ANL-80-97

ARGONNE NATIONAL LABORATORY
9700 South Cass Avenue
Argonne, Illinois 60439

THERMAL-PERFORMANCE STUDY OF
LIQUID METAL FAST BREEDER REACTOR INSULATION

by

Kelvin K. Shiu

September 1980

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, expressed or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product or process is by trade name, trademark, manufacturer, or otherwise, but is not to be taken as an endorsement or recommendation in favor of the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

Based on a thesis submitted to the
Graduate School of the University of Illinois--Urbana
in partial fulfillment of
requirements for the degree of
Doctor of Philosophy in Nuclear Engineering

TABLE OF CONTENTS

	<u>Page</u>
ABSTRACT.....	13
I. INTRODUCTION.....	13
II. LITERATURE SURVEY.....	16
A. Design Approaches, Proposed Concepts, and Factors Affecting Insulation Performance in LMFBR Systems.....	16
B. Types of Insulation.....	26
C. Contact Resistance.....	29
D. Convection.....	31
E. Radiation and Emissivity.....	32
F. Gas Conduction.....	34
G. Sodium and Sodium Oxides.....	35
III. ANALYTICAL STUDIES.....	38
A. Thermal Convection.....	38
1. Natural Convection.....	38
2. Natural Convection within a Porous Medium.....	43
B. Thermal Radiation.....	47
1. Integral-equation Approach.....	48
2. Finite-difference Approximation.....	49
3. Network Method.....	51
4. Gebhart Method.....	54
5. Hottel's Method.....	56
6. Modified Gebhart Method.....	58
C. Contact Resistance.....	61
D. Comprehensive Models.....	63
1. Multiplates.....	63
2. Honeycombs.....	63
3. Screens.....	64
E. Influence of Sodium and Sodium Oxide on Models.....	65
1. Sodium.....	65
2. Sodium Oxide.....	69

TABLE OF CONTENTS

	<u>Page</u>
IV. EXPERIMENTAL SYSTEMS.....	70
A. Design and Setup.....	70
B. Test Samples.....	77
C. Experimental Procedures.....	79
1. Evacuated and Nonevacuated Environment.....	79
2. Sodium and Sodium Oxide Environment.....	82
D. Infrared Measurements and System.....	88
E. Energy Balance.....	91
V. DISCUSSION OF EXPERIMENTAL RESULTS.....	95
A. Multilayer Stainless Steel Plates.....	95
1. Twelve Plates.....	95
2. Twenty-four Plates.....	98
B. Honeycombs.....	100
C. Screen-type Insulation.....	103
1. With 12 Reflective Plates.....	103
2. With 24 Reflective Plates.....	105
3. Sodium Immersion and Oxidation.....	107
VI. COMPARISONS BETWEEN DATA AND THEORY.....	115
A. Discussion of Reduced Experimental Measurements.....	115
1. Honeycombs.....	115
2. Twelve-plate Results.....	116
3. Twenty-four-plate Results.....	117
4. Twelve-screen Results.....	118
5. Twenty-four-screen Results.....	120
6. Twenty-four Screens with Sodium and with Oxide.....	121
B. Comparisons between Types of Insulation and Published Data...	123
1. Comparison between Various Types of Insulation.....	123
2. Comparison with Other Results.....	125
VII. CONCLUSIONS AND RECOMMENDATIONS.....	127

TABLE OF CONTENTS

	<u>Page</u>
APPENDIXES	
A. Uniqueness of the Gebhart and Hottel Solutions.....	129
B. Numerical Application of the Gebhart Method and the Modified Gebhart Method.....	131
C. Error Analysis of Temperature Measurements.....	135
D. Recorder Calibrations.....	139
E. Network-model Derivations.....	140
F. Computer Listing.....	143
ACKNOWLEDGMENTS.....	147
REFERENCES.....	147

LIST OF FIGURES

<u>No.</u>	<u>Title</u>	<u>Page</u>
1.	Cross Section of Fermi Reactor.....	18
2.	Elevation View of EBR-II.....	19
3.	Elevation View of Phenix Reactor.....	19
4.	Elevation View of Prototype Fast Reactor.....	20
5.	Elevation View of SuperPhenix Reactor.....	20
6.	Elevation View of Commercial Fast Reactor.....	21
7.	Elevation View of BN-600.....	22
8.	Elevation View of Fast Flux Test Facility.....	22
9.	Elevation View of SNR-300.....	23
10.	Elevation View of Monju.....	23
11.	Vapor Pressure of Liquid Sodium below 1250 K.....	25
12.	Sodium Aerosol Concentration.....	26
13.	Multilayer Plates after Exposure to Sodium.....	27
14.	Schematic Diagram of Honeycomb Insulation.....	28
15.	Nusselt Number vs Rayleigh Number for Parallel Plates.....	31
16.	Wetting Time vs Sodium Temperature.....	37
17.	Flow Pattern for Bernard Cell.....	39
18.	Rayleigh Number vs $\Delta T L^3$ for Parallel Plates.....	39
19.	Ni vs Wave Number a	42
20.	Nusselt Number vs Rayleigh Number for Stainless Steel Cylinder....	43
21.	Rayleigh Number vs Permeability of the Medium.....	45
22.	Configuration-shape-factor Notation.....	48
23.	Enclosure with Continuous Variation of Radiosity, Surface Temperature, and Surface Heat Flux.....	48
24.	Equivalent Networks for Energy Exchange between Surfaces.....	51
25.	Enclosure to Illustrate the Gebhart Method.....	54
26.	A Cubical Enclosure.....	58
27.	Contact Resistance vs Load Pressure for Soldered Joint.....	61
28.	Thermal Resistance per Contact vs Load.....	62
29.	Thermal Conductivity of Sodium vs Temperature.....	65

LIST OF FIGURES

<u>No.</u>	<u>Title</u>	<u>Page</u>
30.	Definition of Wetting Angles.....	66
31.	Wetting Angles of Different Materials vs Temperature.....	66
32.	Wetting Angles of Stainless Steel with Different Surface Finish vs Temperature.....	67
33.	Effects of Oxidation on Wetting Angles of Stainless Steel vs Temperature.....	67
34.	Effects of Preoxidized Stainless Steel on Wetting Angles vs Time..	67
35.	Effects of Oxygen Content in Sodium on Wetting Angles at Different Temperatures.....	68
36.	Design Drawing of Test Vessel.....	71
37.	Locations of Thermocouples Exterior to Test Vessel.....	73
38.	Arrangement of Thermocouples on Test Sample.....	74
39.	Electrical Circuit of Power Supply.....	75
40.	Test Vessel within 55-gal Drum Filled with Vermiculite.....	76
41.	Twenty-four Multiplates with Supports.....	77
42.	Screen Composites.....	78
43.	Schematic Diagram of Test Facility.....	80
44.	Upper Section of Test Vessel.....	80
45.	Schematic Diagram of Thermocouple Connections.....	81
46.	Mobile Sodium Pig.....	83
47.	Schematic Diagram of Mobile Sodium Pig.....	84
48.	Oxygen Content in Sodium vs Temperature.....	84
49.	Relation of Sodium Viscosity and Temperature.....	85
50.	Transfer of Sodium-wetted Test Sample.....	87
51.	Schematic Diagram of Infrared System.....	88
52.	Experimental Setup of Infrared Measurement.....	89
53.	Emissivity of Stainless Steel with 2B Cold-rolled Finish at Different Temperatures.....	90
54.	Emissivity of Nickel-based Brazed Alloy vs Temperature.....	90
55.	Emissivity vs Time at 427°C for 2B Finish Stainless Steel.....	91
56.	Emissivity vs Temperature after 45 Hours at 427°C.....	91

LIST OF FIGURES

<u>No.</u>	<u>Title</u>	<u>Page</u>
57.	Global-energy Balance.....	92
58.	Thermal Conductance of Wet Pak vs Temperature.....	93
59.	Energy Balance within Insulation Test Sample.....	94
60.	Temperature Distribution of 12 Multiplates, Nonevacuated, with Hot-face Temperature at 340°F (171°C).....	95
61.	Temperature Distribution of Test-vessel Wall Concurrent to Fig. 60 Measurements.....	95
62.	Temperature Distribution of 12 Multiplates, Nonevacuated, with Hot-face Temperature at 615°F (324°C).....	96
63.	Temperature Distribution of 12 Multiplates, Evacuated, with Hot-face Temperature at 390°F (199°C).....	97
64.	Temperature Distribution of 12 Multiplates, Evacuated, with Hot-face Temperature at 548°F (287°C).....	97
65.	Temperature Distribution of 12 Multiplates, Evacuated, with Hot-face Temperature at 690°F (366°C).....	97
66.	Temperature Distribution of Test-vessel Wall Concurrent to Fig. 63 Measurements.....	97
67.	Temperature Distribution of Test-vessel Wall Concurrent to Fig. 64 Measurements.....	98
68.	Temperature Distribution of 24 Multiplates, Nonevacuated, with Hot-face Temperature at 442°F (228°C).....	98
69.	Temperature Distribution of 24 Multiplates, Nonevacuated, with Hot-face Temperature at 543°F (284°C).....	98
70.	Temperature Distribution of 24 Multiplates, Evacuated, with Hot-face Temperature at 350°F (177°C).....	99
71.	Temperature Distribution of 24 Multiplates, Evacuated, with Hot-face Temperature at 440°F (227°C).....	99
72.	Temperature Distribution of 24 Multiplates, Evacuated, with Hot-face Temperature at 550°F (288°C).....	99
73.	Temperature Distribution of Honeycomb, Nonevacuated, with Hot-face Temperature at 310°F (154°C).....	100
74.	Temperature Distribution of Honeycomb, Nonevacuated, with Hot-face Temperature at 480°F (249°C).....	100
75.	Temperature Distribution of Honeycomb, Evacuated, with Hot-face Temperature at 320°F (160°F).....	101

