoided by operating all chamber baffles at 140°R, or by operating the upper baffle at 540°R and the lower baffle at 140°R; these conditions were representative of those maintained in the performance of Tests IV-4B and IV-4A, respectively.

The 140°R upper and lower baffle environment produced a calorimeter heat flow of 0.17 Btu/hr (0.05 watt). Under the same conditions a heat flow of 0.20 Btu/hr (0.06 watt) was obtained with System No. 15. No extraneous heat flows appeared to be introduced into the calorimeter system at the time of the System No. 17 test compared with the System No. 15 test.

With a warm upper baffle, System No. 17 produced a heat flow of 7.3 Btu/hr (2.14 watts), compared with 6.1 Btu/hr (1.8 watts) for System No. 15. This represents a 20% increase in the heat flow to the upper insulation half. A portion of this increase can be identified as a heat input from the penetration, which was undergoing a temperature transient during the test period. For example, at the beginning of test the end-cap temperature was 163°R; at the end of test the cap temperature was 154°R. From these temperatures, wall conduction heat flows of 0.29 and 0.18 Btu/hr (0.085 and 0.053 watt) were calculated for the respective temperatures under the assumption of no radiation heat flow in the penetration. If a mean value of 0.24 Btu/hr (0.07 watt) is assumed for the heat flow during the test period, this accounts for one-fifth of the difference between the two systems.

As noted earlier, temperatures within the tank MLI were measured only at location "b", because the readout capabilities of the instrumentation were limited. A comparison of the temperatures at this location for the two systems indicated about 20°R lower temperatures for System No. 17. This could have resulted from a minor shift of the tank MLI toward the bottom. A uniform downward shift of the MLI would result in an increased compressive load on the shield-spacer system, which is usually attended by increased heat flow. During the modification of the penetration, the tank calorimeter was placed in a jig and inverted several times; this rotation and inversion could have caused the insulation to shift.

In summary, the heat flow value for the upper-half insulation in a 540°R environment increased by approximately 1 Btu/hr (0.3 watt). This increase was added to the value of 10 Btu/hr (3.0 watts) obtained in Test III-2A for the entire tank MLI, giving a total of 11.2 Btu/hr (3.3 watts) assigned to the tank MLI for the purposes of determining the penetration heat flow in all 540°R environments with System No. 17.

(2) System 17, Heated Penetration Results. Five tests were performed with a heated penetration. Power to the electric heater at the free end of the penetration was adjusted in each test to maintain the cap temperature at a predetermined value. Nominal cap temperatures were 450, 540, and 750°R. One test was performed at each cap temperature with 140°R environmental temperature. In addition, one test was performed at each of the two latter cap temperatures with a 540°R environment.

Test IV-4C, performed with 411°R end-cap temperatures and 140°R environment, resulted in a penetration heat flow of 6.1 Btu/hr (1.8 watts); this result was in good agreement with the predicted value of 6.5 Btu/hr (1.9 watts). The heat transfer from warm to cold end is dominated by wall conduction, as noted in Table 14, from the heat flow values at locations 3 and 6.

Tests IV-4D and -4E (Table 15) were performed with nominal 540°R cap temperatures and 140 and 540°R environment temperatures, respectively. In the former, both the measured and predicted heat flows were 14.3 Btu/hr (4.2 watts). In the latter test, the experimental heat flow was 17 Btu/hr (5.0 watts), compared with a predicted value of 14.3 Btu/hr (4.2 watts). The 2.7 Btu/hr (0.8-watt) increase appeared to be due to added tank MLI heat leak rather than increased penetration heat flow. The electric power to the cap in this case was 14.3 Btu/hr (4.2 watts). Because the "B" cap and the chamber environment were both at 540°R, the heat loss or gain by the cap was negligible. Thus, all the electric power into the cap should have been evidenced as penetration heat flow. An increase in the penetration heat flow above 14.3 Btu/hr (4.2 watts) would require a corresponding increase in the "B" cap power input. An added consideration is that Tests IV-3B and -4E were identical in all respects except for the different caps. In Test IV-3B, the heater input power was 13.6 Btu/hr (4.0 watts), compared with a measured penetration heat leak of 14 Btu/hr (4.1 watts).

The tests performed with "B" cap temperatures at nominal 750°R also showed a decided difference in heat leak when the 140 and 540°R baffle test results were compared. Test IV-4F produced a penetration heat flow of 46.0 Btu/hr (13.5 watts) with a 540°R baffle; Test IV-4G produced a heat flow of 42.0 Btu/hr (12.3 watts) with a 140°R baffle. In both cases the 6-inch penetration viewed the same 750°R environment. However, as the cap in Test IV-4F was 8°R higher than that in Test IV-4G, the former would be expected to produce a 1.4 to 1.7 Btu/hr (0.4 to 0.5 watt) higher penetration heat flow on the basis of the power rates developed in the preceding section. After compensation for cap temperature, there was still a difference of 2.0 to 2.4 Btu/hr (0.6 to 0.7 watt) attributed to chamber environmental temperature. A similar difference (approximately 3.8 Btu/hr (1.1 watts)) was observed in comparisons of Tests IV-3C and -3D, discussed in the preceding section. As the higher heat flow was associated with 540°R baffle temperature, it is valid to assume that the added heat flow was associated with the tank MLI performance and not with the penetration performance.

A further investigation was made to substantiate the foregoing assumption. The tank MLI in the vicinity of the "B" cap underwent an abrupt change in environmental conditions in Test IV-4G. From the data presented in Table 13 for the 10C1 through 10C6 temperature points, the heat flow from the outer shield of the MLI to the 140°R environment was computed; the integrated heat flow over an area of 4.3 ft² surrounding the penetration was 0.24 Btu/hr (0.07 watt). If the temperature measured at location 10C6 was assumed to exist over the remaining area of the lower-half surface of the tank calorimeter, the heat flow away from the MLI had a computed value of 0.85 Btu/hr (0.25 watt). However, only a fraction of this heat loss would be assigned to reduced penetration heat flow.

From the data presented in Table 13, the heat flow radiated from the warm cap was 19, 31, and 45% of the total penetration heat flow for the nominal 400, 540 and 750°R end-cap temperatures, respectively. Thus, as the amount of radiant energy in the cavity increases, the optical properties of the interior surfaces have greater significance in predictions of the total penetration heat flows. Further discussion is presented in Section IV-C-2.

c. Penetration Temperatures

The penetration wall temperatures obtained with the mathematical and experimental models are shown in Figure 39. The shapes of the temperature distributions are in excellent agreement. Note that the experimental end-cap temperatures at the nominal 540 and 750°R temperature levels do not match the conditions of the computer runs. Of the several thermocouples located on the "B" cap, one that was improperly located was used in setting the cap temperature level and for the computer input data. It was later determined that the actual experimental end-cap temperature was lower than the value indicated. Had the computer results been based on the actual conditions, the absolute values as well as the shape of the distribution would be in good agreement.

A typical temperature distribution in the penetration MLI is shown in Figure 40 for Test IV-4G.

d. Summary

The heat flows measured with the "B" cap were comparable in value with those obtained with the "A" cap at corresponding end-cap temperatures. The heat flows were large and probably could not be tolerated for long-term storage of cryogenics in space. The penetration MLI appears to contribute a negligible portion of the total penetration heat flow. Changes in the penetration MLI between the "A" and "B" cap configurations did not appear to have significant effects. Predicted and experimental penetration wall temperatures showed excellent agreement. The difference between predicted and experimental heat flows ranged from 5% at a low cap temperature to about 20% at a high cap temperature.

TABLE 12
DATA SUMMARY, INSULATION SYSTEM NO. 17, TEST SERIES IV-4, CALORIMETER NO. 1

Insulation System: Ten radiation shields of polyester film aluminized on both sides and eleven spacers, each consisting of two layers of silk netting

Test Facility: Arthur D. Little, Inc.

Penetrations or Gaps: 6-inch diameter pipe penetration ("B" cap)

Tank Surface Area: 40.6 sq ft based on surface area of the calorimeter tank and penetration

Tank Liquid: Liquid nitrogen

Temperature of Neck Guard: -318°F

Test No.	Data Period		Test Duration, hrs.	Avg. Vac., torr	Temp. of Penetr. Warm, End, °R	Upper Baffle, °R	Lower Baffle, °R	Total Measured		Heater Power to Penetr. End Btu/hr (watts)	
	Date & Time	Date & Time						Calorimeter, Btu/hr (watts)			
IV-4A	9-30-67/0830	10-3-67/0830	72	2 x 10 ⁻⁷	154	540	140	7.3	(2.14)	0	--
IV-4B	10-5-67/1630	10-9-67/1630	96	1 x 10 ⁻⁷	147	140	140	0.17	(0.05)	0	--
IV-4C	10-13-67/0830	10-16-67/0830	72	1 x 10 ⁻⁷	405	140	140	6.1	(1.80)	9.9	(2.9)
IV-4D	10-21-67/0930	10-23-67/0830	47	1 x 10 ⁻⁷	528	140	140	14.3	(4.20)	23.8	(7.0)
IV-4E	10-26-67/1630	10-31-67/0830	112	3 x 10 ⁻⁷	533	540	540	28.3	(8.30)	13.3	(4.2)
IV-4F	11-2-67/0830	11-5-67/1530	79	5 x 10 ⁻⁷	731	540	540	56.2	(16.50)	64.1	(18.8)
IV-4G	11-7-67/0830	11-10-67/0830	72	1 x 10 ⁻⁷	720	140	140	42.0	(12.30)	68.0	(20.0)

TABLE 13

TEMPERATURE DISTRIBUTION, SYSTEM NO. 17, TEST SERIES IV-4

T.C. NO.	Temperature (°R)						
	IV-4A	IV-4B	IV-4C	IV-4D	IV-4E	IV-4F	IV-4G
1b	228	139	140	139	241	231	140
2b	271	144	144	139	400	322	150
4b	327	148	152	145	416	417	160
7b	381	144	146	144	422	489	160
10b	418	147	145	139	535	536	151
0pi	---	---	---	---	146	---	---
1pi	139	139	144	142	154	175	173
2pi	139	139	150	157	168	204	201
3pi	140	139	179	205	218	303	296
4pi	142	139	211	264	276	400	392
5pi	145	141	251	324	335	487	477
6pi	148	143	302	397	399	560	551
7pi	152	144	344	447	450	615	606
8pi	154	146	398	519	523	716	705
9pi-2	154	147	405	528	533	731	720
3p2	151	143	239	316	369	493	460
6p2	150	144	330	438	448	620	610
8p2	153	146	395	518	523	720	709
3p7	167	150	318	440	496	649	621
6p7	155	146	370	496	510	698	686
8p7	154	146	396	521	527	722	713
3p10	---	---	---	---	---	---	---
8p10	---	---	---	497	528	533	521
10C1	159	151	316	422	533	655	595
10C2	160	152	276	350	535	593	470
10C3	157	150	214	260	537	553	339
10C4	160	150	178	198	536	540	---
10C5	163	158	---	---	527	535	---
10C6	170	164	171	180	536	535	189
BC3	151	144	398	522	532	724	713

TABLE 14a

MATHEMATICAL MODEL RESULTS

INSULATION SYSTEM NO. 17

(Power in Btu/hr)

Computer Run No.	101	103	104	107	102
Test Simulation	IV-4C	IV-4D	IV-4E	IV-4F	IV-4G
Warm End Temp. (°R)	411	540	541	757	749
Cold End Temp. (°R)	(140)	(140)	146	(140)	(140)
Environment Temp. (°R)	140	140	540	540	140
Pen. Wall Thickness (in.)	0.027	0.027	0.027	0.027	0.027
Warm End Emittance	0.34	0.34	0.34	0.34	0.34
Cold End Emittance	0.27	0.27	0.27	0.27	0.27
Wall Emittance	0.41	0.41	0.41	0.41	0.41
Heat Flow (Btu/hr) (1)	2.18	7.35	0	19.95	27.35
(2)	1.23	4.30	4.30	16.20	16.20
(3)	5.01	9.20	9.20	18.15	18.15
(4)	0.85	3.03	3.03	11.10	11.10
(5)	0.10	0.41	0.41	1.46	1.46
(6)	5.96	12.64	12.64	30.77	30.77
(7)	0.38	1.26	1.26	4.98	4.98
(8)	6.48	13.97	13.97	35.8	35.8
(9)	1.23	3.07	0.61	8.96	11.70
(10)	0.75	3.07	0.51	8.10	11.45
(11)	0.82	2.73	0	7.48	10.15
Heater Power (Btu/hr)	9.65	23.92	14.11	63.26	73.40
Side Wall Balance	0	0	0	-0.06	-0.06
Cold End Balance	-0.14	-0.07	-0.07	-0.05	-0.05
"B" Cap Balance	-0.31	-0.07	+0.20	+0.02	+0.09
Penetration Balance	+0.17	-0.13	+0.14	+0.03	+0.10

Heat Balance Equations

$$\begin{aligned}
 \text{Heater Power} &= (1) + (2) + (3) + (9) \\
 \text{Side Wall Balance} &= (3) + (4) + (5) - (6) \\
 \text{Cold End Balance} &= (6) + (7) - (8) \\
 \text{"B" Cap Balance} &= (9) - (11) - (5) \\
 \text{Penetration Balance} &= \text{Heater Power} \\
 &\quad - (1) - (11) - (8)
 \end{aligned}$$

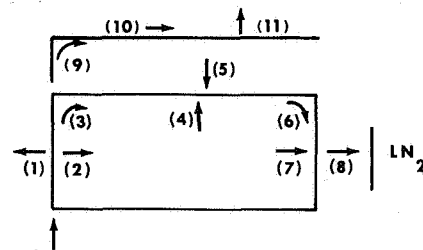


TABLE 14b
MATHEMATICAL MODEL RESULTS,
INSULATION SYSTEM NO. 17

	(Power in watts)				
Computer Run No.	101	103	104	107	102
Test Simulation	IV-4C	IV-4D	IV-4E	IV-4F	IV-4G
Warm End Temp. (°R)	411	540	541	757	749
Cold End Temp. (°R)	(140)	(140)	146	(140)	(140)
Environment Temp. (°R)	140	140	540	540	140
Pen. Wall Thickness (in.)	0.027	0.027	0.027	0.027	0.027
Warm End Emittance	0.34	0.34	0.34	0.34	0.34
Cold End Emittance	0.27	0.27	0.27	0.27	0.27
Wall Emittance	0.41	0.41	0.41	0.41	0.41
Heat Flow (watts) (1)	0.64	2.16	0	5.85	8.01
(2)	0.36	1.26	1.26	4.75	4.75
(3)	1.47	2.70	2.70	5.32	5.32
(4)	0.25	0.89	0.89	3.26	3.26
(5)	0.03	0.12	0.12	0.43	0.43
(6)	1.75	3.71	3.71	9.03	9.03
(7)	0.11	0.37	0.37	1.46	1.46
(8)	1.9	4.1	4.1	10.5	10.5
(9)	0.36	0.90	0.18	2.63	3.44
(10)	0.22	0.90	0.15	2.76	3.36
(11)	0.24	0.80	0	2.19	2.97
Heater Power (watts)	2.83	7.02	4.14	18.55	21.52
Side Wall Balance	0	0	0	-0.02	-0.02
Cold End Balance	-0.04	-0.02	-0.02	-0.02	-0.02
"B" Cap Balance	+0.09	-0.02	+0.06	+0.01	+0.04
Penetration Balance	+0.05	-0.04	+0.04	+0.01	+0.04

Heat Balance Equations

Heater Power = (1)+(2)+(3)+(9)
 Side Wall Balance = (3)+(4)+(5)-(6)
 Cold End Balance = (6)+(7)-(8)
 "B" Cap Balance = (9)-(11)-(5)
 Penetration Balance = Heater Power
 -(1)-(11)-(8)

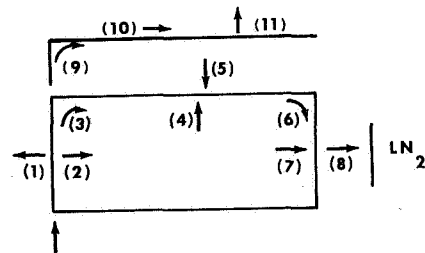


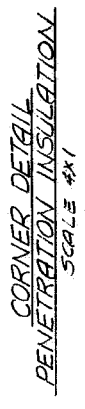
TABLE 15

SYSTEM NO. 17, COMPARISON OF COMPUTED AND EXPERIMENTAL RESULTS

TEST NO.	TEMPERATURE, °R		COMPUTED HEAT FLOW, Btu/hr(watts)		EXPERIMENTAL HEAT FLOW, Btu/hr(watts)			PERCENT DIFFERENCE	
	Upper Lower Cap Baffle Baffle		Heater Power	Penetr. Effect	Heater Power	Calor. Boil-off	Calor. Insulation	Penetr. Effect	
IV-4A	152 540	140	-	-	0	7.3 (2.1)	7.3 (2.1)	0	
IV-4B	147 140	140	-	-	0	0.17 (0.05)	0.17(0.05)	0	
IV-4C	411 140	140	9.6 (2.8)	6.5 (1.9)	9.9 (2.9)	6.1 (1.8)	0	6.1 (1.8)	+ 6
IV-4D	540 140	140	23.8 (7.0)	14.3 (4.2)	23.8 (7.0)	14.3 (4.2)	0	14.3 (4.2)	0
IV-4E	541 540	540	14.0 (4.1)	14.3 (4.2)	14.3 (4.2)	28.3 (8.3)	11.2 (3.3)	17.0 (5.0)	-16
IV-4F	757 540	540	63.0(18.5)	35.8(10.5)	64.1(18.8)	56.2 (16.5)	11.2 (3.3)	45.0(13.2)	-20
IV-4G	749 140	140	73.2(21.5)	35.8(10.5)	68.2(20.0)	42.0 (12.3)	0	42.0(12.3)	-15

FIGURE 36 MULTILAYER INSULATION 6" "B" CAP PIPE PENETRATION DETAIL

HOLDOUT FRAME



8	50" DIA		SPACER MATERIAL	TO PIPE SCOTT FOAM 1/8 THK
7				
6	1		STUD (WELD TO TANK)	304 ST. STL. 1/2-13 UNC x 1 1/2 LG.
5	1		HEX NUT	1/2-13 UNC 57. STL
4	1	B-3522-028	BASE RING	COPPER
3	1	B-3522-027	CLAMPING PLATE	COPPER
2	1	B-3522-026	PIPE SIMULATION ELEMENT	
1	1	B-3522-024-1	HEATED CAP	ALUMINUM