LIST OF FIGURES

<u>No.</u>	<u>Title</u>	<u>Page</u>
76.	Temperature Distribution of Honeycomb, Evacuated, with Hot-face Temperature at 430°F (221°C).....	101
77.	Temperature Distribution of Honeycomb, Evacuated, with Hot-face Temperature at 670°F (354°C).....	102
78.	Temperature Distribution of Honeycomb, Evacuated, with Hot-face Temperature at 695°F (368°C).....	102
79.	Temperature Distribution of 12 Screen Plates, Nonevacuated, with Hot-face Temperature at 300°F (149°C).....	103
80.	Temperature Distribution of 12 Screen Plates, Nonevacuated, with Hot-face Temperature at 365°F (185°C).....	103
81.	Temperature Distribution of 12 Screen Plates, Nonevacuated, with Hot-face Temperature at 470°F (243°C).....	103
82.	Temperature Distribution of 12 Screen Plates, Evacuated, with Hot-face Temperature at 350°F (177°C).....	104
83.	Temperature Distribution of 12 Screen Plates, Evacuated, with Hot-face Temperature at 490°F (254°C).....	104
84.	Temperature Distribution of 12 Screen Plates, Evacuated, with Hot-face Temperature at 620°F (327°C).....	104
85.	Temperature Distribution of 12 Screen Plates, Evacuated, with Hot-face Temperature at 580°F (304°C).....	104
86.	Temperature Distribution of 24 Screen Plates, Nonevacuated, with Hot-face Temperature at 407°F (208°C).....	105
87.	Temperature Distribution of 24 Screen Plates, Nonevacuated, with Hot-face Temperature at 475°F (246°C).....	105
88.	Temperature Distribution of 24 Screen Plates, Nonevacuated, with Hot-face Temperature at 556°F (291°C).....	106
89.	Temperature Distribution of 24 Screen Plates, Evacuated, with Hot-face Temperature at 345°F (174°C).....	106
90.	Temperature Distribution of 24 Screen Plates, Evacuated, with Hot-face Temperature at 480°F (249°C).....	106
91.	Temperature Distribution of 24 Screen Plates, Evacuated, with Hot-face Temperature at 596F (313°C).....	106
92.	Temperature Distribution of 24 Screen Plates, Evacuated, with Hot-face Temperature at 650°F (343°C).....	106

LIST OF FIGURES

<u>No.</u>	<u>Title</u>	<u>Page</u>
93.	Temperature Distribution of 24 Screen Plates with Sodium as a Heat Source.....	107
94.	Temperature Distribution of 24 Screen Plates Wetted with 5 in. (127 mm) of Sodium, Hot-face Temperature at 390°F (199°C).....	108
95.	Temperature Distribution of 24 Screen Plates Wetted with 5 in. (127 mm) of Sodium, Hot-face Temperature at 455°F (235°C).....	108
96.	Temperature Distribution of 24 Screen Plates Wetted with 5 in. (127 mm) of Sodium, Hot-face Temperature at 840°F (449°C).....	108
97.	Temperature Distribution of 24 Screen Plates Wetted with 5 in. (127 mm) of Sodium, Hot-face Temperature at 845°F (452°C).....	109
98.	Temperature Distribution of 24 Screen Plates Wetted with 5 in. (127 mm) of Sodium, Oxidized, Hot-face Temperature at 698°F (370°C).....	109
99.	Temperature Distribution of 24 Screen Plates Wetted with 5 in. (127 mm) of Sodium, Oxidized, Hot-face Temperature at 560°F (293°C).....	110
100.	Temperature Distribution of 24 Screen Plates Wetted with 5 in. (127 mm) of Sodium, Oxidized, Hot-face Temperature at 430°F (221°C).....	110
101.	Temperature Distribution of 24 Screen Plates Wetted with 8 in. (203 mm) of Sodium, Hot-face Temperature at 857°F (458°C).....	111
102.	Temperature Distribution of 24 Screen Plates Wetted with 8 in. (203 mm) of Sodium, Hot-face Temperature at 690°F (366°C).....	111
103.	Temperature Distribution of 24 Screen Plates Wetted with 8 in. (203 mm) of Sodium, Hot-face Temperature at 520°F (271°C).....	112
104.	Temperature Distribution of 24 Screen Plates Wetted with 8 in. (203 mm) of Sodium, Oxidized, Hot-face Temperature at 448°F (231°C).....	112
105.	Temperature Distribution of 24 Screen Plates Wetted with 8 in. (203 mm) of Sodium, Oxidized, Hot-face Temperature at 615°F (324°C).....	112
106.	Temperature Distribution of 24 Screen Plates Wetted with 8 in. (203 mm) of Sodium, Oxidized, Hot-face Temperature at 820°F (438°C).....	113
107.	Sodium Oxide Deposits on Plate.....	113
108.	Example of Sodium Oxide Deposits on First Few Screens.....	114

LIST OF FIGURES

<u>No.</u>	<u>Title</u>	<u>Page</u>
109.	Example of Sodium Oxide Deposits on Last Few Screens.....	114
110.	Normalized Q/DT vs Temperature for Honeycomb.....	115
111.	Effective Thermal Conductivity vs Temperature for Honeycomb.....	116
112.	Normalized Q/DT vs Temperature for 12 Multiplates.....	116
113.	Effective Thermal Conductivity vs Temperature for 12 Multiplates.	117
114.	Normalized Q/DT vs Temperature for 24 Multiplates.....	118
115.	Effective Thermal Conductivity vs Temperature for 24 Multiplates.	118
116.	Normalized Q/DT vs Temperature for 12 Screen Plates.....	119
117.	Effective Thermal Conductivity vs Temperature for 12 Screen Plates.....	119
118.	Normalized Q/DT vs Temperature for 24 Screen Plates.....	120
119.	Effective Thermal Conductivity vs Temperature for 24 Screen Plates.....	120
120.	Effective Thermal Conductivity vs Temperature for 24 Screen Plates with Sodium and Sodium Oxide.....	122
121.	Effective Thermal Conductivity vs Temperature for Screen Plates, Composite Insulation by Lemerrier et al.	125
122.	Comparison of Effective Thermal Conductivity of Honeycomb by Rohr and by This Study.....	126
C.1.	Insulated Thermocouple Attached to Massive Solid.....	135
C.2.	Insulated Thermocouple Attached to Thin Member.....	137
C.3.	Temperature Errors vs Thermocouple Conductance.....	138
E.1.	Derivation of $\epsilon + \tau = 1$	140
E.2.	Derivation of $\epsilon + \rho + \tau = 1$	140
E.3.	Total Thermal Resistance between Two Plates Separated by a Non- reflective and Nonopaque Medium.....	141
E.4.	Total Thermal Resistance between a Black Surface and Its Adjacent Nonreflective and Nonopaque Medium.....	141
E.5.	Total Thermal Resistance between Two Plates Separated by a Re- flective, Transparent, and Emissive Medium.....	142

LIST OF TABLES

<u>No.</u>	<u>Title</u>	<u>Page</u>
I.	Summary of French Screen-type Insulation.....	28
II.	Values of N_i as a Function of Wave Number a , from Catton (1966).	43
III.	Fabrication and Inspection Requirements for Test Vessel.....	72

THERMAL-PERFORMANCE STUDY OF
LIQUID METAL FAST BREEDER REACTOR INSULATION

by

Kelvin K. Shiu

ABSTRACT

Three types of metallic thermal insulation were investigated analytically and experimentally: multilayer reflective plates, multilayer honeycomb composite, and multilayer screens. Each type was subjected to evacuated and nonevacuated conditions, where thermal measurements were made to determine thermal-physical characteristics. A variation of the separation distance between adjacent reflective plates of multilayer reflective plates and multilayer screen insulation was also experimentally studied to reveal its significance. One configuration of the multilayer screen insulation was further selected to be examined in sodium and sodium oxide environments. The emissivity of Type 304 stainless steel used in comprising the insulation was measured by employing infrared technology.

A comprehensive model was developed to describe the different proposed types of thermal insulation. Various modes of heat transfer inherent in each type of insulation were addressed and their relative importance compared. Provision was also made in the model to allow accurate simulation of possible sodium and sodium oxide contamination of the insulation. The thermal-radiation contribution to heat transfer in the temperature range of interest for LMFBR's was found to be moderate, and the suppression of natural convection within the insulation was vital in preserving its insulating properties. Experimental data were compared with the model and other published results. Moreover, the three proposed test samples were assessed and compared under various conditions as viable LMFBR thermal insulations.