<u>LEGEND</u>	<u>DESCRIPTION</u>
<u>THERMOCOUPLE</u>	
1A THRU 8P1	WALL PENETRATION
OP AND 9P	BOTTOM & TOP OF PENETRATION
9P25	OUTSIDE INSULATION- CAP
3P2, 7, 10	PENETRATION INSULATION T.C. AT LOCATION 3P
6P2, 7, 10	PENETRATION INSULATION T.C. AT LOCATION 6P
8P2, 7, 10	PENETRATION INSULATION T.C. AT LOCATION 8P

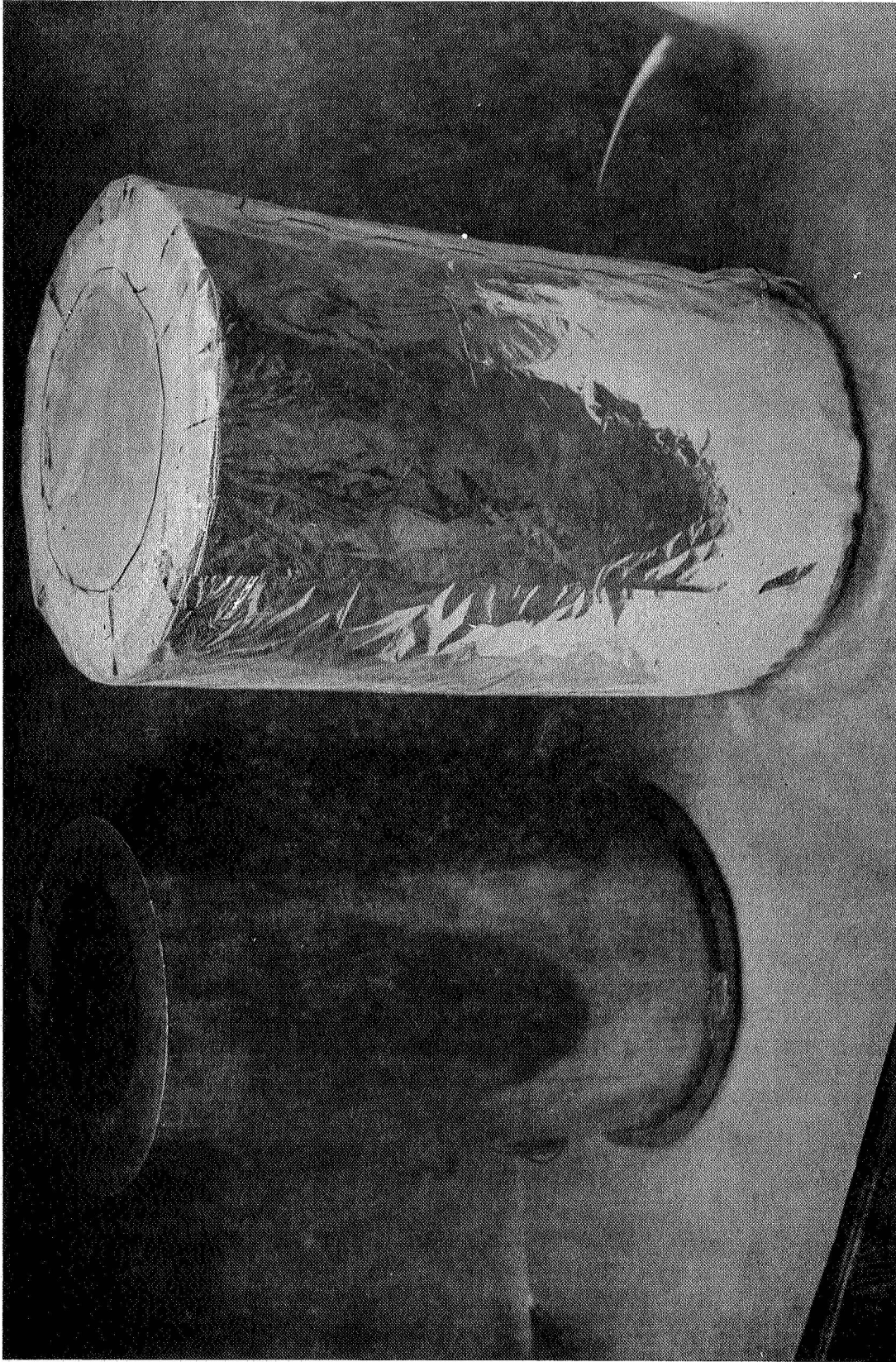


FIGURE 37 PENETRATION "B" CAP AND CAP INSULATION SYSTEM

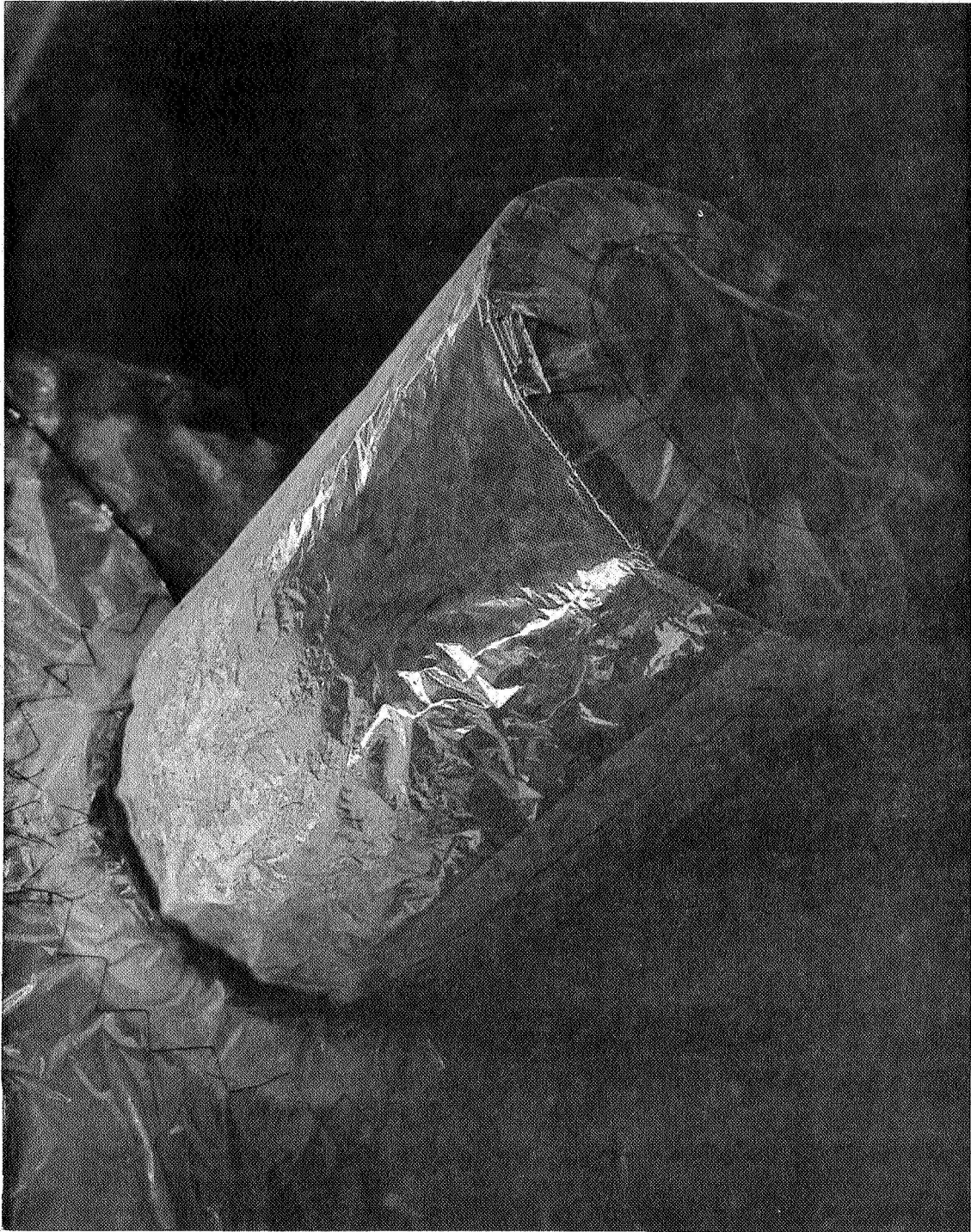


FIGURE 38 INSULATION SYSTEM NUMBER 17

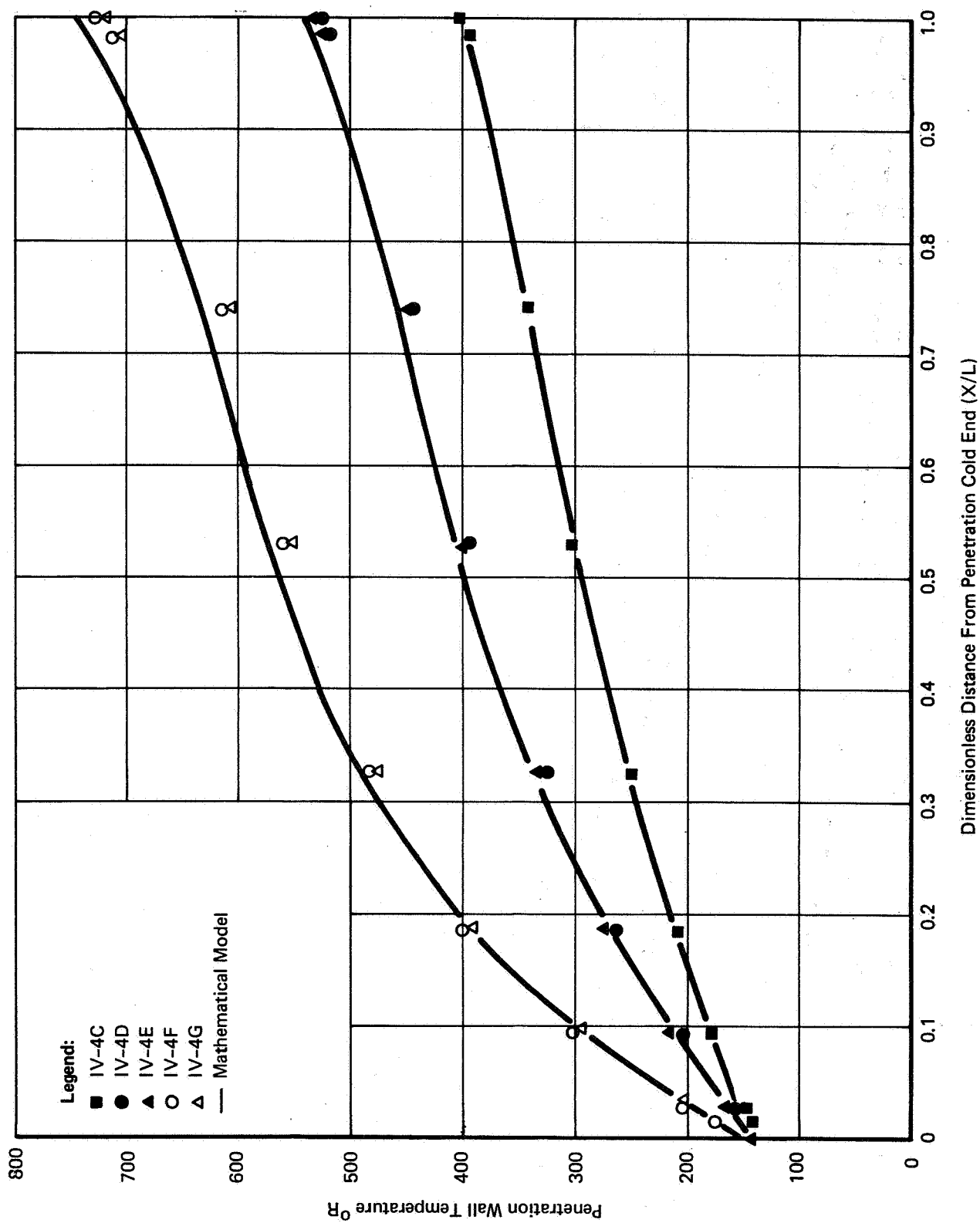


FIGURE 39 INSULATION SYSTEM NUMBER 17 - PENETRATION WALL TEMPERATURES. TEST SERIES IV-4

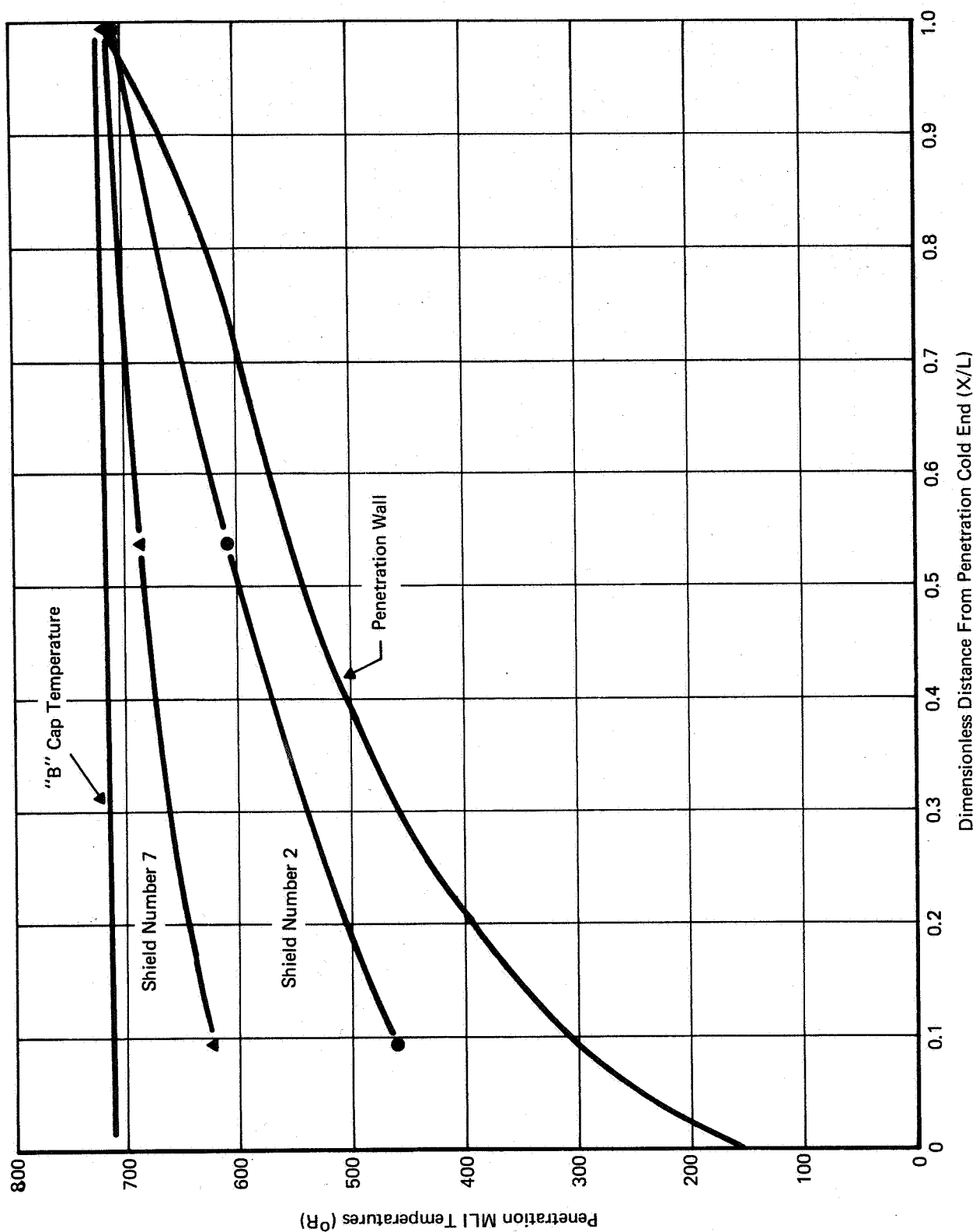


FIGURE 40 INSULATION SYSTEM NUMBER 17 - PENETRATION
MLI TEMPERATURES, TEST IV-4G

D. INSULATION SYSTEM NO. 18

1. Introduction

The experimental and analytical results obtained in the IV-3 and IV-4 test series show good agreement. Further, they show that the penetration heat leaks are large and possibly intolerable for a space mission. For example, the heat flow through the 6-inch penetration to the calorimeter is about 13.6 Btu/hr (4 watts) when the warm end of the penetration is 540°R; over a six-month period, this heat inflow is enough to vaporize 300 pounds of liquid hydrogen.

The final portion of this study was, therefore, directed to a significant reduction of the penetration heat flow resulting from internal radiation. Computer studies of the mathematical model, demonstrated that the heat transport in this mode could be reduced significantly by selectively lowering the emittance of the interior surface of the penetration. These analyses were based on the assumption that reflection from the low-emittance surfaces was completely diffuse. This assumption appeared to be valid for the penetrations tested in the IV-3 and IV-4 series, but it is probably less valid for low-emittance specular surfaces.

To achieve a significant reduction in radiation heat flow under the conditions investigated, we believe it will be necessary to use low-emittance surfaces on portions of the penetration interior and/or internal radiation shields placed perpendicular to the pipe axis.

In an actual space vehicle, the pipe connected to the propellant tank would likely extend beyond the insulated section. The uninsulated portion was simulated in our tests by a high-emittance cap. However, if the penetration heat flow is to be significantly reduced, this concept must be abandoned and provisions made within the pipe for incorporating a low-emittance baffle at the point where the insulation terminates. This can be accomplished practically by the use of an internal lightweight butterfly or similar valve to block the pipe opening. When fluid flow is desired, the valve opens like a check valve and offers little flow resistance.

We analyzed the effect of placing a low-emittance surface at the cold cap of the penetration and on the pipe interior. The warm end of the penetration was maintained at 750°R and the cold end at 140°R. The results of this work, which are presented in Section III-C-3, indicate a reduction of nearly 50% in the penetration heat flow. In terms of the radiation transport mechanism, the heat flow was reduced to a small fraction of its original value. Accordingly, the emittances of the interior surfaces of the penetration were altered, and three tests were performed with the modified system (No. 18) to verify the trend predicted by the mathematical model.

2. System No. 18 Description

The penetration configuration used in System No. 18 was identical to that of System No. 17, described in the preceding section. The emittance of the interior surfaces of the penetration was selectively reduced to lower the radiation transport between the warm and cold ends. The cold end, warm end, and portions of the wall were lined with aluminized Mylar as shown in Figure 41. The shield material was attached to the surface with double-faced masking tape.

The lengths of the cylindrical zones established in the mathematical model were used to define the lengths of the low-emittance bands. The first three zones at the cold end of the penetration had a total length of 1.6 inches, and the length of the first zone at the warm end had a length of 4.4 inches (see Figure 9). These dimensions were chosen as the low-emittance bandwidths at their respective locations. However, the positions of the two bands were reversed when the test article was modified, and the error was not discovered until the IV-5 test series was completed.

When the mathematical model was used to simulate the experimental model and conditions, it was necessary to use a computed average for the emittance in two zones. The most critical one was the first cylindrical zone at the warm end, which had a length of 4.4 inches. The length of the low-emittance band installed at this location was 1.6 inches. The remaining portions of the zone retained their original emittance value of 0.41. An average value of the emittance for this zone was computed from the relation $(A_1 e_1 + A_2 e_2) / A_1 + A_2$ where A_1 and A_2 were the areas associated with the different emittances e_1 and e_2 , respectively. The average emittance of the fourth cylindrical zone from the cold end was computed in the same manner.

3. Test Results

a. Test Conditions

Three tests were performed with System No. 18 in the IV-5 test series. As in other tests discussed previously, the relevant data were measured after the calorimeter system had achieved quasi steady-state conditions. The duration of the steady-state period varied from 28 to 52 hours. The space simulation chamber vacuum was less than 10^{-6} torr at the time these measurements were made. The baffles were maintained at their required temperature levels of 140°R and 540°R generally within 2°R both before and during the measurement period.

b. Heat Flow Results

The prevailing conditions of test and the heat flow determined from the calorimeter boil-off are presented in Table 16. A tabulation of the average

temperatures measured at the penetration wall and within the penetration and tank MLI is presented in Table 17. Some of the data is missing because the sensor leads were broken while the penetration was being modified. These sensors could not be easily replaced, because the MLI would have had to be partially dismantled to gain access to them.

The predicted penetration heat flows and other relevant data obtained with the mathematical model are presented in Table 18. Table 19 compares the predicted and experimental heat flow results.

The reduction in the penetration heat flow predicted from the mathematical model was achieved in the experimental model. A comparison of the radiation fluxes predicted for System No. 17 (see heat flows 2, 4, and 7, Table 18) indicates a significant radiation heat interchange for the 750°R end-cap temperature. Thus, the low-emittance surfaces on the interior of the System No. 18 penetration would reduce the penetration heat flow more significantly at a high cap temperature than at the lower cap temperatures. The data below summarizes the conditions and results of this comparison for the values obtained with the mathematical model.

Penetration Warm-end Temperature (°R Nominal)	540	750	
Chamber Baffle Temperature (°R Nominal)	540	140	540
System 17 Heat Flow, Btu/hr (watt)	13.9 (4.1)	35.8 (10.5)	35.8 (10.5)
System 18 Heat Flow, Btu/hr (watt)	12.0 (3.5)	23.0 (6.8)	23.0 (6.8)
Percent Reduction in Heat Flow	14	35	35

A similar comparison of the experimental results achieved with Systems 17 and 18 provides general confirmation of the predicted trend, as shown below.

Penetration Warm-end Temperature (°R Nominal)	540	750	
Chamber Baffle Temperature (°R Nominal)	540	140	540
System 17 Heat Flow, Btu/hr (watt)	17.0 (5.0)	42.0 (12.3)	45.0 (13.2)
System 18 Heat Flow, Btu/hr (watt)	11.2 (3.3)	26.9 (7.8)	27.9 (8.2)
Percent Reduction in Heat Flow	35	37	40

As noted previously for Systems 16 and 17, the measured values of the penetration heat flows at a cap temperature of 750°R were larger than the predicted values. However, the differences between the measured and predicted values were smaller for System No. 18 than for the other systems. The better correlation may have resulted from the reduction in radiation energy inside the penetration achieved through the use of low-emittance interior surfaces.

The penetration wall temperature distributions obtained with the mathematical and experimental models are shown in Figure 42. At the 540°R end-cap temperature the agreement between the two is excellent. On the other hand, the predicted temperatures are as much as 50°R higher than the available experimental temperatures obtained in the two tests at the 750°R end-cap temperature level. Checks and reruns of the computer model did not alter this result.

A possible source of the discrepancy noted above is the use of a computed average emittance at the first cylindrical zone at the warm end. This emittance had a value of 0.34, which was only 17% less than that used for the wall emittance in tests performed with Systems 16 and 17. However, the first 1.6 inches of wall at the warm end had an emittance of 0.025 in the experimental model; this would have produced significant radiation decoupling at the warm end and been evidenced by a steep temperature gradient in the wall. Verification of this could be obtained by redefinition of zones in the mathematical model, but this approach was not feasible at the time.

4. Summary

This test series demonstrated that the penetration heat flow can be reduced by reducing the radiation energy within the penetration. This was brought about by selective lowering of the interior surface emittance. Significant reductions were produced, and the surface changes were easily made. Greater reductions are possible by further lowering of the emittance and by the addition of baffles. These techniques can be combined with an increased L/D ratio and lower wall conductance to produce an order-of-magnitude reduction in the penetration heat flow. Of great significance is the ability of the mathematical model to predict the heat flows and temperature distributions in the penetration walls. This indicates that significant portions of the penetration problem can be studied through use of the mathematical model.

TABLE 16

DATA SUMMARY, INSULATION SYSTEM NO. 18, TEST SERIES IV-5, CALORIMETER NO. 1

Insulation System: Ten radiation shields of polyester film aluminized on both sides and eleven spacers, each consisting of two layers of silk netting.

Test Facility: Arthur D. Little, Inc.

Penetrations or Gaps: 6-inch diameter pipe penetration ("B" cap) with low-emittance ends

Tank Surface Area: 40.57 sq. ft. based on surface area of the calorimeter tank and penetration

Tank Liquid: Liquid nitrogen

Temperature of Neck Guard: -318°F

Test No.	-----Data Period----- Date & Time to Date & Time	Test Duration, hrs	Avg. Vac., torr	Temp. of		Upper Baffle, °R	Lower Baffle, °R	Total Measured Calorimeter, Btu/hr (watts).	Heater Power to Penetr. End Btu/hr (watts)
				Penetr.	Warm End, °R				
IV-5A	12-7-67/0830 12-8-67/1230	28	9×10^{-7}	540		540	540	22.5 (6.6)	9.9 (2.9)
IV-5B	12-19-67/1250 12-20-67/1600	27.5	3×10^{-7}	745		540	540	38.2 (11.5)	37.8 (11.1)
IV-5C	12-21-67/1250 12-23-67/1630	52	1×10^{-7}	745		140	140	26.6 (7.8)	42.5 (12.5)

TABLE 17

TEMPERATURE DISTRIBUTION, INSULATION SYSTEM NO. 18, TEST SERIES IV-5

Temperature (°R)			
<u>T. C. No.</u>	<u>IV-5A</u>	<u>IV-5B</u>	<u>IV-5C</u>
1b	225	225	140
2b	326	325	150
4b	417	417	164
7b	488	488	161
10b	---	---	---
0pi	142	148	147
1pi	150	163	162
2pi	---	---	---
3pi	---	---	---
4pi	257	334	331
5pi	326	---	426
6pi	393	513	510
7pi	---	---	---
8pi	---	---	---
9pi-1	545	745	746
9pi-25	540	535	233
3p2	---	---	---
6p2	---	---	---
8p2	531	728	725
3p7	495	653	631
6p7	---	---	---
8p7	531	731	728
3p10	---	---	---
8p10	---	---	---
10C1	537	673	631
10C2	537	610	515
10C3	537	557	361
10C4	537	540	251
10C5	---	---	---
10C6	535	534	525
BC3	539	739	735

TABLE 18a
MATHEMATICAL MODEL RESULTS,
INSULATION SYSTEM NO. 18
(Power in Btu/hr)

Computer Run No.	163	164	165
Test Simulation	IV-5A	IV-5B	IV-5C
Warm End Temp. (°R)	540	745	745
Cold End Temp. (°R)	142	148	147
Environment Temp. (°R)	540	540	140
Pen. Wall Tkns. (in.)	0.027	0.027	0.027
Warm End Emittance	0.025	0.025	0.025
Cold End Emittance	0.025	0.025	0.025
Wall Emittance	0.025/0.41	0.025/0.41	0.025/0.41
Heat Flow (Btu/hr) (1)	0	19.35	26.70
(2)	0.37	1.36	1.40
(3)	11.17	20.19	20.16
(4)	0.17	0.61	0.61
(5)	0.41	1.47	1.47
(6)	11.75	22.27	22.24
(7)	0.20	0.75	0.78
(8)	11.95	23.02	23.02
(9)	0.41	8.68	11.40
(10)	0.41	8.68	11.40
(11)	0	7.21	9.93
Heater Power (Btu/hr)	11.95	49.58	59.66
Side Wall Balance	0	0	0
Cold End Balance	0	0	0
"B" Cap Balance	0	0	0
Penetration Balance	0	0	+0.01

Heat Balance Equations

Heater Power = (1)+(2)+(3)+(9)
Side Wall Balance = (3)+(4)+(5)-(6)
Cold End Balance = (6)+(7)-(8)
"B" Cap Balance = (9)-(11)-(5)
Penetration Balance = Heater Power
- (1)-(11)-(8)

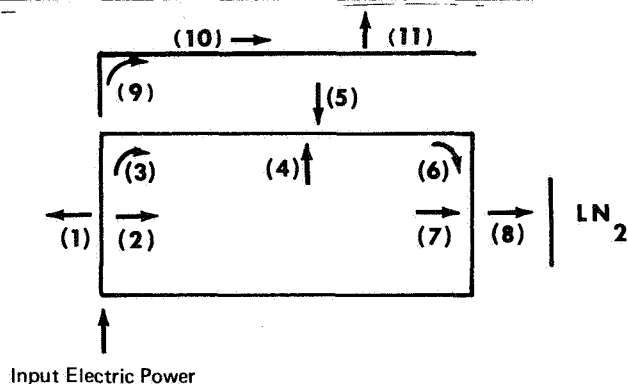


TABLE 18b
MATHEMATICAL MODEL RESULTS,
INSULATION SYSTEM NO. 18
(Power in Watts)

Computer Run No.	163	164	165
Test Simulation	IV-5A	IV-5B	IV-5C
Warm End Temp. (°R)	540	745	745
Cold End Temp. (°R)	142	148	147
Environment Temp. (°R)	540	540	140
Pen. Wall Thickness (in.)	0.027	0.027	0.027
Warm End Emittance	0.025	0.025	0.025
Cold End Emittance	0.025	0.025	0.025
Wall Emittance	0.025/0.41	0.025/0.41	0.025/0.41
Heat Flow (Watts)(1)	0	5.69	7.85
(2)	0.11	0.40	0.41
(3)	3.28	5.93	5.92
(4)	0.05	0.18	0.18
(5)	0.12	0.43	0.43
(6)	3.45	6.54	6.53
(7)	0.06	0.22	0.23
(8)	3.51	6.76	6.76
(9)	0.12	2.55	3.35
(10)	0.12	2.55	3.35
(11)	0	2.11	2.92
Heater Power (watts)	3.51	14.56	17.53
Side Wall Balance	0	0	0
Cold End Balance	0	0	0
"B" Cap Balance	0	0	0
Penetration Balance	0	0	0

Heat Balance Equations

Heater Power = (1)+(2)+(3)+(9)
Side Wall Balance = (3)+(4)+(5)-(6)
Cold End Balance = (6)+(7)-(8)
"B" Cap Balance = (9)-(11)-(5)
Penetration Balance = Heater Power
- (1)-(11)-(8)

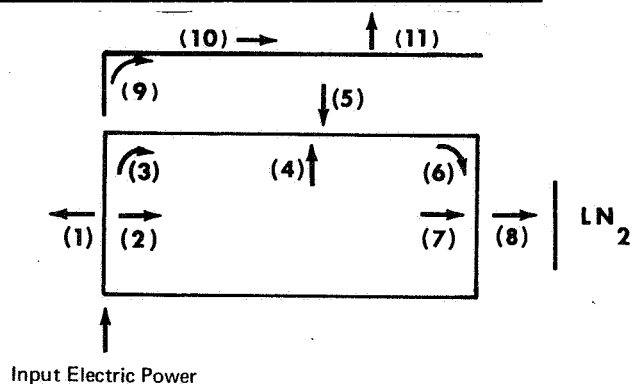


TABLE 19

SYSTEM NO. 18, COMPARISON OF PREDICTED AND EXPERIMENTAL RESULTS

TEST NO.	TEMPERATURE, °R			COMPUTED HEAT FLOW, Btu/hr(watts)			EXPERIMENTAL HEAT FLOW, Btu/hr(watts)				PERCENT DIFFERENCE
	Cap Baffle	Lower Baffle	Upper Baffle	Heater Power	Penetr. Effect	Heater Power	Calor.		Penetr. Effect		
							Boil-off	Insulation			
IV-5A	540	540	540	120 (3.5)	12.0(3.5)	9.9 (2.9)	22.5 (6.6)	11.2(3.3)	11.2(3.3)	+ 6	
IV-5B	745	540	540	99.7(14.6)	23.0(6.8)	37.8(11.1)	38.2(11.5)	11.2(3.3)	27.9(8.2)	-15	
IV-5C	745	140	140	59.6(17.5)	23.0(6.8)	42.6(12.5)	26.6(7.8)	0	26.6(7.8)	-11	

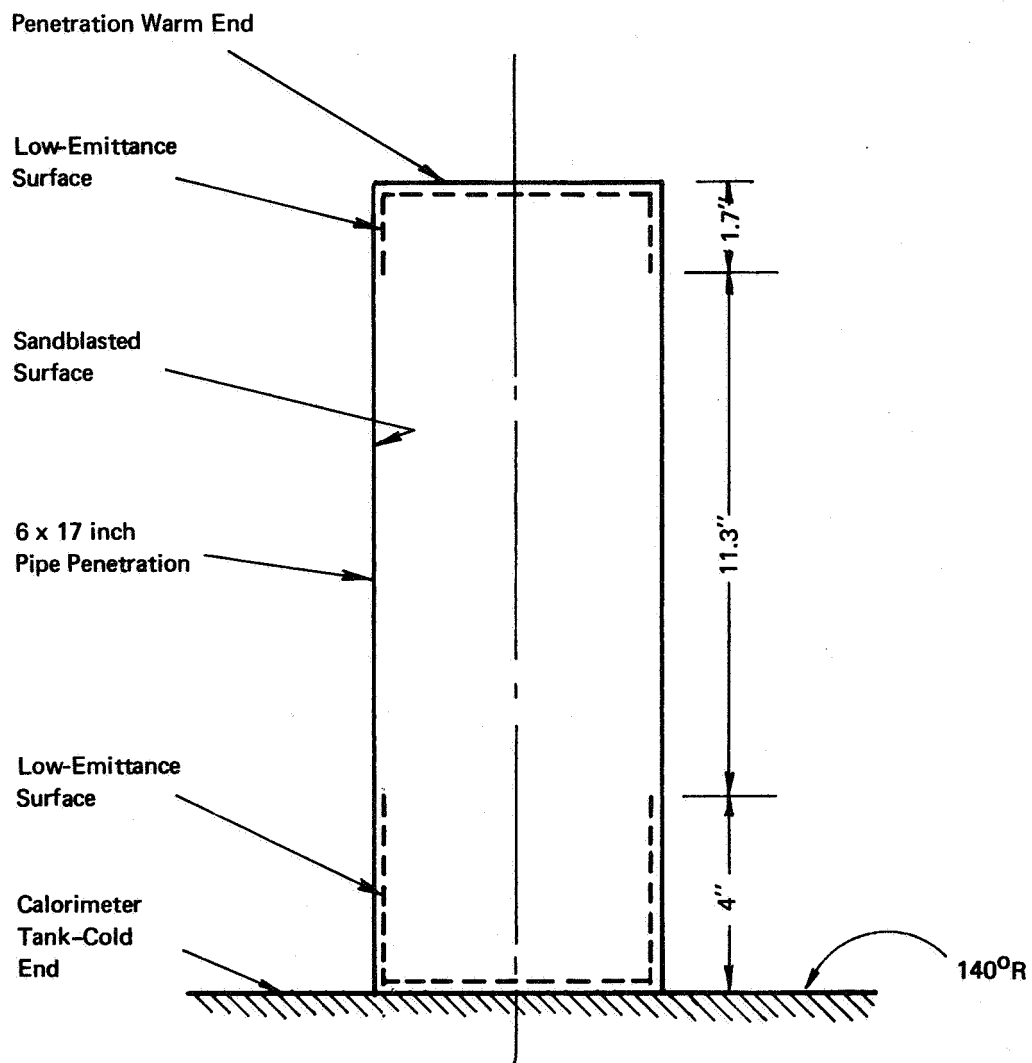
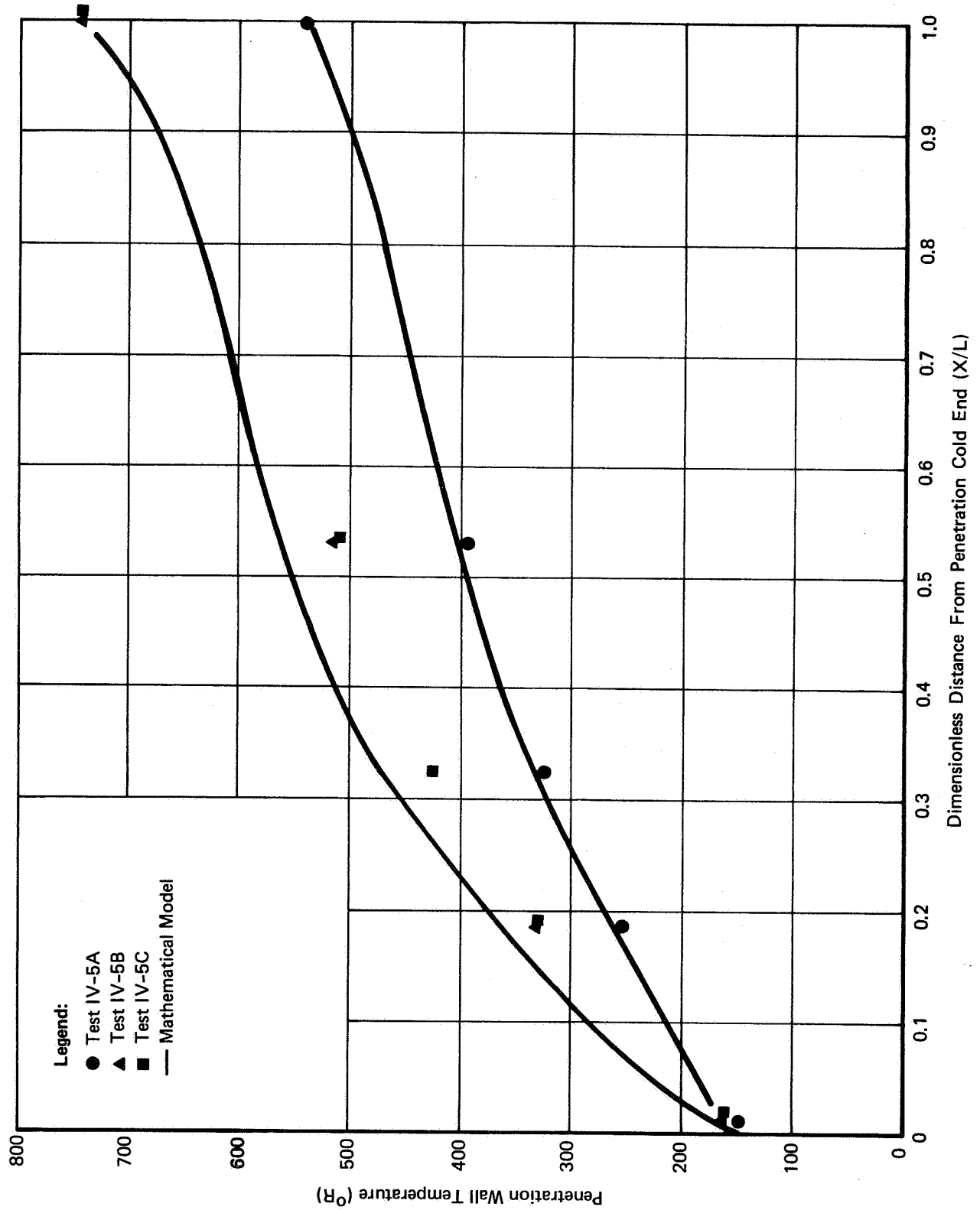