I. INTRODUCTION

Motivated by the concepts of producing more fissionable materials than it consumes and of better heat-transfer capabilities from fuel to coolant, the Liquid Metal Fast Breeder Reactor (LMFBR) was conceived in the middle 1940's. Because of the inherent characteristic that a nuclear reactor's power output is primarily limited by the ability to transfer heat away from the fuel, liquid metals with their superb heat-transfer characteristics were suggested for use

as reactor coolants. Both sodium and a mixture of sodium and potassium (NaK) were two liquid metals that received a great deal of attention in the early development of the LMFBR. However, when more became known about these liquid metals, particularly their fire-hazard potential and NaK's corrosive properties, NaK was rendered less attractive than sodium as a reactor coolant, and sodium is being used almost exclusively in today's LMFBR systems. Another major attraction of sodium in an LMFBR is its low neutron-absorption cross section at high energies. This enhances the reactor's ability to produce more fuel than it consumes if fertile fuel blanket assemblies are placed around the reactor core to capture this breeding capability.

Various design approaches of the primary system and the reactor vessel were proposed. Presently, all LMFBR's can be categorized as either pool or loop types. The pool-type design can be characterized by a large reactor vessel within which is contained the primary heat-transport system, which comprises both the primary pumps and the intermediate heat exchangers. These components are immersed in a pool of sodium covering the reactor core, and thus, the name pool-type LMFBR. For a loop-type reactor, the components are located outside the reactor vessel and are interconnected by piping forming "loops" to allow sodium to flow to and from each component. Because of this major difference in component locations, the loop reactor requires a smaller-diameter reactor vessel. For a small power reactor, e.g., Clinch River Breeder Reactor (CRBR) (300 MWt), the closure of the reactor vessel comprises only rotating plugs, whereas, for a similar power-size pool reactor, the closure consists of the rotating plugs as well as a shield deck.

In the recent designs of a commercial-size plant (1200 MWe), the diameter of the pool-type LMFBR reactor vessel is 60-80 ft (18.3-24.4 m); its loop-type counterpart is about 40 ft (12.2 m). Moreover, the closure head of a loop-type reactor vessel is proposed to consist of the rotating plugs and a shield deck. There are indications that the use of a large reactor vessel might mitigate effects of events such as core-disruptive accidents. Besides the safety features, the large-diameter vessel is also favored because, with the large sodium inventory inside, it provides a means to reduce thermal transient effects. There is increasing evidence that the large reactor vessel will be used for loop-type LMFBR's. Both General Electric and Westinghouse, in their loop Prototype Large Breeder Reactor (PLBR) conceptual designs, proposed 40-ft (12.2-m)-dia reactor vessels.

Various deck designs were proposed, and some of them were built. Experimental Breeder Reactor II (EBR-II) and Phenix use the integral-deck approach, where the closure deck and the vessel wall form an integral member, subjecting the closure to elevated temperature. Thermal insulation is placed on the outside of the closure to prevent heat losses. The advantage of an integral deck stems from its ability to form a fairly tight boundary to contain sodium aerosols transport. The requirement imposed on the thermal insulation are also less stringent. However, operating experience indicates certain unattractive features of this approach, one of which is that thermal-transient effects

resulted in significant stresses on the integral member. This has prompted the French reactor designers to explore an alternative approach, namely, the cold-deck design, in which the shield deck is maintained at only a slightly elevated temperature [$\sim 150^{\circ}\text{F}$ (66°C)]. Instead of a complete steel structure, a web approach filled with high-density concrete is suggested. The load of the deck is taken up by the web structure, and the concrete is used only for gamma-shielding purposes. To maintain the required temperature level on the deck structure, thermal insulation is used on the underside of the deck, where it is exposed to sodium aerosols. Furthermore, a cooling system is also implemented within the deck structure to ensure desired temperature levels under all operating conditions. As noted earlier, the closure of small loop reactors comprises only the rotating plugs, and because of the ability of these plugs to allow free expansion, thermal-insulation requirements become less stringent. However, a situation similar to the pool reactor is encountered when a large loop reactor is considered where the shield-deck requirements are comparable.

Due to the similarities between both the large loop and pool decks, from now on only the pool will be used as an example for illustrative purposes, in fact, the arguments and discussion also apply to large loop reactors. Because of the enormous size of the deck, both axially and radially, any significant variations in temperature distributions within the structure would impose unwarranted stresses upon the deck structure, causing the structure to bow. Consequently, if unchecked, this could lead to disruption of fuel handling and of reactor shutdown controls. The amount of heat transferred through the deck structure and the concrete should always be maintained within an acceptable low level. Thus, a reliable and safe operation of the deck structure becomes the criterion by which performance of an insulation is evaluated.

Besides the above constraints, there are additional requirements that an acceptable thermal insulation must satisfy. First, the insulation cannot in any way interact, or physically or chemically degrade, when it comes into contact with sodium. If the insulation has to be used on the underside of the deck, access for inspection, repair, or maintenance will be severely limited. Hence, it must be reliable throughout the lifetime (~ 30 - 40 yr) of the plant without scheduled maintenance or inspection. Moreover, since the insulation is exposed to high-temperature sodium $\sim 950^{\circ}\text{F}$ (510°C), large quantities of aerosols are expected to be transported from the hot sodium surface to the colder surfaces of the insulation on which they will condense. This, in effect, will reduce the insulating capability enormously over a long period of time. The insulation should therefore exhibit the inherent characteristic of high resistance to aerosol penetration. For large reactors, sodium sloshing caused by seismic disturbances and impulses, or any other safety-related hypothetical design events, could impose on the insulation tremendous impacts to which it is not normally subjected. Hence, it must also be able to sustain the aforementioned events without compromising its integrity.

In view of the above requirements, many types of commonly considered thermal insulation can be eliminated, for example, fiber glass or calcium silicate,

since these materials are incompatible with sodium. Most refractory materials, due to their ceramic nature, cannot be considered either. Some metallic reflective foils, a few mils thick, can indeed sustain long time exposure to sodium, and yet their lack of physical strength poses support and seismic problems. Layers of metallic plates spaced a certain distance apart were proposed to be used as PLBR insulations. The British use this approach in their Prototype Fast Reactor (PFR). The advantage of the multiplates is that they are simple and inexpensive.

A different kind of insulation is proposed to be used by the French in their SuperPhenix (1200-MWe) commercial-size plant. It comprises layers of metallic plates, each separated by layers of screens. Other more exotic types of insulation were also suggested, e.g., honeycomb composites. All these are to be made of either Type 304 or 316 stainless steel, whose compatibility with sodium is well known. Each of them provides its own uniqueness in realizing the objectives as LMFBR insulation. Certainly, other kinds of insulation might lend themselves to be possible thermal insulation for an LMFBR system. This study, however, is confined to discussion of the above three types.

Among the types of insulation proposed, the multiplate approach is the most studied one. However, the effects on thermal properties for various numbers of layers need to be examined. The French-type screen insulation, on the other hand, is the least known. To effectively use the screen-type insulation, various aspects of heat-transfer processes will have to be identified and understood. Metallic honeycomb has been widely applied in many areas, but not as a thermal insulation. Its thermal characteristics as a function of temperature require further investigation. The presence of sodium and/or sodium oxide on any of these types of insulation may substantially affect their thermal performance. Little information is available on these long-term effects. Therefore, it is important to have experimental results using sodium to simulate conditions in which insulation performance in a reactor can be studied. All these constitute different vital areas in which analytical and experimental studies would lead to increased confidence in plant design and reliability and form the basis for this study.

II. LITERATURE SURVEY

A. Design Approaches, Proposed Concepts, and Factors Affecting Insulation Performance in LMFBR Systems

Since the mid 1940's, different types of fast reactors were built, many having very small cores as well as small thermal outputs. Clementine, the first fast reactor developed at Los Alamos Scientific Laboratory, had a core 5.9 in. (150 mm) in diameter and 5.5 in. (140 mm) high, completely surrounded by steel. This was reported by Journey (1954) and Arnold (1953). Later, reactors with similar core sizes but various core designs were built, e.g., USSR-BRS, EBR-II, and homogeneous-core reactors LAMPRE-I and -II. By the late

1950's, the United Kingdom built Dounreay with a 21-in. (533.4-mm)-dia, 21-in. (533.4-mm)-high core. All the reactor closures aforementioned had only very simple dome structures. Also, because of their reduced physical sizes, thermal-related problems were only of a minor nature. All these systems were cooled by liquid metals, e.g., sodium, mercury, or NaK.

By the 1960's, the United States embarked on a nuclear-reactor program in which two different design concepts were studied, reviewed, analyzed and finally built. They were the Experimental Breeder Reactor II (EBR-II) and the Fermi Reactor. These two design concepts represented a marked difference in design approaches from then until the present. A major characteristic of EBR-II is that, unlike its predecessors, both its primary pumps and its intermediate heat exchangers are placed within the reactor vessel. Despite the fact that the core of EBR-II is almost six times smaller than the one of Fermi, the reactor vessel of EBR-II is 26 ft (7.93 m) in diameter versus 14.5 ft (4.42 m) for the Fermi Reactor. As can be seen from Fig. 1, the closure of the Fermi Reactor essentially comprises the fuel-handling and -control mechanisms that form an integral part of the closure. Figure 2 shows an elevation view of the primary tank of EBR-II. The cover is made of a carbon-steel plate connected to the primary tank wall, forming a boundary for the sodium aerosols. Conventional thermal insulation is placed on the top side of this carbon-steel plate to provide the necessary thermal gradient for the upper structure.