FIGURE 41 PENETRATION WITH LOW-EMITTANCE ENDS



**FIGURE 42 INSULATION SYSTEM NUMBER 18 - PENETRATION
WALL TEMPERATURES, TEST SERIES IV-5**

VI. GOLD SHIELD SYSTEM RETEST - TASK II

A. INTRODUCTION

A high-performance multilayer insulation was developed and tested by Arthur D. Little, Inc., under NASA/LeRC Contract NAS3-6283. This system consisted of five gold-coated Mylar shields and silk netting spacers. When applied to the tank calorimeter, it was identified as System No. 14. It produced a heat flux value of 0.33 Btu/hr-ft^2 on the calorimeter at cold and warm boundary conditions of 140°R and 540°R respectively; this is approximately 50% larger than the theoretical value based solely on the radiation flux as computed from the measured shield emittances. The insulation also had an excellent weight performance, as evidenced by a value of $0.00027 \text{ Btu-in. -lb/hr-ft}^5 \text{ }^\circ\text{R}$ obtained for the K_p products.

The current effort was undertaken to determine the stability of the insulation's thermal performance after a storage period of one year. Tests were also performed to establish system performance with liquid hydrogen for comparison with tests using liquid nitrogen. This work was performed in accordance with the experimental requirements established under Task II of the contract work statement.

In the present program, System 14 was retested with liquid nitrogen at the Arthur D. Little, Inc., test facilities. Subsequently, similar tests were performed at the NASA/Plum Brook Station, J-3 facility, with both liquid nitrogen and liquid hydrogen. Emissometer and flat-plate thermal conductivity apparatus measurements were also repeated with the original samples.

B. SYSTEM DESCRIPTION

Insulation System No. 14 consisted of five radiation shields and six spacers. The insulation applied to the calorimeter tank had an area of 39.5 ft^2 and an approximate weight of 1.2 pounds. The completed system is shown in Figure 43, and detailed characteristics are presented in Table 20.

1. Shields

The radiation shields consisted of $\frac{1}{4}$ -mil polyester film (Mylar) coated with gold on both surfaces. This material had been obtained from Hastings & Company, Inc., of Philadelphia in three different lots. Shields No. 1 and 2 were made up from 100 ft^2 of purchased sample material; this film had gold coatings approximately 2000 angstroms thick and an average emittance of 0.0160. Shields 3 and 4 and part of shield 5 were made up from a second purchase of 180 ft^2 of production material having coating thicknesses in the range of 2000 to 3000 angstroms and an average emittance of 0.0175. A small portion of shield 5 was made

up from a 50-ft² sample coated with 23K gold instead of the 24K gold used in the others. The 23K coatings had an inferior emittance but represented only about 2% of the total shield area. The average emittance for all shields was 0.0172; this value was established from the data of Table 21 on a weighted basis, taking into account the average emittance and surface area of each shield. Table 21 includes two measurements performed on a portion of a shield sample used in the flat-plate thermal conductivity apparatus measurement.

The gold coatings were of excellent quality. First, the coatings were of high brilliance, indicating small grain structure and low emittance. Second, the Scotch-tape test produced no coating lift-off. Third, after the gold coatings were subjected to various environments, ⁽¹⁾ tests showed that their emittance value was stable.

2. Spacers

Each spacer of the multilayer system was made up from three layers of silk netting. This material was purchased from the Jordan Marsh Company of Boston. It was applied to the calorimeter in the as-received condition, and no special treatments were given to it except for careful handling to prevent soiling and tearing. Additional information is given in Table 20.

3. Application

Application of the spacers and shields was practically identical to the procedure used with the aluminized MLI (see par. V-A-1-c). The only differences were that 1) each spacer had three layers of netting instead of two, and 2) the cylindrical segment of each shield was gored at intervals of 4-5 inches instead of 3 inches.

4. Storage

Subsequent to the tests performed with System No. 14 under the previous contract, the insulated calorimeter was in storage for a period of one year. For eight months of this period the 5-foot-diameter vacuum chamber was used as a storage container for the calorimeter. The chamber was at atmospheric pressure and had originally contained nitrogen gas. No special precautions were taken to insure a nitrogen atmosphere within the chamber during this period.

During the last four months of the storage period, the calorimeter was attached to the fixture used during fabrication of the insulation. Because this work was done in a fabrication shop area, the calorimeter was covered with polyethylene film to provide protection from coarse dust.

Prior to the current series of tests, the calorimeter was inspected. Some thermocouple lead wires were found to be broken at the vacuum feed-through connector. Also, the visible surface of the outer shield appeared to have a light coating of dust. However, in all other respects the calorimeter and insulation appeared to be in their original condition.

C. TEST RESULTS AND DISCUSSION

1. Tank Calorimeter

Three heat flux tests were performed with Insulation System No. 14. Two of these were performed with liquid nitrogen and the third with liquid hydrogen. The method of determining the heat flux was the same as that described in Section III.

a. Liquid Nitrogen Tests

The first liquid nitrogen test was performed at the Arthur D. Little, Inc., test facilities. Heat flux data were taken for a period of 117.5 hours at equilibrium conditions and with the chamber baffles maintained at 540°R. The experimental results obtained in this test, IV-1A, are summarized in Table 22, as are the results obtained in Test III-7 a year earlier.

A heat flux of 0.331 Btu/hr ft² was obtained from these tests. This exceeds the value of 0.325 Btu/hr ft² obtained one year previously by only 2%, which is within the experimental accuracy ascribed to the tank calorimeter measurements. The interpretation given to this result is that storage of the insulation system for one year had no deleterious effect.

The next liquid nitrogen test performed with System No. 14 was at the J-3 facility at NASA/Plum Brook Station. The insulated tank calorimeter was placed in a shipping container and moved by truck from Boston to the Plum Brook Station, an approximate distance of 500 miles. In the process, the system was undoubtedly subjected to a minimum of 10 hours of low-frequency vibration and moderate shock loadings. The system was inspected on arrival at Plum Brook Station. On the basis of a visual inspection the system was found to be undamaged; there was no indication that the insulation had shifted on the tank in any manner.

Test No. IV-1B was performed at the J-3 facility using the same experimental procedures as those followed at Cambridge. As indicated in Section III-B, the two facilities have basically similar equipment in size and function. Heat flux data were taken for a period of 60 hours at equilibrium conditions and with the chamber jacket and baffles maintained at 540°R. The experimental results are summarized in Table 22.

The heat flux measured in Test IV-1B was 0.321 Btu/hr-ft² - approximately 3% less than the value of 0.331 determined prior to shipment. As before, this difference is within the experimental accuracy ascribed to the calorimeter measurement. It is concluded, therefore, that the two tests produced identical results and that truck shipment had no deleterious effect on the multilayer insulation.

The most evident result of the comparison of the nitrogen tests is that the three thermal performance measurements are in excellent agreement. The insulation system was clearly not degraded either by storage for a period of one year or by road shipment from Boston to Cleveland and the related handling.

b. Liquid Hydrogen Results

Before Test IV-1C was run with liquid hydrogen in the calorimeter, a value was predicted for the heat flux based upon the previously measured radiation shield emittances and the experimental data obtained with liquid nitrogen. For boundary conditions of 140°R and 540°R and an average shield emittance of 0.0172 a value of 0.25 Btu/hr-ft² was computed for the radiation flux. As the actual flux measured with liquid nitrogen for these conditions was 0.33 Btu/hr-ft², the solid conduction component was about 0.08 Btu/hr-ft². The radiation heat flux is, theoretically, only negligibly affected when the test is performed with liquid hydrogen; however, an increase in the temperature across the insulations from 400 to 500°F brought about by the use of hydrogen should theoretically increase the conduction heat flux to 0.10 Btu/hr-ft². Thus, the predicted heat flux value for System No. 14 in tests with liquid hydrogen was 0.35 Btu/hr-ft².

The measured value of the heat flux obtained in Test IV-1C was 0.378 Btu/hr-ft². This was 8% greater than the predicted value. We have studied the data thoroughly and have found no specific cause or causes to account for the difference between the two values. Our findings are summarized below.

(1) Constants. We carefully reviewed data obtained in Test IV-1C and all the steps in the calculation of the heat flux value. All constants used in the calculation, such as latent heat of vaporization, density of the gas at standard conditions, and specific heat of hydrogen, were checked against other sources.

The latent heat of vaporization for equilibrium hydrogen used in our calculations was 190 Btu/lb -- a value obtained from Table 5.002 of the "Compendium of the Properties of Materials at Low Temperature," Volume I, by V.J. Johnson. Use of the Simon and Lange equation from the same reference yields the commonly used value of 194.4 Btu/lb for the latent heat of vaporization of hydrogen at its boiling point of 36.8°R, however, use of this higher value would only increase the difference between the predicted and experimental values.

(2) Baffle Temperatures. We investigated the possibility that average baffle temperatures differed between the tests. The baffle temperatures in the liquid hydrogen test were graphed as a function of time. The upper baffle averaged about 540°R and the lower baffle averaged about 539°R. Comparison of these temperatures with those prevailing at the time the liquid nitrogen tests were in progress showed them on the average to be about $\frac{1}{2}$ °R lower. This would tend to produce a lower heat flux in the hydrogen test, rather than the higher value observed.

(3) Vent Gas Flow. Another source of error could have involved the calibration of the gas meter used to measure the volume of boil-off gas from the calorimeter. However, during the preceding test performed with liquid nitrogen (IV-1B) the same meter was used in series with a wet test meter. For a total flow of 96 cu ft the wet test meter indicated 0.05 cu ft more than the dry gas meter; this represents agreement between the two meters of 0.05% and, therefore, does not account for the difference noted.

(4) Ortho to Para Conversion. The conversion of ortho to para hydrogen in the calorimeter can produce the same result as an added insulation heat leak. Tracing back the history of the liquid hydrogen used at Plum Brook, we found that the NASA purchase specification calls for a minimum 95% concentration of para hydrogen. When received, the liquid hydrogen is stored at NASA in a large spherical tank before being delivered to the test sites. The content of the storage tank is periodically tested, and the measured para concentration is found generally to have a value of 98 to 99%.

Should the liquid hydrogen used in the tank be 95% para hydrogen, the heat of conversion in the calorimeter tank would contribute 1.2 Btu/hr, or approximately 8% of the total heat flow rate measured during the test. If liquid hydrogen is 98% converted, the contribution is estimated to be 0.11 Btu/hr, or approximately 3/4% of the total heat flow rate.

(5) System Operation. For reasons of safety the liquid hydrogen line between the storage tank and calorimeter tank was not disconnected. A small gas leak through the closed valve would have caused an apparent increase in the calorimeter boil-off.

We also studied the effect of incorrectly estimated barometric pressure variation, the effect of hydrogen liquid level in the calorimeter neck at the beginning of the test, and other possible causes of inaccuracy, but their influence was found to be negligible.

(6) Shield Temperature Profiles. The temperatures of the shields in Tests IV-1B and IV-1C were compared. Figure 44 shows the temperature distribution for both tests; the results of the same measurements made one year prior are shown in Figure 45. A comparison of these plots shows that no significant changes took place in the interval between the two tests. This is further indication of the stable and reproducible performance of the System 14 MLI.

We also calculated the radiation heat fluxes between the shields, using experimentally measured shield temperatures and the known emissivities of the shields (0.0172) and tank wall (0.9). The calculation was made in two ways. First, temperatures for thermocouple "b" only were used, and the heat flux over an entire tank shield was based on this single value. Second, we plotted shield temperatures as a function of height from the top of the tank, and this allowed us to estimate the shield temperatures at various heights; from the latter values, a graphical integration was carried out to determine the total heat flux from local temperature values. The results obtained by both techniques were slightly different, but no trend was obvious. The simple use of thermocouple "b" to yield an average shield temperature appears to be quite satisfactory in most cases.

Table 23 shows the radiation heat fluxes between neighboring shields. The calculations are based on the known emissivity of the shields and on the temperatures indicated by thermocouples in location "b". The results for each test agree very well with the estimated overall radiation heat flux of 0.25 Btu/hr-ft². They also confirm that the two tests are identical with regard to radiation heat flux in the MLI.

In summary, therefore, we can find no demonstrable cause for the contradiction between the predicted and measured heat fluxes in the liquid hydrogen test. Likely causes are (1) a gas leak into the system through a closed valve and (2) conversion of ortho to para hydrogen during the progress of the test.

2. Thermal Conductivity Apparatus

In the previous contract, heat flow measurements were made on a 12-inch-diameter sample of System No. 14 multilayer insulation with the guarded flat-plate thermal conductivity apparatus (see Section III-A-2). These measurements, performed in March 1966, are summarized in Table 24. The zero heat flux measured for 140 and 540°R boundary conditions was 0.32 Btu/hr-ft². This was practically identical to the value of 0.33 Btu/hr-ft² obtained at that time with System 14 on the tank calorimeter.

The 12-inch-diameter sample was retested during the current program in March 1967, after storage for one year. From the first zero-load measurement performed in Test 3091a, a heat flux value of 0.32 Btu/hr-ft² was obtained,

which was identical to that measured one year prior. In the second zero-load test, 3091e, performed after loadings at 0.01 and 0.1 psi, the heat flux rose about 3% to 33 Btu/hr ft². This demonstrates the excellent recovery properties of the system.

The heat flux values obtained from the sample at loadings of 0.01 and 0.1 psi from the current tests are comparable to the values obtained a year ago.

The sample thickness at each measurement point is also summarized in Table 24. Note that the thickness at zero load reported for the current tests is about half of that measured in 1966. The initial thicknesses were large because the shield and spacer materials had creases and wrinkles that prevented close conformance of one layer to the next. However, after being stored flat for a period of even a few days, the creases and wrinkles become flattened, and the surface-contacting layers conform closely to one another. After one year of storage, this process reduced the sample thickness from 0.45 inch to about 0.2 inch. The latter is still some three times the theoretical thickness of the sample, which is estimated to be 0.065 inch.

3. Emissometer

A sample for emittance testing was cut from the top radiation shield of the 12-inch-diameter sample (#3077). The test showed the emittance of this sample to be 0.0164 compared with values of 0.0160 to 0.0172 obtained one year ago on various samples of the system insulation.

D. CONCLUSIONS

- (1) The thermal performance of MLI System No. 14 was not affected by storage in a shop-warehouse environment for a period of one year. In addition, its performance was unchanged after shipment over a distance of about 500 miles by moving-van-type carrier.
- (2) The thermal performance of System 14 measured with the tank calorimeter was confirmed by measurements performed in the guarded flat-plate thermal conductivity apparatus with a previously stored sample. The current measurements of shield emittance were also comparable to measurements performed a year earlier.

TABLE 20

INSULATION SYSTEM NO. 14, SPECIFICATION

Description Five radiation shields of polyester film metallized on both sides with gold and five inner spacers, each consisting of three layers of silk netting. The sixth (outer) spacer consisted of one layer of silk netting; the multilayer was applied over Calorimeter No. 2.

Shield

Material	DuPont, Type S polyester film
Thickness	Film 0.00025 inch; gold coatings, 1000 to 3000 angstroms
Emittance	0.0172 average for all shields
Geometry	Sides, straight gores 4 to 5 inches wide; head, full circles
Joints	All seams overlapped and contacting except for gored elements, which are butt-jointed.

Spacer

Material	Three layers of silk netting, Jordan Marsh No. 5517
Thickness	0.0105 inch (0.0035 inch per silk layer)
Weight	0.0036 lb/sq ft
Density	4.1 lb/cu ft
Open Area	84%
Geometry	Side, gored edges of side sheet; head, full circles
Joints	All joints butted

Boundary Emittance

Calorimeter Tank	0.86 at 140°R
Chamber Baffles	0.93 at 540°R

System Properties

Surface Area	39.5 sq ft
Weight	0.0282 lb/sq ft

TABLE 21

INSULATION SYSTEM NO. 14,
SHIELD EMITTANCE DATA SUMMARY,
GOLD-COATED 1/4-MIL POLYESTER FILM

<u>Date</u>	<u>Sample From System/Shield</u>	<u>Sample No.</u>	<u>Roll No.</u>	<u>Coating Thickness, angstroms</u>	<u>Emittance Test No.</u>	<u>Emittance</u>
3-4-66	14/1	351B	122265	2440	542	0.0201
3-7-66	14/1	351A	122265	2080	545	0.0145
1-12-66	14/1	330B	122265	2072	465	0.0127
1-11-66	14/1	330A	122265	1840	463	0.0146
3-15-66	14/2	352A	122265	2680	550	0.0176
3-16-66	14/2	352B	122265	3340	551	0.0166
3-15-66	14/2	353B	21166	3360	549	0.0173
3-10-66	14/3	353A	21166	2200	546	0.0195
2-23-66	14/3	343B	21166	3100	527	0.0176
2-22-66	14/3	343A	21166	2220	526	0.0190
3-14-66	14/4	354B	21166	2140	548	0.0146
3-10-66	14/4	354A	21166	1740	547	0.0166
1-24-66	14/5(circle)333		101565	953	486	0.0259
2-3-66	14/5(circle)336		101565	1050	503	0.0228
3-9-67	3091*	362	----	----	696	0.0164
4-26-67	3091	362	----	----	717	0.017

NOTE:	Average emittance shields 1 and 2	0.0160
	Average emittance shields 3, 4, 75% of 5	0.0175
	Average emittance shield 5 (25%)	0.0243
	Computed average emittance - all shields	0.0172

* Emissometer sample taken from one of five shields of flat-plate thermal conductivity apparatus sample 3091 after completion of Test 3091e.

TABLE 22

DATA SUMMARY, INSULATION SYSTEM NO. 14, TEST SERIES IV-1, CALORIMETER NO. 2

Insulation System: Five radiation shields of polyester film metallized on both sides with gold and six spacers, each consisting of three layers of silk netting except for outer spacer, which consisted of one layer of silk netting.

Test Facility: NASA/Plum Brook Station test facility and Arthur D. Little, Inc., test facility

Penetrations or Gaps: None

Boundary Conditions: Warm boundary: 540°R, 0.93 emittance; cold boundary: Tests IV-1B, IV-1A and III-7: 140°R, Test IV-C: -37°R, 0.86 emittance

Tank Surface Area: 39.5 sq ft

Test No.	Tank Liquid	-----Data Period----- Date & Time to Date & Time	Test Duration, hrs	Avg. Vac., torr	Guard Liq. Temp., °R	Warm Boundary, °R	Total Heat Flow, Btu/hr	Measured Heat Flux, Btu/hr-sq ft
III-7*	LN ₂	3-17-66/0830 3-21-66/0830	96	2×10^{-6}	149	540	12.8	0.325
IV-1A*	LN ₂	2-23-67/1600 2-26-67/2030	76.5	6×10^{-6}	148	540	13.0	0.330
	LN ₂	2-27-67/1830 3-1-67/1230	42	5×10^{-6}	148	540	13.2	0.333
IV-1B**	LN ₂	3-21-67/0900 3-23-67/1100	50	2×10^{-6}	149	540	12.7	0.321
IV-1C**	LH ₂	3-27-67/2100 3-30-67/0900	60	5×10^{-7}	37	540	15.7	0.378

* Test results obtained at the ADL facility. Test III-7 performed under Contract NAS 3-6283.

** Test results obtained at the NASA/Plum Brook Station, J-3 Facility.

TABLE 23
CALCULATED RADIATION HEAT FLUXES
FOR RUNS IV-1B AND IV-1C
 (Btu/hr-sq ft)

<u>POSITION</u>	<u>IV-1B</u>	<u>IV-1C</u>
Wall - Shield 1		
From TC "b"	0.22	0.24
Integrated Value	0.23	0.24
Shield 1 - Shield 2		
From TC "b"	0.30	0.27
Integrated Value	0.28	0.26
Shield 2 - Shield 3		
From TC "b"	0.24	0.26
Integrated Value	0.27	0.27
Shield 3 - Shield 4		
From TC "b"	0.28	0.27
Integrated Value	0.27	0.28
Shield 4 - Shield 5		
From TC "b"	0.25	0.24
Integrated Value	0.26	0.25

TABLE 24

TESTS OF A MULTILAYER INSULATION WITH GOLD COATED

RADIATION SHIELDS

Sample No. 3077 and 3091*, Tank Insulation System No. 14

Sample Description: Five radiation shields of 0.00025-inch-thick x 11-inch-diameter polyester film, gold coated on both sides, and eighteen 12-inch-diameter, 0.003-inch-thick silk netting. (Three layers of silk netting per spacer.)

Sample Weight: 0.03 pound

Sample Chamber Pressure: Below 5×10^{-5} torr

Warm- and Cold-plate emittance: 0.9

Warm-plate Temperature: $540^{\circ}\text{R} \pm 5^{\circ}\text{R}$ (except during run 3077a where temperature was 530°R)

Cold-plate Temperature: 140°R

MECHANICAL COMPRESSION, psi	HEAT FLUX, Btu/hr-sq ft	SAMPLE THICKNESS, in.	TEST DATE	TEST NO.
0	0.28	0.453	3-6-66	3077a
0	0.32	0.453	3-8-66	3077b
0.01	0.55	0.315	3-8-66	3077c
0.10	2.11	0.090	3-8-66	3077d
0	0.32	0.208	3-6-67	3091a
0.01	0.45	0.171	3-12-67	3091b
0.10	2.32	0.087	3-14-67	3091c
0.01	1.35	0.098	3-15-67	3091d
0	0.33	0.198	3-16-67	3091e

* Sample 3091 is sample 3077 after 1 year storage.

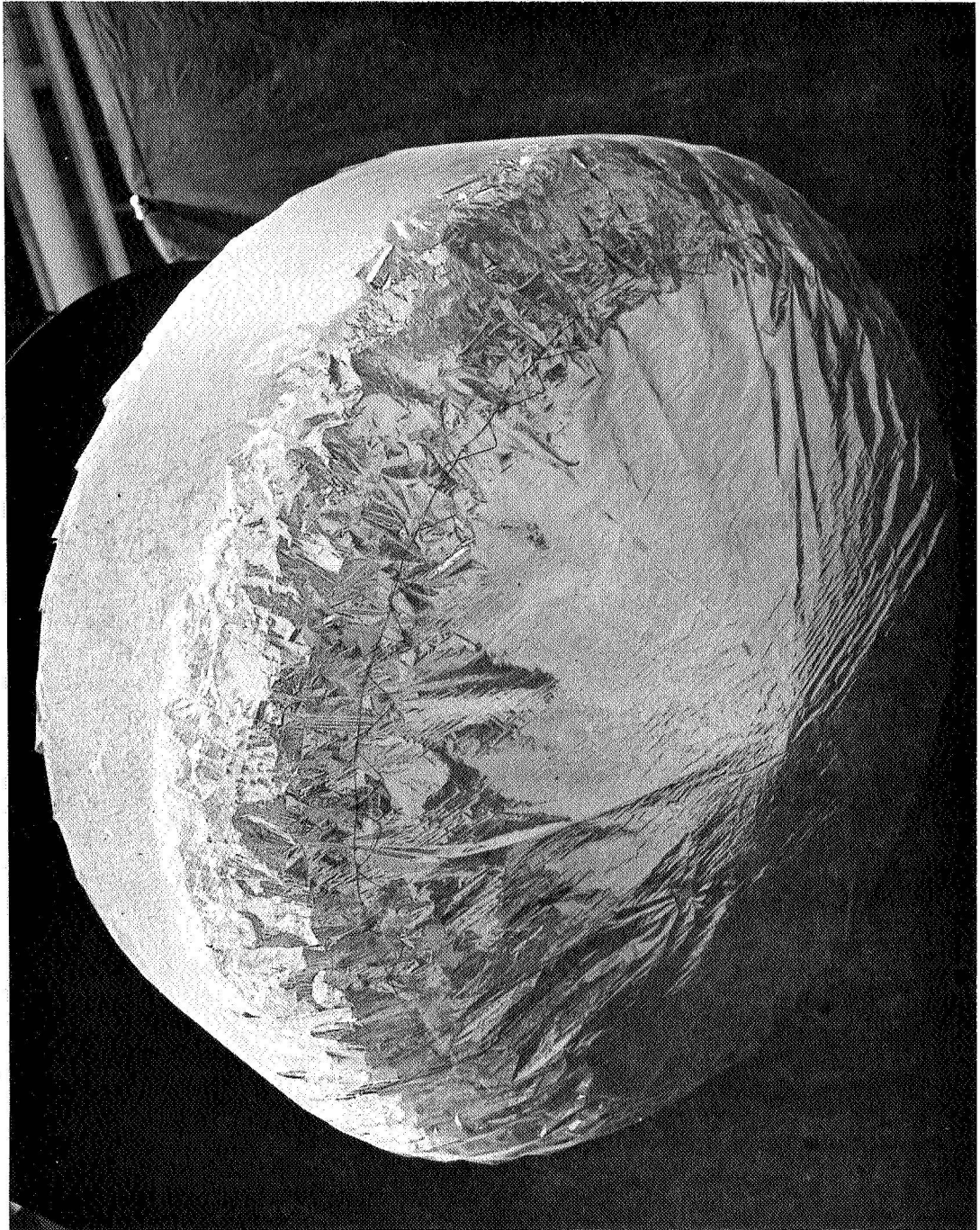


FIGURE 43 INSULATION SYSTEM NO. 14

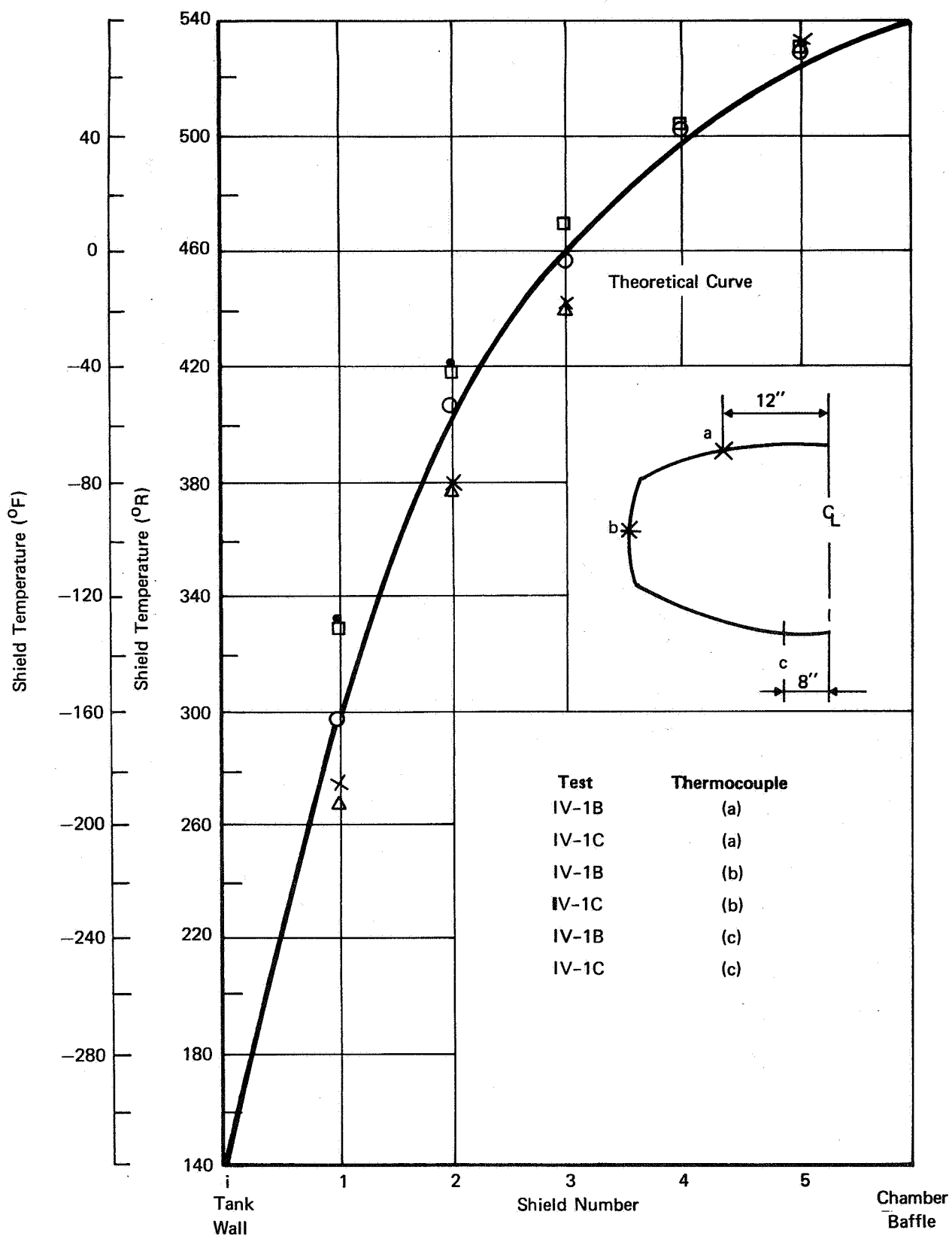


FIGURE 44 INSULATION SYSTEM NUMBER 14 – SHIELD TEMPERATURE DISTRIBUTION, TEST SERIES IV- 1

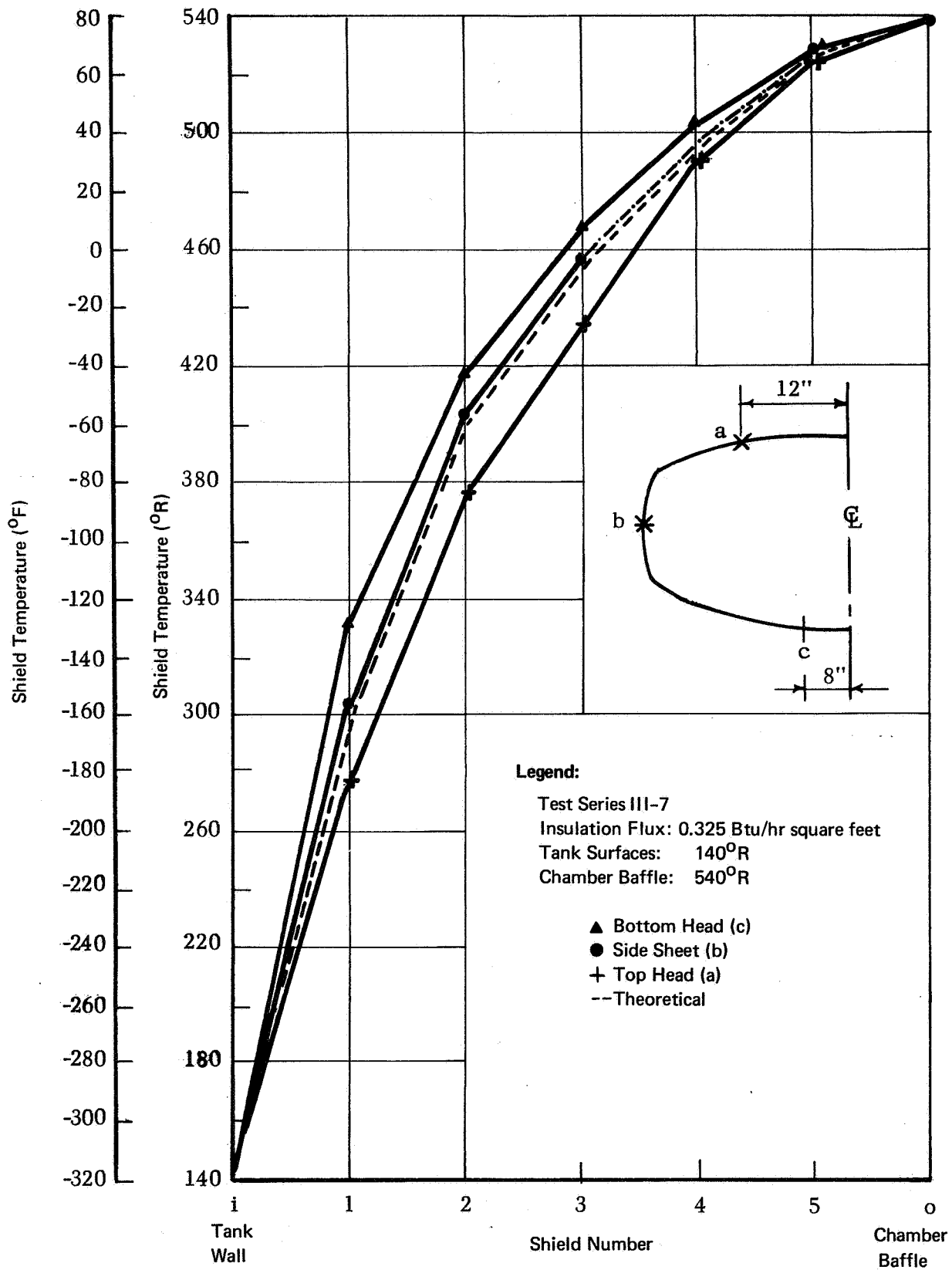


FIGURE 45 INSULATION SYSTEM NUMBER 14, SHIELD TEMPERATURE DISTRIBUTION, TEST SERIES III-7

REFERENCES

1. Bonneville, J. M., Design and Optimization of Space Thermal Protection for Cryogens -- Analytical Techniques and Results, Arthur D. Little, Inc., Report 65958-02-01, NASA CR-54190 for NASA Contract NAS3-4181.
2. Advanced Studies on Multilayer Insulation Systems, Arthur D. Little, Inc., Report 67180-00-04, NASA CR 54929.
3. Price, J. W. and Lee, T. G., Analysis, Design, and Testing of Heat-Short-Isolation Components for High-Performance Insulation Systems, Advances in Cryogenic Engineering, Vol. 12, K. D. Timmerhaus, Editor, Plenum Press, N. Y., 1967.
4. Sterbentz, W. H. and Baxter, J. W., Thermal Protection System for a Cryogenic Spacecraft Propulsion Module, Technical Report, LMSC-A794993, NASA CR-54879.
5. Basic Investigations of Multilayer Insulation Systems, Arthur D. Little, Inc., Report ADL 65958-00-04, NASA CR-54191.
6. Oppenheim, A. K., Radiation Analysis by the Network Method, Transactions of the ASME, May 1956, pp. 725-735.
7. Strong, P. F. and Emslie, A. G., The Method of Zones for the Calculation of Temperature Distribution, ASME 65-WA/HT-47.

PRECEDING PAGE BLANK NOT FILMED.
APPENDIX A

DETERMINATION OF DIFFUSENESS OF SANDBLASTED METAL SURFACES

I. INTRODUCTION

The work under Task I of Contract NAS3-7974 was performed to establish the prediction techniques for the flow of heat in a multilayer insulation pipe penetration. The transfer of heat from the warm end to the cold end of the penetration takes place both by conduction in the wall of the penetration and by multiple reflections inside of the penetration cavity. Depending upon the surface characteristics, the internal reflections can be diffuse (i.e., Lambertian or cosine), specular, or most commonly-combinations of these two. The manner in which the surfaces reflect incident radiation has an effect on the warm-to-cold-end heat transfer and on the analytical techniques used for predicting this heat transfer.