Later, the French built the Phenix Reactor, in which major resemblances with EBR-II can be noted. Among the similarities is the reactor cover, as can be seen in Fig. 3. The name pool-type reactor is adopted to describe such a large reactor vessel within which is contained the primary heat-transport system. At about the same time, the United Kingdom constructed the Prototype Fast Reactor (PFR), which is also a pool type. However, instead of an integral deck, the British chose to place the thermal insulation below the deck, thus enabling a low-temperature closure structure, as seen in Fig. 4. Only recently, the French started construction on the full-size (1200-MWe) demonstration plant, which is also a pool reactor with a reactor vessel about 70 ft (21.3 m) in diameter. Contrary to their Phenix design, they adopted a PFR-type deck approach, shown in Fig. 5, in which thermal insulation is required below the closure structure. During the last few years, other pool-type design concepts have been proposed and studied, and most of them favor the cold-deck underside-insulation approach. Some of these concepts are shown in Figs 6 and 7.

An alternative approach, in which the primary heat-transport system is located outside the reactor vessel, is conveniently called the loop concept. The Fast Flux Test Facility (FFTF), shown in Fig. 8, is a design of this kind that is being built in the United States. The West German SNR-300 (Fig. 9) and the Japanese Monju (Fig. 10), both of which are loop reactors, are also being constructed. Despite the fact that these reactors are designed by different countries, the closures of both reactor vessels are extremely similar. For the pool reactors, the closure consists of the rotating plugs and the shield deck,

Fig. 1. Cross Section of Fermi Reactor. Conversion factor: 1 in. = 25.4 mm.

Fig. 2. Elevation View of EBR-II

- A. CONTROL ROD MECHANISM
- B. FUEL TRANSFER ARM CONTROL
- C. INTERMEDIATE HEAT EXCHANGER (6)
- D. PRIMARY CIRCULATION PUMP (3)
- E. FUEL TRANSFER ARM
- F. ROTATING PLUG
- G. SURROUNDING NEUTRON SHIELDING
- H. TOP NEUTRON SHIELDING
- J. CORE
- K. 21 SUSPENSION POINTS
- L. MAIN TANK
- M. LEAK TANK
- N. SAFETY TANK
- O. CONICAL SUPPORT COLLAR FOR REACTOR

Fig. 3. Elevation View of Phenix Reactor

Fig. 4. Elevation View of Prototype Fast Reactor

Fig. 5. Elevation View of SuperPhenix Reactor

Fig. 6. Elevation View of Commercial Fast Reactor

Fig. 7. Elevation View of BN-600

Fig. 8. Elevation View of Fast Flux Test Facility

Fig. 9

Elevation View of SNR-300.
All units in meters.

Fig. 10
Elevation View of Monju

whereas the loop-reactor closure consists of only rotating plugs. Recent PLBR studies for a commercial-size loop-type LMFBR show that larger reactor vessels, in the order of 40 ft (12.2 m), are favored. This will result in a reactor vessel design comparable to the pool concept, in which both the rotating plugs and the shield deck make up the closure of the vessel (information primarily obtained from reports by A. Amorosi et al. and other internal reports at ANL).

A principal reason why thermal insulation has been such an important area in which detailed information is needed, is because, previously, for loop-reactor closures, thermal gradients and thermal expansions can easily be accommodated by free-end support arrangements. Furthermore, the reduced sizes of earlier generations of reactors serve to diminish such difficulties; therefore, the requirements of thermal insulations are much less critical. The integral-deck or hot-roof approach for the first generation of pool reactors poses unusual thermal-stress and thermal-stripping problems during normal operating conditions. These difficulties are further amplified during transients and variations in primary-pump speed conditions. Therefore, coupled with the earlier operating experiences and the fact that reactor-vessel diameters are progressing to larger dimensions, "under-roof" thermal insulation has become increasingly vital and essential for building a reliable LMFBR system.

Only recently has the topic of thermal insulation begun to receive some limited attention. Because of the location at which the insulation will be placed, certain unique situations will have to be considered. In addition to the requirements outlined in Chapter I for thermal insulation, a recent report by Collins (1978) also specifies that adequate flexibility and structural compliance to accommodate normal thermal dilatation between the insulation, the shield deck, and components that penetrate the shield and insulation are needed to avoid damaging thermal stresses. The insulation must retain its integrity during and after design seismic events, including forces from structural accelerations and inertial sloshing of sodium. These become the basic requirements used to evaluate any proposed insulation.

Different materials have been examined for their compatibility with sodium during various nuclear-reactor-related studies, e.g., Balkwill (1974) and Fink (1976). Their conclusions indicate that because of sodium's extraordinary reducing properties, almost all conventional thermal insulation will not be compatible. Some react exothermically with sodium; others react violently. Generally, materials containing silicate react with sodium. Some metals, e.g., copper or iron, are also susceptible to sodium reaction, as reported by McKisson (1970). Only a few metallic materials proved to be attractive, and among them are Types 304 and 316 stainless steel. Low thermal conductivity, comparative low cost, and wide availability render stainless steel of both kinds to be very promising insulation material.

In most proposed LMFBR systems, normal operating sodium temperature is 950-1000°F. Figure 11 shows that at such temperatures sodium vapor pressure is

quite substantial. This implies that large quantities of sodium-vapor aerosols will be created and, by means of diffusion and convection, the aerosols will be transported from the pool surface up to the insulation, where condensation and deposition will take place, as seen from Fig. 12. Any extended penetration of sodium aerosols into the insulation will almost certainly reduce its insulating effectiveness. Perhaps a similar detrimental effect as a consequence of aerosol transport is the oxidation of the sodium into sodium oxide or super oxide, which exhibits a very high thermal-emissivity value. Aside from the thermal degradation caused by increased thermal radiation and conduction heat transfer, both sodium and its various oxides could also prevent proper operation of equipment, e.g., the rotating-plug seals of EBR-II reported by Copper (1969) and control-rod mechanisms and rotating plugs of Phenix reported by Delisle (1977). Therefore, in addition to the requirements stated earlier, the thermal insulation should also be able to provide some form of vapor-transport barrier to reduce the deposits of sodium and sodium oxides on the shield deck as well as on components supported from the deck.

Fig. 11. Vapor Pressure of Liquid Sodium below 1250 K.
Conversion factor: 1 atm = 0.1 MPa.

Fig. 12. Sodium Aerosol Concentration

B. Types of Insulation

Within the last few years, various kinds of metallic thermal insulation have been proposed for these LMFBR systems. Despite the unique requirements presented in Sec. A above, different ideas originally conceived for other industrial applications are examined and modified. One of these is the multi-layer insulation concept. Other ideas of encapsulating conventional thermal insulation are also explored.

Thermal baffles, which are essentially layers of reflective materials, have been widely used as thermal insulation for a long time. During the early development of liquid-metal-cooled reactors, a similar concept was used, e.g., in Hallum and SRE. Figure 13 shows the find of multilayer metallic plates used in the two reactors. The white deposits are sodium and sodium oxides. During

the 1960's the application of multilayer reflective-radiation shield was further perfected by the aerospace industry. Various parametric and thermal characteristic studies were performed. The work is summarized in the NASA report by Glaser et al. (1967). However, these types of insulation were developed for lightweight and compact applications, and mechanical strength was not a major concern. Hence, metallic foils, whose thickness was in the order of a thousandth of an inch or hundredth of a millimeter, were commonly used. Rarely would these insulations exceed an overall thickness of a few inches (200 mm). Due to the lack of strength of the reflective shields, a variety of spacer materials was proposed to remedy the direct contact problem between foils. Paquin (1968) proposed the use of high-purity refractory oxides as spacers. Some of the more common ones are silk, Mylar, dimpled sheets, paper, and mica. They are chosen because of their poor thermal conductivities and, thus, contribute in reducing conduction heat transfer from one layer to another.

Fig. 13. Multilayer Plates after Exposure to Sodium

Plain multiple-layer reflective plates have been proposed by Atomics International and Westinghouse to be used in their PLBR loop systems. This type of insulation is actually being used inside the British PFR system. Nevertheless, reservations concerning the integrity of the insulation over long time exposure to sodium aerosols can be demonstrated by the French revolutionary approach of

a patented concept by Lemerrier (1973) in which metallic screens are added between each pair of reflective shields. Detailed information on the design of this type of insulation is not available. It is unclear whether one of the patent concepts or a modified version of them will actually be used in the SuperPhenix Reactor. Limited sources indicate that screen-filled multilayer reflective insulation will be used. Both French patents describe some sort of encapsulation with a stainless steel container sealed by capillary effects. Different screen-wire dimensions and mesh sizes have been presented. A paper presented in 1976 by one of the inventors, Lemerrier, reported the conclusions of a testing program of this type of insulation; but variance is noted on such critical dimensions of the insulation as screen-wire size, mesh size, and thickness when compared to the patents. A summary is presented in Table I. Recently, insulation of layers of screens and layers of plates was proposed by General Electric to be used in their PLBR pool design.