At the beginning of the current program the interior surfaces of the pipe penetration test article were sandblasted. The assumption was made at that time that this type of surface would have the property of a diffuse reflector. As the experimental work progressed, certain anomalies arose between the experimental and analytical results which required confirmation of this property. Through use of a goniometer instrument developed by Arthur D. Little, Inc., a series of experiments were performed subsequently to provide a basis for this confirmation.

II. SUMMARY OF RESULTS

On the basis of indirect but certain deductions from visible light measurements, the sandblasted aluminum, stainless steel, and copper surfaces appeared to be effectively diffuse (i.e., Lambertian or cosine) reflectors.

III. DESCRIPTION OF EXPERIMENT

A. EXPERIMENTAL APPROACH

Flat discs of the same materials as the pipe wall and sandblasted in the same way were prepared and measured in an optical instrument, a three-dimensional optical goniometer that determines angular distribution of reflected monochromatic light from the samples with a single narrow beam incident onto

the sample. The coordinate system of this goniometer is shown in Figure A-1. In such an experiment, diffuse reflection has a known and recognizable reflected radiation angular distribution (cosine). Deviation from the diffuse distribution can be detected as a relative intensification of reflected radiation in the general direction of specularity.

The surface roughness of the sandblasted and other samples was evaluated optically with a microscope as discussed in Section IV below.

B. APPARATUS

The goniometer and its operation have been described previously (A1, A2). Briefly, the apparatus measures the distribution of angular reflection produced by a sample that is illuminated by a narrow-beam source at a fixed incident angle at one wavelength. The apparatus consists of a three-dimensional goniometer based on a spherical coordinate system as indicated in Figure A-1, with the sample located at the origin. The sample is flat and is illuminated by a 1-cm-diameter beam of light smaller than the sample, so that all incident radiation is either reflected, absorbed, or transmitted by the sample and none falls off the edge of the sample. A monochromatic radiation collector, operating on a fixed radius 9.06 inch (23 cm) from the sample center, can be located in (almost) any position in the hemisphere surrounding the sample so as to collect reflected radiation at any chosen angular position with respect to the sample.

In addition to measuring the angular distribution of reflected radiation from a sample, the goniometer also measures the spectral reflectance of the sample (reflected spectral flux divided by incident spectral flux) by a process in which the individual goniometer readings are summed to give sample reflectance (described in references A1 and A2).

The apparatus is capable of measuring reflectance at any wavelength from 0.2 to 1.0 micron. The measurements reported here were made at a wavelength of 0.65 micron (visible red).

C. MEASUREMENT PROCEDURE

The usual mode of operation of the goniometer instrument is to lock the incident beam in a fixed position with respect to the sample (in these experiments, 33° from the normal to the sample) and then to locate and perform measurements with the collector in a series of fixed angular positions in the hemisphere surrounding the sample. For best accuracy the necessary number of readings is large but can be reduced by symmetry considerations. For most types of samples, reflection is found to be equal in the two halves of the polar space on either side of the plane of incidence. Therefore, the number of readings may be reduced by a factor of two by obtaining readings on one side of the

plane of incidence and assuming reflection on the other side to be of equal amplitude and angular distribution. This assumption is accurate for all samples that do not have a lay due to machining or forming.

Data were obtained for the reflected radiation at 22 measurement angles on one side of the incident plane distributed at equal angular intervals in the half-hemisphere. The reflection reading taken at each of these angles was recorded on a laboratory data sheet and reduced by computer to a directional intensity and also to a (net) value of directional spectral reflectance for the sample.

IV. RESULTS

A. SAMPLES

Three sample materials were tested for diffuseness: aluminum, stainless steel, and copper. A sample of each was prepared consisting of a flat disc 2.5 inches in diameter by 1/8 inch thick. Each was dry sandblasted on one side, washed with water, dried, and stored in a plastic bag for optical goniometric (and thermal emittance) testing.

The roughened surface of each disc was produced by holding the sample approximately 10 inches from a sandblast nozzle and perpendicular to the air stream for a period of about 30 seconds. During the sandblast operation the sample was moved and rotated to average out any unequal distributions in the grit stream. The result was a very evenly roughened surface with no visually apparent systematic variation of roughness. The grit used for blasting is designated commercially as "washed fine beach sand" and is shown in the photomicrograph of Figure A-6. The arithmetical average diameter of this grit for the 22 particles in the micrograph is 220 microns.*

The roughness of each of the three sample discs is shown in the photomicrographs of Figures A-2, A-3, and A-4.

The three sandblasted surfaces contained both large-scale roughness and finer-scale roughness within the large pores. As discussed subsequently, the large-scale roughness is most significant, because it bears on the diffuseness of these surfaces to thermal radiation. Therefore, the arithmetic average diameter of the large-scale roughness (pores) found at random locations across the surface was taken to be the significant measure of roughness.

*A screen grade rating of this sand has not been reported here because an ambiguity exists in the commercial designations of various vendors.

The large-scale roughness (pore diameter) averaged for ten pores at random locations was measured for each of the roughened materials. The roughness diameter dimensions and the sample designations are shown in Table A-1.

For comparison, the copper sample disc was measured on the un-sandblasted side (side B). This side had a standard ground finish with the appearance of a rough polish, i.e., glint; some images were discernible, but it was very poor as a mirror. Under a microscope at 100 power this surface appeared as a featureless plain with bright narrow scratches, as seen in Figure A-5. By contrast, the sandblasted surfaces resembled a randomly rough landscape when viewed at the same magnification. Under microscopic examination at 400 power and 1000 power, the ground finish showed a typical pore size of less than 2-1/2 microns.

One additional sample, an MgO smoked platen, was prepared because its preparation procedure is standard and reproducible, and because the reflection of the resulting surface is well known and characteristically very diffuse to visible light. The surface appears extremely white, uniform, and diffuse, and has a reflectance to red light reported in the literature (and also measured in our apparatus) ranging between 0.96 and 0.98. We measured a reflectance of 0.98 for the particular sample used in these tests.

B. MEASURED DIFFUSENESS TO VISIBLE LIGHT

The measured values of spectral reflectance for the sandblasted, ground-finish, and MgO samples are shown in the first column of Table A-2. The MgO white sample had a nearly unity reflectance. The ground finish copper had a reflectance of 0.67. The effect of sandblasting alone was apparent in that the same copper disc, sandblasted, had a reflectance of only 0.44. The sandblasted aluminum had identically the same reflectance to red light as sandblasted copper. Sandblasted stainless steel had the lowest reflectance of all the tested materials--0.31.

The third column of Table A-2 lists the percentage of the reflectance of each sample that is estimated to be diffuse. For present purposes, this estimate was made by a simple graphical method and defined by an easily measured ratio. This ratio may also be called the diffuse fraction and is the ratio of "diffuse reflection" to "all the reflected radiation" (diffuse and non-diffuse included). The "diffuse" fraction is Lambertian or cosine.

A narrow incident beam illuminates a small spot on a sample. The reflected light is represented as a set of vectors, the directions of which point in the direction of each reflected light ray leaving the illuminated spot. The lengths of each vector are proportional to the light intensity in that direction. For this geometry a "diffuse" (cosine) reflector will generate a set of vectors

the outer ends of which lie in a spherical surface, the whole set having the appearance of a spherical porcupine that is just touching the sample surface at the illuminated spot. Unlike the diffuse sample, a specular sample will show a narrow bundle of long vectors in the plane of incidence and reflecting off in the specular direction. Most real materials are neither fully diffuse nor fully specular; they show a nonspherical porcupine shape bulged elliptically in the direction of specularity (pear shaped).

The purpose of the tests was to estimate the fraction of sample reflectance that could be considered "diffuse." The diffuse fraction was estimated graphically by determining the size of the vector-end sphere that would just fit inside the approximately ellipsoidal surface containing the vector ends of the plotted sample reflection. This vector-end sphere represented the diffuse fraction of the sample's reflectance. The diffuse fraction was determined by taking the ratio of the length of a diffuse vector in the specular direction to the specular vector in the specular direction. The numerator of the ratio (diffuse fraction) was found by measuring the length of any vector terminating in the diffuse sphere having the same zenith angle ($\phi = 33$) as the specular beam but having a distinctly different azimuthal angle θ well away from the plane of incidence. For these measurements a convenient choice was $\theta = 100^\circ$. Any other value of θ away from the plane of incidence would give the same reading. The numerator so determined gives essentially the diffuse intensity alone in the specular direction as though the specular component were not present.

The denominator of the ratio -- the intensity reading in the specular direction -- was measured directly. The ratio so defined has the form:

$$f_d = \frac{\rho_d}{(\rho_d + \rho_s)}$$

f_d = fraction of reflectance which is estimated to be diffuse,

ρ_d = diffuse reflectance in the specular direction inferred by a measurement out of the plane of incidence,

ρ_s = specular reflectance in the specular direction (in the plane of incidence),

$(\rho_d + \rho_s)$ = the intensity actually measured in the specular direction.

This formula gives a first approximation to the fraction of net sample reflectance which is diffuse. A more refined basis would take into account a summation of all readings at all zenith and azimuth angles, but the above formula is adequate for the present purposes.

As indicated in Table A-2, f_d is 1.00 for a theoretically perfect diffuse reflector, 0 for a theoretically perfect specular reflector, and some intermediate value for a real sample. Thus, the MgO smoke sample tested had a diffuse fraction of 99%, the shiny mill-finish copper was found to have a diffuse fraction of 1%, and the same copper sandblasted had a diffuse fraction of 80%.

The three sandblasted samples tested had diffuse fractions ranging between 60% and 80%.

The second column in Table A-2 lists the diffuse component of reflectance and is computed for each sample by multiplying the reflectance value listed in the first column by the diffuse fraction listed in the third column.

C. MEASURED DIFFUSENESS TO THERMAL RADIATION

The degree to which the samples tested act diffusely to thermal radiation can be deduced from their diffuseness to visible radiation. Although not usually possible, this deduction can be made here because the dimensions of the large-scale roughness of the sandblasted samples are comparable to the longest thermal wavelength of interest. Thermal radiation (for 300°K) has its peak at 10 microns and extends between 6 microns and 40 microns, with 5% irradiance remaining in each of the Planck-law tails. As seen from Table A-1, the large-scale (pore) roughness of all of the samples tested is comparable with the longest thermal wavelength of interest. The argument may be applied that a random rough surface of large characteristic dimension (e.g., similar to shiny, metal-plated gravel) will behave 1) as a diffuse reflector, by virtue of a large number of randomly oriented, small, specular surfaces, and 2) diffusely independently of wavelength for all wavelengths of incident radiation smaller than the characteristic roughness dimension of the pores.

The samples tested here approximate such surfaces. Under a microscope they show randomly oriented and shiny facets and pores of characteristic dimension slightly larger than 40 microns. Accordingly, we have concluded that, to the level of approximation required here, the diffuse fraction deduced from the visible light measurements apply as well to the reflection behavior of these samples to thermal radiation. Therefore, the diffuse fraction of these samples to thermal radiation may be taken as the values given in the third column of Table A-2.

For an additional reason, the effective diffuseness of the three sandblasted surfaces tested is actually higher than indicated in Table A-2. Table A-2 is based on dividing reflectance into two parts -- "diffuse" and "other including specular." For the stainless steel sample, for example, 60% is diffuse and 40% is "other including specular." We have not in the present work determined how much is contained in the specular fraction alone, but we estimate that it is small; for all three sandblasted samples the specular component alone is believed to range from 5% to 10% of the sample reflectance. On this basis, the stainless steel sample, for example, would have a breakdown of reflectance components as follows:

	<u>Percent</u>	<u>Component of Reflectance</u>
Diffuse	60	0.19
Estimated specular	5-10	0.015-0.030
"Other"	<u>30-35</u>	<u>0.09 -0.11</u>
Total reflectance components	100	0.31

The "other" portion makes up a substantial component of reflectance. It consists of scattered radiation which is nevertheless not completely (cosine) scattered and therefore is not strictly diffuse.

The "other" component does in effect behave more like diffuse radiation than specular. In terms of reflective propagation down a pipe, the average specularly reflected ray propagates in a few bounces. The experience of a typical diffuse ray, however, involves possibly an order of magnitude more bounces. In these terms a typical path included in the component called "other" is believed to be scattered a number of times, more closely approaching the diffuse regime than the specular; therefore, the "other" component of reflectance contributes an additional component which, in effect, acts diffusely. In the case of the stainless steel surface, the effective diffuse behavior probably lies somewhere between 60% and 90% and is probably 80-90% effectively diffuse.

V. CONCLUSIONS

(1) The diffuse behavior of all three samples as established by optical measurements is also the behavior that may be expected at thermal (540°R) radiation wavelengths. As indicated in Table A-2, the diffuse fraction of reflectance to thermal radiation ranges from 60% to 80%.

(2) In terms of the heat pipe model, the sandblasted surfaces represented by the three test samples are believed to be effectively

more diffuse than 60% to 80%, due to an additional component of scattered radiation which, though not "diffuse" (Lambertian or cosine), is nevertheless not specular. The overall effective diffuseness of these samples was not measured or calculated, but we estimate that the incident energy had an effective diffuse fraction of 80 to 90%.

- (3) By inference, a pipe of aluminum, stainless steel, or copper, dry-sandblasted internally with fine beach sand as prescribed, acts substantially as a diffuse-walled pipe (80-90% diffuse) to the propagation of thermal radiation associated with temperatures of the order of 540°R.
- (4) A more precise determination of the diffuse fraction and a determination of the specular fraction would require a more complex mathematical analysis and computation but no additional data; the existing laboratory data sheets for these samples contain the necessary information for a more accurate determination of the diffuse and specular fraction if such numbers are required in the future.

TABLE A-1

ROUGHNESS MEASUREMENTS OF
SANDBLASTED SAMPLES

<u>Sample</u>	<u>Average Large-Scale Roughness Dimension</u>	<u>Estimated Large- Pore Depth</u>
Aluminum 1100-H14 sandblasted, Sample #372, Side A	56.5 microns (2,240 microinches)	(1/2 of diameter), 28.2 microns (1,120 microinches)
Stainless Steel 304, sandblasted, Sample #375, Side A	44.5 microns (1,780 microinches)	(1/3 of diameter), 14.83 microns (594 microinches)
Copper, sandblasted, Sample #376, Side A	48.5 microns (1,940 microinches)	(1/2 of diameter), 24.2 microns (970 microinches)
Copper, mill finish as received, Sample #376, Side B	< 2.5 microns (100 microinches)	

TABLE A-2

SPECTRAL REFLECTANCE
AND
PERCENT OF REFLECTANCE WHICH IS DIFFUSE

Sample Material and Designation	Incident Angle $\psi = 33^\circ$ Wavelength $\lambda = 0.65\mu$		
	Reflectance	Diffuse Reflectance	% Diffuse
Aluminum 1100-H14 sandblasted, sample #372, side A, experiment No. 3	0.44	0.33	75
Stainless 304 sandblasted, sample #375, side A, experiment No. 2A	0.31	0.19	60
Copper sandblasted, sample #376, side A, experiment No. 1A	0.44	0.35	80
Copper, ground finish, sample #376, side B, experiment No. 1B	0.67	0.007	1
MgO smoked plate, experiment No. 44	0.98	0.97	99
Theoretical 100% diffuse reflector (by definition)	(1.00)	(1.00)	(100)
Theoretical 100% specular reflector (by definition)	(1.00)	(0)	(0)