TABLE 1. Summary of French Screen-type Insulation

	Patent No. 73 23338	Patent No. 73 23339	Lemerrier (1976)
Wire diameter (mm)	0.4	0.4	20.6 ^a
Wire mesh (mm)	4	NA	NA
Overall thickness (mm)	NA	120	NA

^aWire gauge screen.

Other ideas of an acceptable thermal insulation have also been suggested, one of which is the honeycomb approach, which will receive an extensive treatment here. The purpose of using honeycomb-type material in different applications varies widely from structural strength, to noise control, to solar collectors. Metallic honeycombs are widely used in the aircraft industry to build wings, fuselages, and other parts of planes. Incidentally, at Argonne National Laboratory, the idea was conceived that if face sheets, which were essentially reflective thermal-radiation shields, could be placed between a honeycomb core forming a sandwich-type thermal insulation, as shown in Fig. 14, then, by arranging alternate layers of honeycomb core on face sheets, any

Fig. 14

Schematic Diagram of
Honeycomb Insulation

thickness of such insulation could be made. This concept, on the other hand, can be viewed as a multilayer reflective-shield insulation using honeycomb cores as spacers. Also in the PLBR pool studies, other concepts such as the "Solami" have also been explored by Collins (1978).

Most of the authors, in their study of heat transport through the insulation, assume contributions from different modes of transfer, the sum of which is the total amount of heat transferred across the insulation.

C. Contact Resistance

The study of contact resistance, also referred to as interface resistance, can be dated back to the early forties when Mersman (1943) showed a one-dimensional heat-conduction problem of two plane-boundary semi-infinite homogeneous solids of dissimilar materials with an imperfect contact along their interface. This contact was represented by a discontinuity in temperature at the interface. After elaborate mathematical theorems were established, a complex solution resulted. Schaaf (1974) later furthered such treatment by incorporating a heat source in the original problem. Despite its practical applications, interest in this area was primarily confined to exotic mathematical treatment due to lack of adequate calculating facilities to render the subject more realistic. This interest was revised in the sixties by the rapid development in the aerospace industry. Precise understanding and knowledge of contact resistance is critical in numerous space related heat-transfer studies, e.g., high- and low-temperature shields (Barzelay, 1954, 1955). Aside from thermal radiation, conduction is the only other means of heat transfer in such environments.

Therefore, up until the early sixties, thermal resistance was either estimated or experimentally measured, e.g., by Willis and Ryder (1949) and Brunot and Buckland (1949), or simply neglected.

Later some theoretical treatments analyzed heat transfer by conduction through contacts and the interface fluid. Some used cylindrical geometries to model the contact areas. Contact surfaces of steel, brass, and aluminum were also experimentally studied. Semiempirical analysis was made by using data on steel to brass and steel to aluminum contacts. Fenech and Rohsenow (1963), neglecting both radial-conduction and natural-convection heat transfer within the contact area and using average boundary conditions, solved the Laplace equation, from which they obtained an expression of thermal conductance across the interface. Based on this approach, more detailed calculations using an analog computer were reported by Henry and Fenech (1964). Clausen and Chao (1965) summarized the assumptions inherent in theories presented up to 1965 in the literature, in which (1) the actual areas of contact were uniformly distributed over the entire area, (2) the contact areas were circular, (3) the asperities deform plastically under the load, so that the average pressure exerted between them equaled the microhardness, and (4) the film surface resistance was negligible. The authors further proposed a model where the contact

resistance was represented by the sum of the macroscopic constriction resistance, microscopic construction resistance, and the film resistance of the interface. Later, Clausing (1966) extended the model to describe heat transfer between dissimilar metals. More refined theoretical treatments and experimental data were presented by Clausing (1966), Mikic and Rohsenow (1966), and Roca and Mikic (1971).

As the mathematical model describing the interface improved, the lack of accurate information about the contact surface became more pronounced. Previously, different idealized models could assume certain well-behaved boundaries, whereas more sophisticated treatment required detailed data about the contact surfaces. Clausing (1966) used the microscopic constriction concepts to incorporate the surface into his model. Probabilistic density-distribution functions were derived by Mikic and Rohsenow (1966) to account for variations of the material surfaces. Different variances in different directions of the surface were considered. Copper et al. (1969) further challenged the Gaussian-distribution assumption of heights and proposed a different set of statistical distribution functions arrived at considering the original surface profile.

More idealized treatment of the interface surface assuming parabolic contacts was addressed by Yovanovich (1970), in which complex transformation to the ellipsoidal coordinates provided a solution to the Laplace equation with mixed boundary conditions. The thermal-contact resistance, it was reported, was directly proportional to a geometrical function of the contact paraboloids. Experimental results were also presented to substantiate the validity of the theory. Effects of cyclic loading, over a pressure range of $4.45\text{--}34.5 \times 10^5 \text{ N/m}^2$ (64.5–500 psi), on thermal conductance were measured experimentally by McKinzie (1970). Agreement was found between data and theories proposed by Clausing (1966) as well as McKinzie. Similar measurements were made by Cassidy and Herman (1969) at ambient pressure of 1 atm (0.101 MPa) to $3 \times 10^{-12} \text{ m/m Hg}$ ($4 \times 10^{-10} \text{ Pa}$), and limited agreement was calculated with Mikic theory. In addition to accurately modeling the surface conditions, pressure distribution on the interface was also considered in the theoretical treatment by Roca and Mikic (1971).

Despite these exotic treatments on the subject of thermal-contact resistance, there is still a chasm between engineering-application data and theories developed, especially for complex geometrical contacts. Experimental data for soldered joints bolted or screwed sheet metal joints were reported by Yovanovich and Tuarze (1969), and Veilleux and Mark (1969). Smuda and Gyorog (1969) conducted an experimental investigation on thermal characteristics of materials inserted between plane-parallel metal surfaces. Load pressures and mean junction temperatures were varied from 25 to 1000 psi (0.172 to 6.89 MPa) and -100 to 200°F (-73.3 to 93.3°C). Screens of various mesh sizes and materials were reported tested where acceptable thermal-conductivity properties coupled with their superb mechanical strength rendered the use of these materials as an insulation uniquely attractive. Later, Gyorog (1970) further extended the investigation to include other interstitial materials and configurations. A dimensionless correlation was presented, based on the functional relationships of load, screen dimensions, and material properties. Notable improvements were observed in thermal-insulation characteristics by separating the wire screens with shim materials.

D. Convection

In 1900, Bernard experimentally studied the critical temperature gradient that, when exceeded, would establish natural convection for horizontal plates heated from below. Lord Rayleigh (1926) later laid the foundations for the theoretical treatment of Bernard's experiment.

DeGraaf and Van Der Held (1952) later reported the observation of Bernard cells by injecting smoke into the test cavity. The critical Rayleigh number was found to be around 2000. Later, other studies were performed to further investigate the onset of natural convection between parallel plates. Chandrasekhar (1971) summarized both the theoretical and experimental studies, where a critical Rayleigh number of 1709 was reported. Figure 15 depicts the Nusselt number

Fig. 15. Nusselt Number vs Rayleigh Number for Parallel Plates. ANL Neg. No. 113-77-351.

as a function of the Rayleigh number. For Rayleigh numbers less than 1709, the Nusselt number is virtually unity, implying that conduction prevails. It was noted later that, by introducing side walls between horizontal plates heated from below, thermal stability was further enhanced, thereby delaying the onset of natural convection to a much higher Rayleigh number.

Theoretical treatments of infinitely long cylinders were presented by Ostronmov (1958) and Yih (1959). Boundary conditions of perfectly conducting and perfectly insulating wall were both addressed. Analytically, Malkus and Veronis (1958) used an integral technique, applied by Stuart (1958) to the mathematically analogous problem of predicting momentum transfer through a fluid between two closely spaced, concentric rotating cylinders. Ostrach and Pnueli (1963) had considered perfectly conducting walls of cylinders of arbitrary cross section using an approximation scheme. The results of Catton et al. (1974) showed that the amount of suppression by continuing walls is a function of both cavity-aspect ratios. Interferometric study of critical Rayleigh numbers by Norden and Usmanov (1972) also helped to independently evaluate the condition of natural-convection onset. Lighthill (1953) used the integral method to study natural convection in a closed-end tube of various aspect ratios with constant-temperature walls. Experimental verification was obtained by Martin (1955) and Harnett and Welsh (1957). Low Rayleigh-number convection inside a spherical cavity was investigated by Drakhlin (1952). Using finite-difference methods, Wilkes (1963) numerically solved the transient and steady-state problems for natural convection in rectangular cavities with isothermal walls and either perfectly conducting or perfectly insulating horizontal surfaces with a unity aspect ratio. It was demonstrated that, for a Rayleigh number of about 10^5 , horizontal isotherms were observed in the interior with a temperature gradient established in the vertical direction such that temperature increased upwards. De Vahl Davis (1968) concluded that, in a similar configuration, the vertical temperature gradient in the center of the cavity is essentially zero for small Rayleigh numbers. As for increasing Rayleigh numbers, the slopes of the isotherms tend to become negative and the temperature gradient approaches an asymptotic positive value, which is highly dependent upon horizontal-wall conditions.

E. Radiation and Emissivity

The literature on heat transfer from radiating stationary media without conduction, convection, and other energy-transfer mechanisms are summarized by Jakob (1957). Early discussions by McAdams (1954) were confined to radiative heat transfer in boiler furnaces and related areas. Radiative exchange between surfaces separated by an absorbing and scattering medium involves considerations of (1) the configuration of the surfaces, (2) the radiative properties of the surfaces and the medium, and (3) their temperature distributions. In McAdams (1954), Hottel used a finite-difference method to present a solution of gray absorbing and emitting gas at constant temperature with heat source and heat sink. An improved prediction was reported by Hottel and Cohen (1958), accounting also for variation of temperature in the medium. The solution to

the heat-exchange problem by radiation using the electrical-network method was suggested by Oppenheim (1956). Bevens and Dunkle (1960) extended the concept and solved a multinode network problem. A vigorous demonstration of the analogy of the electrical-network model to the solution of the two integral equations that described the process of radiative heat transfer in a closed system with absorbing and scattering surfaces was reported by Adrianov (1959), in which the two integral equations could be approximated by a system of linear algebraic equations, which would result in the equivalent solution of the network model.

Jakob (1957) has summarized the literature of configuration factors, also known as views factors, shape factors, angle factors, or geometric factors, for a few simple geometries for radiation through absorbing and nonabsorbing media. Certain other straightforward geometries were investigated and tabulated by Pittman and Bushman (1961), Hamilton and Morgan (1952), and Person and Leuenberger (1957), and Tripp et al. (1962). Instead of evaluating the double-area integrals, Sparrow (1965) proposed contour integrations whereby the expression was reduced to a simpler form. However, more complex but continuous geometries will still have to be analyzed numerically. For highly discontinuous materials, approximations will have to suffice.

The term "radiosity" was defined by Hottel (McAdams, 1954) to denote the amount of energy leaving a surface as a result of the energy emitted and reflected by another surface. With this concept, n equations can be written for an n -surface system based on the energy-conservation principle, whereby the unknown variable can be solved. Gebhart (1961) introduced another method for the solution of gray-body problems. Assumptions required by his method were the same as those used for the Hottel method; that is, surfaces were gray, diffuse, and uniform in temperature.

Radiation-coupled conduction and convection can be divided into two categories. The first involves radiation passing through an absorbing emitting medium, where net radiant energy is transferred to or from each element of the medium. Conduction and convection transfers can be treated as heat sources and heat sinks, and the conservation-of-energy equation is an integrodifferential equation. The other category deals with radiation interaction through the boundary conditions of conduction and convection processes. Viskanta (1960) presented the first complete formulation of infinite parallel plates with an absorbing medium, in which numerical iteration procedures were used to solve the governing equation for several combinations of parameters. A thorough treatment of radiation transfer coupled with free convection was presented by Cess (1964), in which the ratio of the Nusselt number to the Grashof number was expressed in terms of an infinite series. Free convection without radiation reduces the expression to only the first term. Experimental investigations were undertaken to augment the analytical studies, and reasonably good agreements were reported.

Emissivity is expressed as a ratio of a characteristic radiation emitted by a surface to that of a black body. There are different types of emissivities, depending on the various characteristics of the radiation; e.g., spectral emissivity denotes the ratio of radiation of a certain wavelength emitted by the surface conditions of the material to the emission of that wavelength from a black surface. An oxide film of a few microns on the surface is sufficient to alter its value significantly. Only literature pertinent to this study will be reviewed.

Eckert et al. (1957) measured the total normal emissivity of porous materials by an energy balance between convective and radiative heat flow. Two different types of porous surfaces were studied: Poroloy and modified Tyler materials. The former was fabricated of Type 304 stainless steel wire wound on a mandrel, layer after layer, to attain the desired porosity; then the material was sintered. The latter was essentially composed of Type 304 stainless steel screens fabricated by a special weaving process. High porosity and mesh sizes showed increased total normal emissivity.

Wade (1959) and Wade and Slemp (1962) reported that oxide coating formations were observed on all their Type 347 stainless steel samples at an oxidation temperature of 2000°F, and total normal emissivity values changed from 0.3 to 0.85 in the order of hours. A comprehensive literature survey was summarized by Touloukian (1970) in which emissivities of different types of stainless steel under different conditions were presented. The effects of surface films on radiative properties of metallic surfaces was treated theoretically by Cravalho et al. (1969), in which the relative effect was found to depend on the reflective index of the film, and naturally occurring oxide films had no effect on the total hemispheric-radiation properties except at high temperatures. Brannon and Goldstein (1969) reported that an increase in oxide-film-layer thickness would result in a similar increase of emissivity.

Information on emissivity values of sodium oxide is extremely limited and incomplete. To my knowledge, there is no good documentation on such measurement for sodium oxide. Through private communication with the staff of Atomics International, I determined that some crude experimental measurements were made on sodium oxide on one occasion in which the emissivity value was estimated to be between 0.9 and 1.0.

F. Gas Conduction

When the value of the Rayleigh number is well below the threshold at which natural convection occurs, conduction prevails. This becomes especially straightforward for gaseous conduction between parallel plates. On the other hand, gas conduction within honeycomb configurations lately has also received a great deal of attention, primarily from the standpoint of solar-energy applications. Studies include thermal stability and heat transfer of inclined honeycomb cells, e.g., reports by Randall et al. (1977) and Buchberg et al. (1976). Attention is focused primarily in trying to minimize convection in order to

prevent heat losses. Various aspect ratios have been investigated, and rectangular cell structure is found to be an effective device to suppress natural convection. Under this condition, the application of the heat-conduction equation to account for the gas-conduction heat transfer is sufficient. Thermal stability for irregular cell sizes and geometries has not received much attention. Since the convection onset depends heavily on the cell geometry, arbitrary configurations limit the ability to pursue any meaningful studies. However, given the configuration presented by layers of screens stacked together, thermal-stability analysis of other fundamental geometries--namely, horizontal cylinders, vertical flat plates, and inclined flat plates--will have to suffice as some form of estimate of the complex geometry.

Another kind of gas conduction has not been addressed previously, namely, heat transfer in rarefied gases. Despite the fact that, in reactor applications, this type of energy transport is extremely uncommon, experiments were selected for this study to test all insulation assemblies under evacuated conditions in order to be able to better distinguish various forms of heat transfer. If the mean free path of gas molecules is large compared to the characteristic system length, conduction can be approximated well by Knudsen's formula. However, given the equipment-design criteria, the contribution from rarefied-gas heat transfer is not measurable by the existing setup. Therefore it will be assumed that, for low-density gases, conduction can be neglected.

G. Sodium and Sodium Oxides

One consequence of using liquid sodium as a heat-transfer fluid is the formation of sodium aerosols during high-temperature operation that can be transported to and deposited on different components, causing unnecessary maintenance. Since requirements of high thermal efficiency prevail, the operating temperature of the primary sodium is about 1000°F (538°C), and the amount of sodium aerosols generated increases proportionally with temperature. This is clearly shown in Fig. 10, where the vapor pressure of sodium is plotted against temperature. An increase of 100°F, from 900 to 1000°F (482 to 538°C), in temperature will result in a fourfold increase in vapor pressure. Therefore, to estimate the amount of aerosols produced, the rate of vapor-nuclei formation can be evaluated, and one method is suggested by Sheth (1975). Vapor agglomeration will occur when aerosols of various sizes are formed. Knowledge of agglomeration dynamics implies an ability to calculate aerosol sizes that are critical in removal mechanisms or other sodium-related investigations, as described by Files et al. (1976). The authors provide a comprehensive treatment of aerosol removal by settling, surface plating, leakage, and ventilation. Nonetheless, forces that affect aerosol transport behavior are equally important to be understood. Brownian motion arises from the collision of aerosol particles with cover-gas molecules in thermal motion.

Gravity is another type of force to which aerosol particles are subjected. Both of these forces are dealt with by Greenfield et al. (1969). Force and

free convective forces are addressed also by Castelman et al. (1969). Other forces, namely, electrical, thermophoresis (temperature related), diffusiophoresis (mass concentration related), and photophoresis (light related) can also influence aerosol behavior; however, because of their limited influence in this study, they are not considered here.

To a greater or lesser degree, the operating experience of an LMFBR relates to the difficulty imparted from sodium aerosols and vapors in the cover gas. EBR-II reported sodium buildup from aerosol transport on the seals of rotating plugs, as discussed by Cooper (1969). Similar problems were also encountered at the Fermi Reactor. During the 10 years of operation at RAPSODIE and 3 years at PHENIX, problems of condensation of sodium in annular spaces were encountered. Plugging also occurred in gas pipes, filters, and vapor traps. The KNK Reactor reported that nonhomogeneously distributed deposits were formed between the large utility plug and the reactor vessel. These occurrences can be attributed to the geometry of the components and their maldistribution of temperature. Joyo also received its fair share of aerosol-related difficulties, and before Monju was built, a vigorous research and development program was launched to overcome such problems. These are some of the examples reported by the International Working Group on Fast Reactors [Himeno (1976)] et al. to illustrate the immense task of circumvention of sodium-aerosol transport. All countries who participate in LMFBR developments are engaging in experimental and/or analytical programs intended to reduce and control sodium-aerosol transport, as described by the International Working Group on Fast Reactors, IAEA (1977). Other sodium-related experiences are summarized by Funk (1974).