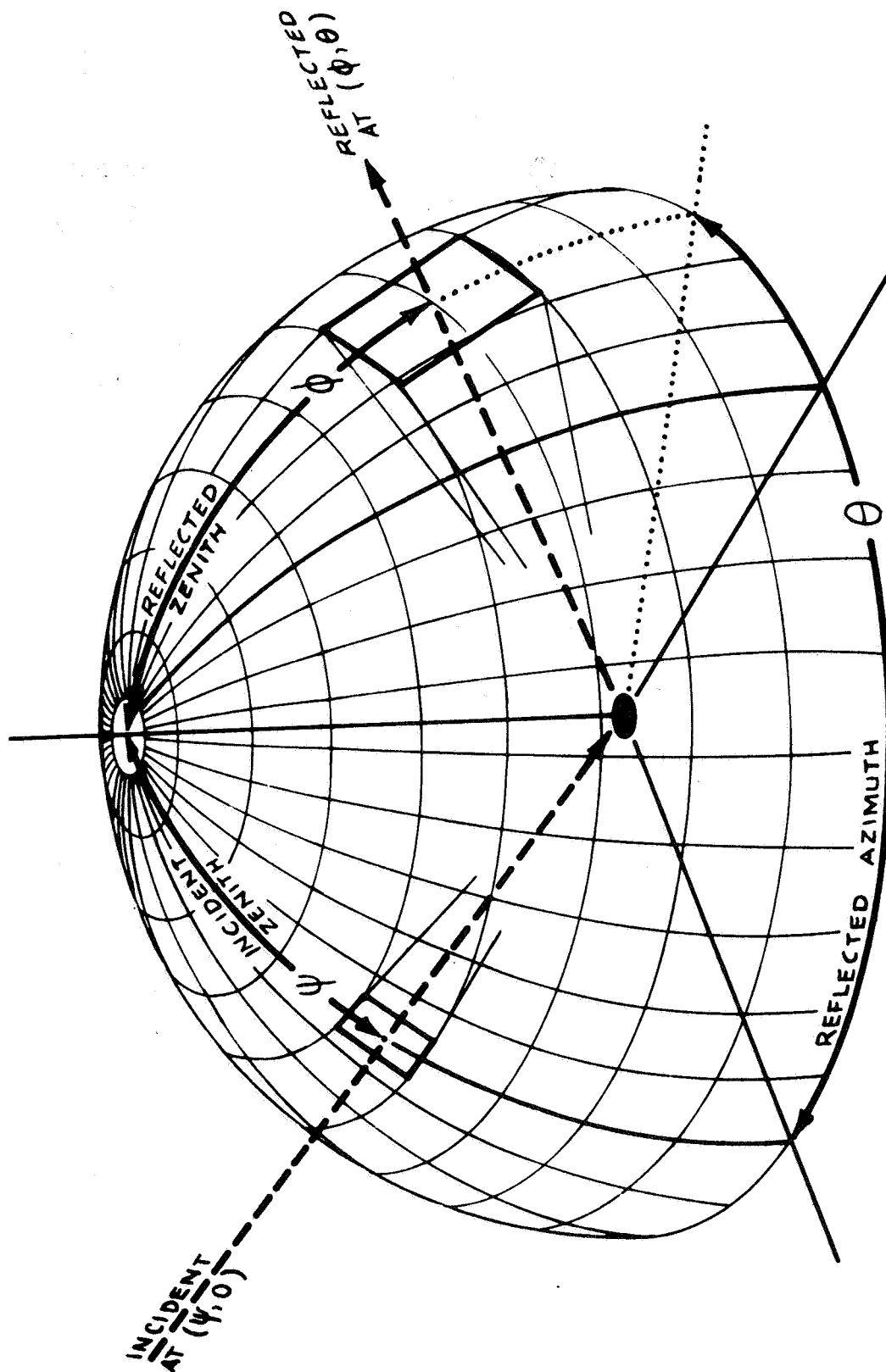
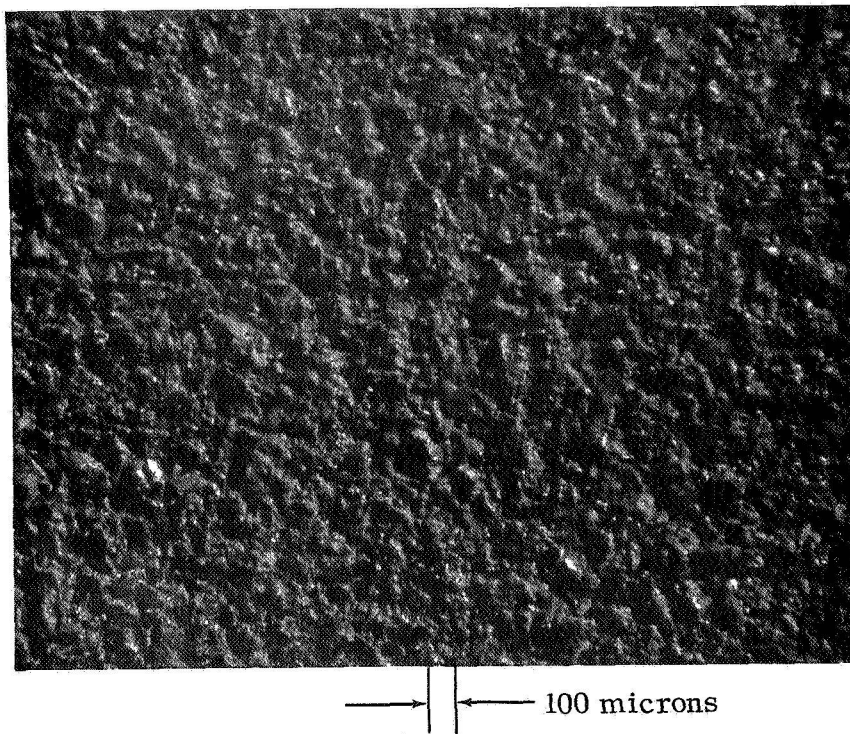
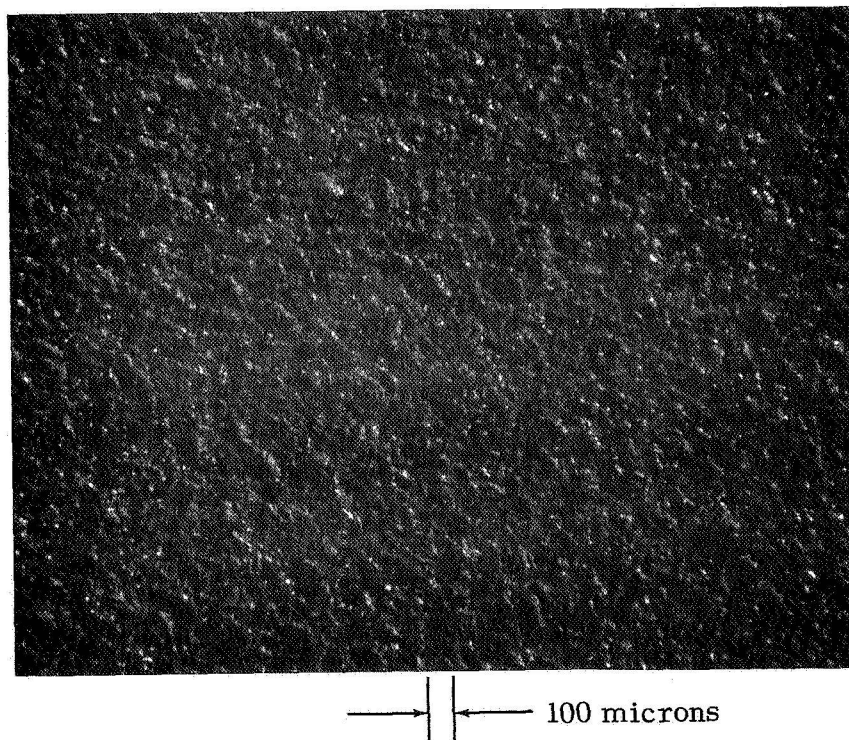


FIGURE A-1 REFLECTANCE COORDINATE SYSTEM



**FIGURE A-2 ALUMINUM, SANDBLASTED (SAMPLE NO. 372,
SIDE A)**
Average Large Scale Roughness—Dimension 56 Microns



**FIGURE A-3 STAINLESS STEEL, SANDBLASTED (SAMPLE NO. 375,
SIDE A)**
Average Large Scale Roughness—Dimension 44 Microns

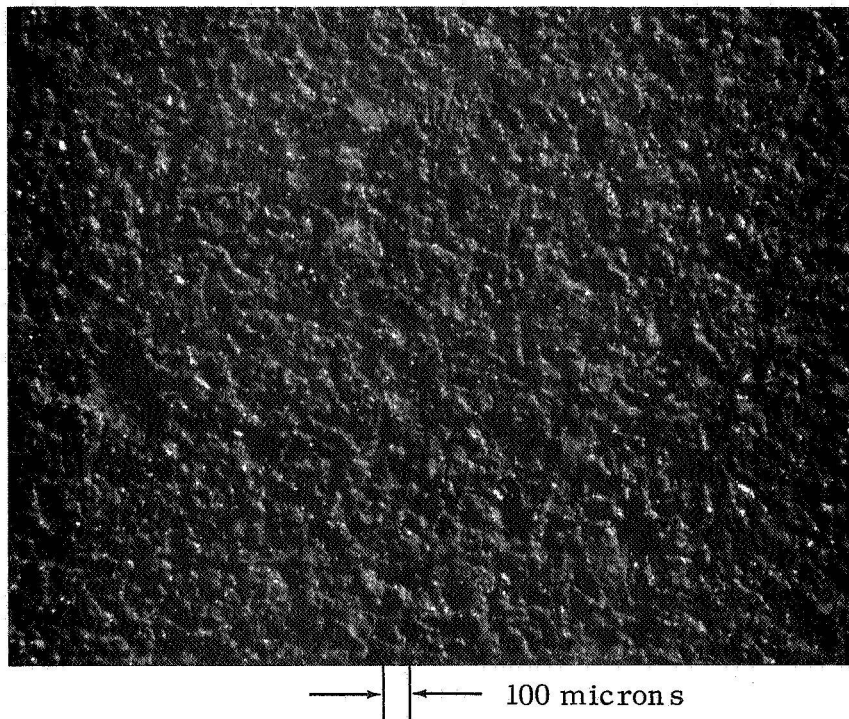


FIGURE A-4 COPPER, SANDBLASTED (SAMPLE NO. 376, SIDE A)
Average Large Scale Roughness—Dimension 48 Microns

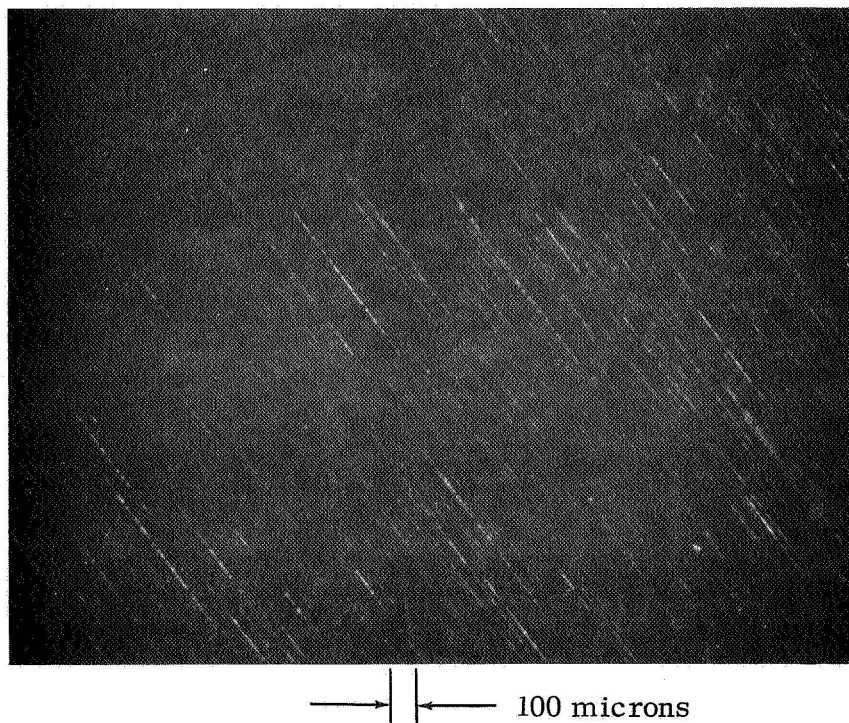
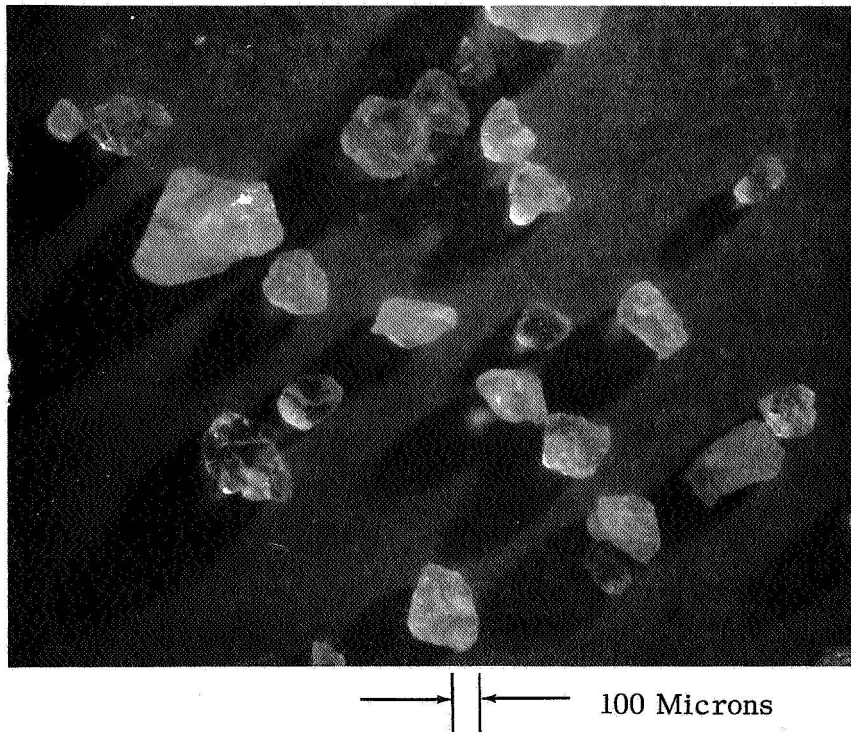


FIGURE A-5 COPPER, GROUND FINISH (SAMPLE NO. 376, SIDE B)
Average Large Scale Roughness—Dimension 2.5 Microns



**FIGURE A-6 SAND USED FOR BLASTING SAMPLES OF FIGURES A-2,
A-3, AND A-4**

Washed Fine Beach Sand, Average-Diameter 220 Microns

REFERENCES

(FOR APPENDIX A ONLY)

- A1. "Measured Directional Reflectances of Chars in the UV, Visible, and Near Infrared," Arthur D. Little, Inc., Report to Jet Propulsion Laboratory, March 1966.
- A2. "Reflectance Measurement by Three Dimensional Goniometry," D. F. Comstock, presented before the Radiometry Session Annual Meeting of the American Optical Society, Detroit, October 10-13, 1967, and also in preparation for publication under the same title.

PRECEDING PAGE BLANK NOT FILMED.

APPENDIX B

EMITTANCE MEASUREMENTS

In this appendix are summarized all of the emittance measurements performed with the Arthur D. Little, Inc., emissometer in the period 1 March, 1966, to 2 June, 1967. Many of these measurements were performed in direct support of the experimental and analytical work presented in this report and are discussed in appropriate sections of this report. A number of measurements were also made under subcontract in support of other NASA programs. Whenever possible, an attempt has been made to identify the surface materials on which these emittance measurements were made; however, in some instances the sample identification was incomplete. To complete the record, other measurements and calibrations made with the emissometer instrument are also reported.

TABLE B-1

EMISSION METER DATA SUMMARY

Test Date	Disc No.	Test No.	Sample No.	Sample Description	Envir. Code/Hrs.	Emittance
3-1-66	5	534	330B	Gold, 2072 Å, Hastings/Mylar, 1/4 mil	S95/100	0.0144
3-1-66	5	535	51	Aluminum, 862 Å, ADL/Mylar 1/4 mil	S95/100	0.020
3-2-66	5	536	45	SiO ₂ , 75 Å, ADL/Silver, 745 Å, ADL/Mylar, 1/4 mil	S95/100	0.0165
3-2-66	5	537	336	Gold, 1050 Å, Hastings/Mylar, 1/4 mil	S95/100	0.0202
3-2-66	5	538	348	MIT Sample, II-1/Mylar, 1/4 mil		0.784
3-3-66	5	539	55	SiO ₂ , 75 Å ADL/Copper, 761 Å, ADL/Mylar 1/4 mil	S95/100	0.0255
3-4-66	5	540	-	LN ₂ Calibration		---
3-4-66	5	541	-	Boiling H ₂ O Calibration		---
3-4-66	5	542	351B	Gold, 3310 Å, Hastings/Mylar, 1/4 mil.		0.0201
3-4-66	5	543	349	MIT Sample II-2		0.734
3-5-66	5	544	350	MIT Sample II-3		0.782
3-7-66	5	545	351A	Gold, 2080 Å, Hastings/Mylar, 1/4 mil		0.0145
3-10-66	5	546	353A	Gold, 2200 Å, Hastings/Mylar, 1/4 mil		0.0195
3-10-66	5	547	354A	Gold, 1740 Å, Hastings/Mylar, 1/4 mil		0.0166
3-14-66	5	548	354B	Gold, 2140 Å, Hastings/Mylar, 1/4 mil		0.0146
3-15-66	5	549	353B	Gold, 3360 Å, Hastings/Mylar, 1/4 mil		0.0173
3-15-66	5	550	352A	Gold, 2600 Å, Hastings/Mylar, 1/4 mil		0.0176
3-16-66	5	551	352B	Gold, 3340 Å, Hastings/Mylar, 1/4 mil		0.0166
3-22-66	5	552		Ag Plating on Alum, Polish side (C-99022 Sample 7, R. Sanborn)		0.0123
3-23-66	5	553		Ag Plating on Alum, Mach. side (C-99022, Sample 7, R. Sanborn)		0.0122
3-24-66	5	554		Ag on Invar (C-99022, Sample 5, R. Sanborn)		0.0153
3-29-66	5	555		Au on Invar (C-99022, Sample 5, R. Sanborn)		0.0321
3-30-66	5	556		Au on Invar (C-99022, R. Sanborn)		0.0318
3-31-66	5	557		Au on Invar (C-99022, R. Sanborn)		0.033

Table B-1 (cont.)

EMISSION METER DATA SUMMARY

Test Date	Disc No.	Test No.	Sample No.	Sample Description	Envir. Code/Hrs.	Emittance
4-1-66		559		Au on Invar (C-99022, Sample 10, R. Sanborn)		0.0324
6-8-66	5	560		Boiling water calibration		0.945
6-8-66	5	661		LN ₂ Calibration Correction -5.3°K		----
6-8-66	5	662		LN ₂ Calibration Correction -5.8°K		----
Remove sample holder & disc assembly. One .003 in. S/S support wire was very slack. Tighten S/S wire and reset disc to .040 in. gap.						
6-9-66	5	663		LN ₂ Calibration Correction -5.3°K		
6-10-66	5	664		Boiling Water Calibration		0.885
6-13-66	5	665		LN ₂ Calibration Correction -4.7°K		----
6-13-66	5	666		Boiling Water Calibration		0.942
6-14-66	5	667	35-1	Au, 940 Å, ADL/Mylar, 1/4 mil		0.0203
Remove sample holder and disc assembly. Receiver disc appears to be slightly out of parallel with the sample. Reset disc. Samples 660 through 667 done by J.C.						
6-17-66	5	668		LN ₂ Calibration Correction -7.2°K		
6-17-66	5	669		Boiling Water Calibration		0.968
6-19-66	5	670		LN ₂ Calibration Correction -6.9°K		
6-19-66	5	671		Boiling Water Calibration		0.972
6-20-66	5	672	35-1	Au, 940 Å, ADL/Mylar, 1/4 mil		0.0194
6-20-66	5	673	14-1	Au, 1335 Å, ADL/Mylar, 1/4 mil		0.0137
6-21-66	5	674	359	Nickel appearing plating w/circular wear marks (EMR C-67918)		0.0400

Table B-1 (cont.)
EMISSION DATA SUMMARY

Test Date	Disc No.	Test No.	Sample No.	Sample Description	Envir. Code/Hrs.	Emittance
6-22-66	5	676	355	Gray Coating (EMR C-67918)		0.248
6-22-66	5	677	356	Gray Coating (EMR C-67918)		0.245
6-22-66	5	678	357	Au Plating (EMR C-67918)		0.0361
6-23-66	5	679	358	Gray Coating, Split Half Sample (EMR C-47918)		0.0380
6-23-66	5	680	359	See above, Repeat of Test 674		0.0367
6-23-66	5	681	357	See Above, Repeat of Test 378		0.0362
6-24-66	5	682	361	Mg F ₂ /Aluminum--Koll Inst./Fused Quartz (2-8642)		0.0283
6-24-66	5	683	361	Same as above with Narcoil 40 Diff. Pump Oil Film Added		0.0200
6-24-66	Receiver disc on emissometer loosened from mounts--Repairs required--T.C. appear to be intact. New Disc No. 6.					
12-9-66	6	684	-	Glass beads cemented on Al disc 3/16 thick C-68647-02	--	0.990
12-9-66	6	685	-	White paint C-68050	--	0.857
12-12-66	6	686	-	White Paint C-68050	--	0.830
12-12-66	6	687	-	White Paint C-68050	--	0.834
12-13-66	6	688	-	White Paint C-68050	--	0.811
12-13-66	6	689	-	Polished Al Disc C-68050	--	0.0186
12-13-66	6	690	-	Polished Al Disc C-68050	--	0.0184
12-14-66	6	691	-	Glass Beads cemented on Al Disc (repeat test 684) C-68647-02	--	0.815
12-14-66	6	691A	353B	Au 3360 Å, Hastings/Mylar, 1/4 mil	--	0.0148
12-27-66	6	692	-	White Paint on Al Disc C-68050	--	0.825
12-27-66	6	693	-	White Paint C-68050	--	0.826
12-28-66	6	694	-	White Paint C-68050	--	0.824
12-28-66	6	695	-	White Paint C-68050	--	0.824

Table B-1 (cont.)