Of all the aforementioned means by which sodium aerosols are transported, convection is the most important. Diffusion or Brownian movement only becomes significant when the transport time due to convection is on the order of the diffusion time. Under typical reactor operating conditions, convection will prevail. The impacts of both natural and forced convection depend on a number of parameters: temperature gradient, temperature, geometrical configuration, fluid properties, and the velocity of the flow. For simple geometries and idealized conditions, both types of convection have received much attention, some of which, for natural convection, is given in Sec. II.D above. In EBR-II, forced convection has been introduced properly around annular regions, e.g., various types of components seals, to eliminate natural-convection cells formed from temperature maldistribution. Since both types of convection are highly sensitive to geometrical configurations, a thorough and complete analytical or experimental effort to model the deck structure and the cover-gas space is not entirely realistic. Simplified models by Shimazaki (1970), (1975) attempt to predict deposition rate and distribution of sodium aerosol due to thermal convection.

In a separate effort, evaporation from a sodium-pool surface was studied experimentally and, in parallel, a mathematical model of vapor formation was developed by Himeno et al. in JAPFNR-243 (1976). Deposition in an annulus

within an enclosed cover-gas space is also addressed by Himeno et al. in JAPFNR-245 (1976). Results of distribution within an annulus versus distance from the inlet are given. Similar studies are being made in the United Kingdom and the United States.

The conversion of sodium to different types of oxides (namely, sodium oxide, Na_2O , and sodium super oxide, Na_2O_2) due to the presence of oxygen from the cover-gas system, regular maintenance leakage, and residual oxygen of the structure and components, further complicate the situation. Oxides of sodium, unlike sodium metal, exhibit characteristics such as low thermal conductivity, high emissivity, low ductility, and high melting temperature. This markedly different behavior in physical and chemical properties of sodium oxides make controlling vapor transport difficult. Elevated temperature was used in the German RSB facility to change the condensed sodium deposit into its liquid state, i.e., pure sodium metals, whereby it can be drained back into the sodium pool. However, if a substantial amount of sodium oxide is present, the viscosity of sodium and sodium oxide mixture, coupled with poor wetting, may prevent effective drainage, as discussed by Jansing et al. (1977).

In the same report, the authors also try to show, through some photographic records, where and in what form sodium is deposited. Nonetheless, available information is very limited. The surface condition of the object on which the vapor will condense is believed to play a significant role. Other factors such as the temperature of the object and the vapor pressure are also important. Some authors have suggested dropwise condensation on the cold surface, where the surface tension of the liquid provides nonwetting conditions for droplets to form. Complete wetting only occurs when the surface is clean and the sodium is at a relatively high temperature. Figure 16 shows the data of wetting time versus temperature collected at the Liquid Metal Engineering Center (LMEC) of Atomics International. Elevated temperatures increase the solubility of oxides in sodium, thereby providing an oxide-free surface in a much shorter time. Therefore, it is conceivable that under various conditions the insulation proposed to be used below the deck could experience either or both of the above-described occurrences. The time needed for a substantial amount of sodium and/or sodium oxide to accumulate could also affect the nature, manner, and thickness of the deposits.

Fig. 16

Wetting Time vs Sodium Temperature.
Conversion factor: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$.

III. ANALYTICAL STUDIES

A. Thermal Convection

1. Natural Convection

A few geometries are of particular interest in convection heat transfer as they pertain to the different types of thermal insulation. They can be identified as the spaces between horizontal parallel plates, the square cells of the honeycombs, and the interstitial spaces between layers of screens. Closely coupled with the question of geometry is the determination of the conditions of the onset of thermal instability, which will result in convection heat transfer.

Based on the basic hydrodynamic equations, namely, the continuity, linear momentum and energy, and by using the Boussinesq approximation, which assumes that ρ is a constant in all terms except the external-force term, Chandrasekhar (1971) obtained the analytical solution for an infinite horizontal layer of fluid in which a steady temperature gradient is maintained. Three different types of boundary conditions were examined: the free surfaces where there is no tangential stresses, the rigid surfaces where no slip occurs at the boundaries, and the combination of these conditions.

For the free surfaces, the Rayleigh number is about 657; for both rigid surfaces, the Rayleigh number is 1708; and for the combined situation, the Rayleigh number is 1100. Experimental efforts by Schmidt and Milverton (1935), Silveston (1958), and others confirmed the analytical results that there is only heat transfer by the conductive mode for Rayleigh numbers below the critical value, e.g., ~ 1700 , in Fig. 15.

Between the Rayleigh number values of 1700 and 3200, the increase in the value of the Nusselt number follows a nearly linear relation. The flow patterns were first observed by Bernard (1900), who reported the formation of cells of similar sizes within which convection prevails, as shown in Fig. 17. Suppression of the existence of these convection cells will result in a unit Nusselt number, characterizing conduction heat transfer. Therefore, this can be simply applied to calculate conduction and convection heat transfer between horizontal plates. In Fig. 18, the Rayleigh number,

$$Ra = \frac{\rho^2 \beta g \Delta T C_p L^3}{k \mu}, \quad (1)$$

where

ρ = density,

β = coefficient of expansion,

ΔT = temperature difference,

L = characteristic length,

k = thermal conductivity,

μ = viscosity,

and

C_p = specific heat,

is plotted as a function of ΔTL^3 for three different fluids: air, argon, and helium. Air has never been and never will be considered as an LMFBR cover gas, but it is presented for the convenience of later experimental analysis. The extraordinary ability of helium to maintain thermal stability is evident from the figure.

Fig. 17. Flow Pattern for Bernard Cell

Fig. 18. Rayleigh Number vs ΔTL^3 for Parallel Plates. Conversion factor: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$.

Different authors have demonstrated that the presence of lateral walls reduces or even suppresses natural convection between horizontal surfaces with the lower surface maintained at a higher temperature. The Malkus power-integral technique can be adopted to estimate the amount of heat transfer from below within such an enclosure, as described by Edwards (1969). It uses an integral expression derived from the energy-conservation equation. The approximate velocity and temperature profiles of the most unstable convection, which are derived from a linear perturbation analysis of the equations of motion, are substituted into the expansion to obtain the final solution. Two different types of sidewall boundary conditions can be assumed: a linear wall temperature

in the upward direction, and an adiabatic condition between the fluid and the wall. At any vertical position, the vertical heat flux consists of a conductive and a convective term

$$q = -K \frac{\partial}{\partial z} T + \rho C_p V_z T, \quad (2)$$

where V_z is the vertical component velocity.

The total upward heat transfer per cell is therefore the sum of vertical heat flux of the fluid and of the wall:

$$\dot{Q} = A_w K_w \overline{\langle -\partial T / \partial z \rangle} + A_f K_f \overline{\langle -\partial T / \partial z \rangle} + A_f \rho C_p \langle V_z T \rangle, \quad (3)$$

where the brackets denote a vertical average and the bar denotes a horizontal average. It is further assumed that only horizontally symmetric temperature profiles are considered. This implies that, for the most unstable disturbances, the net heat transfer between the fluid element and the wall element vanishes. If L is the height of the cell and $-(\partial/\partial z)T$ is written as $(T_h - T_c)/L$, where T_h and T_c are the hot and cold temperatures, respectively,

$$Nu = \frac{\bar{q}L}{K_f(T_h - T_c)} = 1 + \frac{\langle V_z T \rangle}{\alpha \langle -\partial T / \partial z \rangle} \frac{\bar{q}L}{K_f(T_h - T_c)}. \quad (4)$$

The Nusselt number is known only if the temperature and velocity profiles are known. If the velocity vanishes, representing a special case in which only conduction prevails, the Nusselt number reduces to unity, as expected. Substitution of the velocity and temperature profiles obtained from perturbation study would enable the evaluation of the onset of natural convection within such cells.