EMISSION METER DATA SUMMARY

Test Date	Disc. No.	Test No.	Sample No.	Sample Description	Envir. Code/Hrs.	Emittance
3-9-67	6	696	362	Au/Mylar from K sample 3077	-	0.0164
3-10-67	6	697	363	DC 704 Siloxane, 300 Å TRW/Ag 1200 Å-TRW/Mylar, 1/4 mil	-	0.011
3-13-67	6	698	364	DC 704 Siloxide, 300 Å TRW/Ag 1200 Å-TRW/Mylar, 1/4 mil	-	0.0157
3-14-67	6	699	365	DC 704 Siloxane, 300 Å TRW/Ag 1200 Å-TRW/Mylar, 1/4 mil	-	0.0114
3-14-67	6	700	366	Al, 633 Å, Hastings/Mylar, 1/4 mil	-	0.021
3-15-67	6	701	367	Al, 430 Å, Hastings/Mylar, 1/4 mil	-	0.027
3-16-67	6	702	368	Au, 783 Å, Hastings/Mylar, 1/4 mil	-	0.023
3-16-67	6	703	369	Au, 735 Å, Hastings/Mylar, 1/4 mil	-	0.024
3-17-67	6	704	370	Al, 300 Å, Hastings/Mylar, 1/4 mil	-	0.026
3-17-67	6	705	371	Al, 482 Å, Hastings/Mylar, 1/4 mil	-	0.022
3-22-67	6	706	372	Al, Sandblasted, 2.5 in. Ø x 1/8 in. thick, ADL	-	0.342
3-28-67	6	707	373	Al, 525 Å Hastings/Mylar, 1/4 mil	-	0.025
3-28-67	6	708	375	S/S 304, Sandblasted, 2.5 in. Ø x 1/8 in. thick, ADL	-	0.410
3-28-67	6	709	376	Copper, Sandblasted, 2.5 in. Ø x 1/8 in. thick, ADL	-	0.267
3-29-67	6	710	377	Copper, Polished, 2.5 in. Ø x 1.8 in. thick, ADL	-	0.0134
3-29-67	6	711	378	SS 304, Polished, 2.5 in. Ø x 1/8 in. thick, ADL	-	0.104
3-30-67	6	712	379	Al, 467 Å, Hastings/Mylar, 1/4 mil	-	0.020
3-31-67	6	713	374	Al, 682 Å, Hastings/Mylar, 1/4 mil	-	0.027

TABLE B-1 (cont.)
EMISSION METER DATA SUMMARY

Test Date	Disc No.	Test No.	Sample No.	Sample Description	Envir. Code/Hrs.	Emittance
4-18-67	6	714	401	Vaporized Al/Fiberglass Board, 1/16 in.	-	0.028
4-24-67	6	715	382	Al, 425 Å, Hastings/Mylar, 1/4 mil	-	0.028
4-26-67	6	716	314	Chromate, M.I.T./Al 2.5 in. dia. x 0.125 in. thick	-	0.049
4-26-67	6	717	362	Au, Hastings/Mylar, 1/4 mil	-	0.017
4-27-67	6	718	384	Al, 522 Å, Hastings/Mylar, 1/4 mil	-	0.023
4-28-67	6	719	386	Al, 469 Å, Hastings/Mylar, 1/4 mil	-	0.029
5-2-67	6	720	387	Al, 595 Å, Hastings/Mylar, 1/4 mil	-	0.024
5-23-67	6	721	388	Al, 550 Å, Hastings/Mylar, 1/4 mil	-	0.031
5-25-67	6	722	389	Al, 505 Å, Hastings/Mylar, 1/4 mil	-	0.026
5-25-67	6	723	392	Al, 471 Å, Hastings/Mylar, 1/4 mil	-	0.031
5-25-67	6	724	395	Al, 522 Å, Hastings/Mylar, 1/4 mil	-	0.021
5-27-67	6	725	396	Al, 478 Å, Hastings/Mylar, 1/4 mil	-	0.028
5-27-67	6	726	399	Al, 570 Å, Hastings/Mylar, 1/4 mil	-	0.021
5-31-67	6	727	400	Al, 513 Å, Hastings/Mylar, 1/4 mil	-	0.034
5-31-67	6	728	404	Al, 637 Å, Hastings/Mylar, 1/4 mil	-	0.023
6-1-67	6	729	405	Al, 455 Å, Hastings/Mylar, 1/4 mil	-	0.021
6-1-67	6	730	408	Al, 413 Å, Hastings/Mylar, 1/4 mil	-	0.024
6-1-67	6	731	409	Al, 417 Å, Hastings/Mylar, 1/4 mil	-	0.034
6-2-67	6	732	412	Al, 527 Å, Hastings/Mylar, 1/4 mil	-	0.022
6-2-67	6	733	413	Al, 390 Å, Hastings/Mylar, 1/4 mil	-	0.033
6-2-67	6	734	416	Al, 515 Å, Hastings/Mylar, 1/4 mil	-	0.024
6-2-67	6	735	417	Al, 452 Å, Hastings/Mylar, 1/4 mil	-	0.030

REPORT DISTRIBUTION LIST FOR
CONTRACT NO. NAS3-7974

NASA Lewis Research Center (1)
21000 Brookpark Road
Cleveland, Ohio 44135
Attention: Contracting Officer, MS 500-210

NASA Lewis Research Center (8)
21000 Brookpark Road
Cleveland, Ohio 44135
Attention: Liquid Rocket Technology Branch,
MS 500-209

NASA Lewis Research Center (1)
21000 Brookpark Road
Cleveland, Ohio 44135
Attention: Technical Report Control Office,
MS 5-5

NASA Lewis Research Center (1)
21000 Brookpark Road
Cleveland, Ohio 44135
Attention: Technology Utilization Office,
MS 3-16

NASA Lewis Research Center (2)
21000 Brookpark Road
Cleveland, Ohio 44135
Attention: AFSC Liaison Office, MS 4-1

NASA Lewis Research Center (2)
21000 Brookpark Road
Cleveland, Ohio 44135
Attention: Library

NASA Lewis Research Center (1)
21000 Brookpark Road
Cleveland, Ohio 44135
Attention: D. L. Nored, MS 500-209

NASA Lewis Research Center (1)
21000 Brookpark Road
Cleveland, Ohio 44135
Attention: Office of Reliability & Quality
Assurance, MS 500-203

NASA Lewis Research Center (1)
21000 Brookpark Road
Cleveland, Ohio 44135
Attention: E.W. Conrad, MS 100-1

NASA Lewis Research Center (1)
21000 Brookpark Road
Cleveland, Ohio 44135
Attention: W.E. Roberts, MS 3-17

NASA Lewis Research Center (1)
21000 Brookpark Road
Cleveland, Ohio 44135
Attention: R. Knoll, MS 501-2

National Aeronautics and Space Administration (1)
Washington, D. C. 20546
Attention: Code MT

National Aeronautics and Space Administration (1)
Washington, D. C. 20546
Attention: Code RPX

National Aeronautics and Space Administration (1)
Washington, D. C. 20546
Attention: Code RPL

National Aeronautics and Space Administration (1)
Washington, D. C. 20546
Attention: Code SV

National Aeronautics and Space Administration (2)
Washington, D. C. 20546
Attention: Code RV-2

NASA Scientific and Technical Information
Facility (6)
Box 33
College Park, Md. 20740
Attention: NASA Representative, Code CRT

NASA Ames Research Center (1)
Moffett Field, Calif. 94035
Attention: Library

NASA Ames Research Center (1)
Moffett Field, Calif. 94035
Attention: E.R. Streed, SED

NASA Flight Research Center (1)
Box 273
Edwards, Calif. 93523
Attention: Library

NASA Goddard Space Flight Center (1)
Greenbelt, Md. 20771
Attention: Library

NASA Goddard Space Flight Center (1)
Greenbelt, Md. 20771
Attention: W.C. Lund, Code 623

NASA Kennedy Space Center (1)
Cocoa Beach, Fla. 32931
Attention: Library

NASA Langley Research Center (1)
Langley Station
Hampton, Va. 23365
Attention: Library

NASA Langley Research Center (1)
Langley Station
Hampton, Va. 23365
Attention: R.R. Heldenfels

NASA Manned Spacecraft Center (1)
Houston, Tex. 77001
Attention: Library

NASA Manned Spacecraft Center (1)
Houston, Tex. 77001
Attention: Merlyn Lausten (EP-2)

NASA Marshall Space Flight Center (1)
Huntsville, Ala. 35812
Attention: Library

NASA Marshall Space Flight Center (1)
Huntsville, Ala. 35812
Attention: Keith Chandler, R-P&VE -PA

NASA Marshall Space Flight Center (1)
Huntsville, Ala. 35812
Attention: Clyde Nevins

NASA Marshall Space Flight Center (1)
Huntsville, Ala. 35812
Attention: E.H. Hyde

NASA Marshall Space Flight Center (1)
Huntsville, Ala. 35812
Attention: I.G. Yates

Jet Propulsion Laboratory (1)
4800 Oak Grove Drive
Pasadena, Calif. 91103
Attention: Library

Office of the Director of Defense
Research & Engineering (1)
Washington, D. C. 20301
Attention: Dr. H.W. Schulz, Office of
Asst. Dir. (Chem. Technology)

Defense Documentation Center (1)
Cameron Station
Alexandria, Va. 22314

RTD (RTNP) (1)
Bolling Air Force Base
Washington, D. C. 20332

Arnold Engineering Development Center (1)
Air Force Systems Command
Tullahoma, Tenn. 37389
Attention: AEOIM

Advanced Research Projects Agency (1)
Washington, D. C. 20525
Attention: D.E. Mock

Aeronautical Systems Division (1)
Air Force Systems Command
Wright-Patterson Air Force Base, Ohio
Attention: D.L. Schmidt, Code ASRCNC -2

Air Force Missile Test Center (1)
Patrick Air Force Base, Fla.
Attention: L.J. Ullian

Air Force Systems Command (1)
Andrews Air Force Base
Washington, D. C. 20332
Attention: SCLT/Capt. S.W. Bowen

Air Force FTC (FTAT-2) (1)
Edwards Air Force Base, Calif. 93523
Attention: Col. J.M. Silk

Air Force Office of Scientific Research (1)
Washington, D. C. 20333
Attention: SREP, Dr. J.F. Masi

U.S. Air Force (1)
Washington, D. C.
Attention: Col. C.K. Stambaugh, Code AFRST

Bureau of Naval Weapons (1)
Department of the Navy
Washington, D.C.
Attention: J. Kay, Code RTMS-41

Commander (1)
U.S. Naval Missile Center
Point Mugu, Calif. 93041
Attention: Technical Library

Commanding Officer (1)
Office of Naval Research
1030 E. Green Street
Pasadena, Calif. 91101

Director (Code 6180) (1)
U.S. Naval Research Laboratory
Washington, D.C. 20390
Attention: H.W. Carhart

Picatinny Arsenal (1)
Dover, New Jersey
Attention: I. Forsten, Chief
Liquid Propulsion Laboratory

Aerojet-General Corporation (1)
Box 296
Azusa, Calif. 91703
Attention: Librarian

Aerojet-General Corporation (1)
Box 1947
Sacramento, Calif. 95809
Attention: Technical Library 2484-2015A

Aeronutronic Division of (1)
Philco Corporation
Ford Road
Newport Beach, Calif. 92600
Attention: Library

Aeroprojects, Incorporated (1)
310 East Rosedale Avenue
West Chester, Pa. 19380
Attention: C.D. McKinney

Aerospace Corporation (1)
Box 95085
Los Angeles, Calif. 90045
Attention: J.G. Wilder, MS-2293

Aerospace Corporation (1)
Box 95085
Los Angeles, Calif. 90045
Attention: Library-Documents

Air Products and Chemicals, Inc. (1)
Allentown, Pa.
Attention: A. Lopin

Astrosystems, Incorporated (1)
1275 Bloomfield Avenue
Caldwell Township, N.J.
Attention: A. Mendenhall

Atlantic Research Corporation (1)
Shirley Highway & Edsall Road
Alexandria, Va. 22314
Attention: Security Office for Library

Battelle Memorial Institute (1)
505 King Avenue
Columbus, Ohio 43201
Attention: Report Library, Room 6A

Beech Aircraft Corporation Boulder Facility Box 631 Boulder, Colo. 80302 Attention: J.H. Rodgers	(1)	Curtiss-Wright Corporation Wright Aeronautical Division Woodridge, N.J. Attention: G. Kelley	(1)
Beech Aircraft Corporation Boulder Facility Box 631 Boulder, Colo. 80302 Attention: R.L. Reed	(1)	University of Denver Denver Research Institute Box 10127 Denver, Colo. 80210 Attention: Security Office	(1)
Bell Aerosystems, Inc. Box 1 Buffalo, N.Y. 14205 Attention: T. Reinhardt	(1)	McDonnell Douglas Corporation Santa Monica Division 3000 Ocean Park Blvd. Santa Monica, Calif. 90405 Attention: J.W. Price	(1)
Bell Aerosystems, Inc. Box 1 Buffalo, N.Y. 14205 Attention: W.M. Smith	(1)	General Dynamics/Astronautics Box 1128 San Diego, Calif. 92112 Attention: Library & Information Services (128-00)	(1)
Autonetics 3370 Miralona Avenue Anaheim, Calif. 92803 Attention: Dr. Edward Lax Dept. 447, Bldg. 202	(1)	Convair Division General Dynamics Corporation Box 1128 San Diego, Calif. 92112 Attention: Karl Leonhard	(1)
The Boeing Company Aero Space Division Box 3707 Seattle, Wash. 98124 Attention: Library	(1)	Convair Division General Dynamics Corporation Box 1128 San Diego, Calif. 92112 Attention: Paul Stevens	(1)
The Boeing Company Aero Space Division Box 3707 Seattle, Wash. 98124 Attention: C.F. Tiffany	(1)	General Electric Company Flight Propulsion Lab. Department Cincinnati 15, Ohio Attention: D. Suichu	(1)
Chemical Propulsion Information Agency Applied Physics Laboratory 8621 Georgia Avenue Silver Spring, Md. 20910	(1)	Grumman Aircraft Engineering Corporation Bethpage, Long Island, N.Y. Attention: Joseph Gavin	(1)

The Garrett Corporation (1)
1625 Eye Street, N.W.
Washington, D. C.
Attention: G.R. Shepard

IIT Research Institute (1)
Technology Center
Chicago, Ill. 60616
Attention: Technical Library

Goodyear Aerospace Corporation (1)
1210 Massillon Road
Akron, Ohio
Attention: Clem Shriver, Dept. 481

Lockheed Missiles & Space Company (1)
Box 504
Sunnyvale, Calif.
Attention: James Guill

Lockheed Missiles & Space Company (1)
Box 504
Sunnyvale, Calif.
Attention: Library

Marquardt Corporation (1)
16555 Saticoy Street
Box 2013 - South Annex
Van Nuys, Calif. 91404
Attention: Librarian

Marquardt Corporation (1)
16555 Saticoy Street
Box 2013 - South Annex
Van Nuys, Calif. 91404
Attention: W.D. Boardman, Jr.

Martin-Marietta Corporation (1)
Martin Division
Baltimore 3, Md.
Attention: Science-Technology Library

Martin-Marietta Corporation (1)
Martin Division
Baltimore 3, Md.
Attention: R. Crawford

McDonnell Douglas Corporation (1)
Box 6101
Lambert Field, Mo.
Attention: R.A. Herzmark

North American Aviation, Inc. (1)
Space & Information Systems Division
12214 Lakewood Boulevard
Downey, Calif. 90242
Attention: Technical Information Center,
D/096-722 (AJ01)

North American Aviation, Inc. (1)
Space & Information Systems Division
12214 Lakewood Boulevard
Downey, Calif. 90242
Attention: H. Storms

Northrop Space Laboratories (1)
1001 East Broadway
Hawthorne, Calif.
Attention: Dr. William Howard

Purdue University (1)
Lafayette, Ind. 47907
Attention: Technical Librarian

Republic Aviation Corporation (1)
Farmingdale, Long Island, N.Y.
Attention: Dr. William O'Donnell

Rocketdyne Division of
North American Rockwell, Inc. (1)
6633 Canoga Avenue
Canoga Park, Calif. 91304
Attention: Library, Department 596-306

Space-General Corporation (1)
777 Flower Street
Glendale, Calif.
Attention: C.E. Roth

Stanford Research Institute (1)
333 Ravenswood Avenue
Menlo Park, Calif. 94025
Attention: P.R. Gillette

Thiokol Chemical Corporation (1) Reaction Motors Division Denville, N.J. 07834 Attention: A. Sherman	Vought Astronautics (1) Box 5907 Dallas 22, Tex. Attention: Warren C. Trent
Thiokol Chemical Corporation (1) Reaction Motors Division Denville, N.J. 07834 Attention: Librarian	Cryonetics Corporation (1) Northwest Industrial Park Burlington, Mass. Attention: James F. Howlett
TRW Systems, Incorporated (1) 1 Space Park Redondo Beach, Calif. 90200 Attention: G.W. Elverum	Martin-Marietta Corporation (1) Denver Division Denver, Colo. Attention: Library
TRW Systems, Incorporated (1) 1 Space Park Redondo Beach, Calif. 90200 Attention: STL Tech. Lib. Doc. Acquisitions	Martin-Marietta Corporation (1) Denver Division Denver, Colo. Attention: D.W. Murphy
Union Carbide Corporation (1) Linde Division Box 44 Tonawanda, N.Y. 14152 Attention: C.R. Lindquist	National Research Corporation (1) 70 Memorial Drive Cambridge, Mass. 02142
United Aircraft Corporation (1) Corporation Library 400 Main Street East Hartford, Conn. 06118 Attention: Dr. David Rix	New York University (1) University Heights New York, N.Y. Attention: P.F. Winternitz
United Aircraft Corporation (1) Corporation Library 400 Main Street East Hartford, Conn. 06118 Attention: Erle Martin	
United Aircraft Corporation (1) United Technology Center Box 358 Sunnyvale, Calif. 94088 Attention: Librarian	