The temperature profile can be written as the sum of an average temperature and a small temperature perturbation:

$$T(x, y, z) = \bar{T}(z) + T'(x, y, z). \quad (5)$$

When Eq. 5 is coupled with the continuity equation that specifies

$$\overline{V_z} = \frac{\int dx dy V_z}{\int dx dy} = 0, \quad (6)$$

it follows that

$$\overline{V_{zT}} = \overline{V'_{zT}}. \quad (7)$$

Similarly to Eq. 5, the velocity distribution can be written as

$$V_z(x, y, z) = V'_z = \sum_{i,j,k} f_{i,j,k}(x, y) g_k(z) \quad (8)$$

in which the unperturbed average velocity is assumed to vanish. The boundary conditions at $z = \pm L/2$, when invoked, allow V'_z and T' to be written as

$$V'_z = \sum_k \sum_{i,j} f_{i,j,k}(x,y) \cos \frac{k\pi z}{L} \quad (9)$$

and

$$T' = \sum_k \sum_{i,j} U_{i,j,k}(x,y) \cos \frac{k\pi z}{L}. \quad (10)$$

When the averages are taken from both the horizontal and the vertical components, the properties of the periodic functions enable Eq. 9 to be recast in the form

$$\langle V_z T' \rangle = \sum_k \langle V_{z_k} T'_k \rangle. \quad (11)$$

Thus, by substituting the assumed velocity and assumed temperature profiles in the energy equation, Catton and Edwards (1967) showed that the Nusselt number can be written as

$$Nu = 1 + \sum_{i=1}^M Ni \left[1 - \frac{R_i}{R} \right], \quad (12)$$

where

$$R_M < R < R_{M+1}, \quad (13)$$

$$Ni = \frac{\langle Q_i \theta_i \rangle^2}{\langle Q_i \theta_i \rangle^2 - i}, \quad (14)$$

$$Q_i = \frac{L}{\alpha} v_i, \quad (15)$$

$$\theta_i = \frac{\beta g L^3}{\alpha \nu} T'_i, \quad (16)$$

$$R = \frac{\beta g \Delta T L^3}{\alpha \nu}, \quad (17)$$

and M is the highest mode of convection initiated for a given Rayleigh number. Ostrach and Pruehli (1963) estimated R_i to depend on the horizontal wave number, which for a square cell of width D is expressed as

$$a = \frac{\pi L \sqrt{5}}{D} . \quad (18)$$

If the walls of the cell are perfectly conducting, the horizontal wave number will be the same as shown by Eq. 18. For adiabatic walls, the adjusted wave number is

$$a' = 0.75 \pi L \sqrt{5} / D \text{ for conducting walls} \quad (19)$$

and

$$a' = a \text{ for diabatic walls.} \quad (20)$$

The vertical and horizontal wave numbers for closed cells can be approximated by

$$b_i = k\pi + 0.85, \quad (21)$$

$$A_i = a^2 + \frac{b_i^2}{2}, \quad (22)$$

and

$$R_i = \frac{(A_i^2 + b_i^2)^3}{A_i^2} . \quad (23)$$

The values of N_i for infinite horizontal surfaces were determined by Catton et al. (1974) and are presented in Fig. 19 and Table II. These values will be used as an approximation for N_i . A limited application of the above method was reported by Sun and Edwards (1970), where rectangular cells of ratios 2 and 3.6, and 4 and 15.2 were examined for aspect ratios of 4 and 3.87, respectively. For reference, experimental data were also presented for stainless steel, as plotted in Fig. 20. The data collected only for stainless

Fig. 19
Ni vs Wave Number a

TABLE II. Values of Ni as a Function of Wave Number a , from Catton (1966)

a	Values of i				
	1	3	5	7	9
1.0	1.435	1.745	1.850	1.918	1.974
12.0	1.609	1.777	1.865	1.927	1.981
21.0	1.737	1.813	1.881	1.937	1.990
30.0	1.806	1.847	1.898	1.948	2.000
42.0	1.857	1.882	1.921	1.965	2.000
54.0	1.888	1.907	1.940	1.980	2.000

Fig. 20. Nusselt Number vs Rayleigh Number for Stainless Steel Cylinder

steel cylinders, show that, with L/D equal to unity, the critical Rayleigh number is about 2×10^4 , which is considerably higher than for parallel plates.

2. Natural Convection within a Porous Medium

The phenomenon of natural convection through porous media has received increasing attention due to its widespread applications, principally in the areas of geophysics and thermal-insulation designs. Under the assumption that the space to be considered is homogeneous, the actual flow of fluid in a porous medium from a phenomenological standpoint closely resembles the hypothetical convective motion in a fluid layer of the same rate. However, the flow-resistance expressions for the two cases do differ in that, for porous

media, instead of the shearing stress being proportional to the velocity gradient, Darcy's Law is applied,

$$\left(\frac{\mu}{K}\right)\tilde{V} = -\nabla p, \quad (24)$$

where μ is the viscosity, K is the permeability of the medium, $\tilde{V}$ is the macroscopic velocity, and p is the pressure of the fluid. If Cartesian coordinates are adopted for the porous medium that is uniform between two horizontal impermeable surfaces separated by a distance ℓ , the equations of continuity, motion, and energy can be expressed as

$$\frac{D\rho}{Dt} + \rho \nabla \tilde{V} = \frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \tilde{V}) = 0, \quad (25)$$

$$\rho \frac{D\tilde{V}}{Dt} = \rho \frac{\partial \tilde{V}}{\partial t} + \rho \tilde{V} \cdot \nabla \tilde{V} = -\rho g - \nabla p - \frac{\mu}{K} \tilde{V}, \quad (26)$$

and

$$\frac{DT}{Dt} = \alpha \nabla^2 T, \quad (27)$$

where ρ is the density of the fluid, g is the vector presentation of gravitation acceleration, T is the temperature of the porous medium, and α is its thermal diffusivity. Note that the conventional viscosity term is replaced by Darcy's Law, which is valid for Reynolds number based on one pore diameter being less than unity, as stated by Lapwood (1948).

Based on these equations, the onset criterion of natural convection can be evaluated for the $\tilde{V}_z$ components in ways similar to those used to predict the horizontal fluid layer, except for vanishing velocity boundary conditions at $z = 0, \ell$. Lapwood derived the condition to be

$$Ra \frac{K}{\ell^2} = 4\pi^2, \quad (28)$$

where

$$Ra = \frac{g\beta\ell^3\Delta T}{\nu\alpha}, \quad (29)$$

in which β and ν denote, respectively, the coefficient of expansion and the kinematic viscosity, and ΔT represents the temperature difference between the bounding surfaces. Before Eq. 28 can be used, two parameters would first have to be determined: the medium's thermal diffusivity and its permeability K .

Strictly speaking, the time-dependent density derivative in Eq. 25 should be replaced by $(\partial/\partial t)\epsilon\rho$, where ϵ is the porosity of the medium. However, in obtaining Eq. 29, we assume the variation of density to be negligible except due to temperature differences. Second-order approximation terms are neglected. Lastly, it is also assumed that the criterion of onset rendered the partial time derivative zero.

If Eq. 29 is plotted with Rayleigh number versus K/ℓ^2 , for Ra less than 8×10^{-1} , the Rayleigh number continues to decrease, as plotted in Fig. 21. This is contrary to onset conditions reported for horizontal parallel plates. Thus, in the region where K/ℓ^2 is very large, the Rayleigh number can be expected to asymptotically approach $Ra = 1708$, the critical Rayleigh number, whereas for small permeability the critical Rayleigh number can be expressed by Eq. 29. The exact analysis of the transitional region is complex and difficult. Katto and Masuoka (1967) tried to arrive at the solution by adding the ordinary viscosity resistance $\mu \nabla^2 \tilde{v}$ to the momentum equation and were able to connect the theories of horizontal layer fluid and porous medium by using K , the permeability, as a parameter. The dashed line in Fig. 21 indicates the even solution obtained by the authors. The thermal diffusivity α should be that of the fluid expressed as

$$\alpha = \frac{k_m}{(C_p \rho)_{\text{fluid}}} \quad (30)$$

instead of

$$\alpha = \frac{k_m}{(C_p \rho)_m}, \quad (31)$$

where m denotes the porous medium.

Fig. 21
Rayleigh Number vs Permeability of the Medium

Based on the friction-factor assumption for laminar flow inside the packed columns, the superficial velocity, which is the average linear velocity the fluid would acquire in the column if no packing exists, is defined as

$$V_0 = \langle V \rangle \epsilon, \quad (32)$$

where ϵ is the void fraction

$$\epsilon = \frac{\text{Volume of void}}{\text{Volume of bed}} . \quad (33)$$

Based on the Fanning friction-factor expression, the average velocity can be written as

$$\langle V \rangle = \frac{(P_i - P_o) R_\kappa^2}{2\mu L} \quad (34)$$

where R_κ is the hydraulic radius,

$$R_\kappa = \frac{\text{Cross section available for flow}}{\text{Wetted perimeter}} , \quad (35)$$

and can be further written as

$$R_\kappa = \frac{\frac{\text{Volume of void}}{\text{Volume of bed}}}{\frac{\text{Wetted surface}}{\text{Volume of bed}}} = \frac{\epsilon}{\xi} . \quad (36)$$

Using Eq. 36, we can rewrite Eq. 32 as

$$V_o = \frac{P_o - P_L}{2\mu L} \frac{\epsilon^2}{\xi^2} \epsilon . \quad (37)$$

With the application of Darcy's Law (Eq. 24), the permeability of the medium, K , can be expressed as

$$K = \frac{\epsilon^3}{2\xi^2} . \quad (38)$$

B. Thermal Radiation

Thermal radiation is recognized as energy in the form of electromagnetic waves with wavelength, in general, ranging from 3 to 13 μm emitted by a medium due solely to the temperature of that medium. The Stefan-Boltzmann equation describes the energy flux for black body radiation,

$$E_b(T) = n^2 \sigma T^4, \quad (39)$$

where n is the index of refraction and σ is the Stefan-Boltzmann constant. However, normally encountered media and surfaces do not generally satisfy this equation. Consequently, assumptions and modifications of the equation will have to be made to better describe real media and surfaces. The assumptions are summarized as follows:

1. Gray body: This implies that ϵ_λ and α_λ are both independent of wavelength. In reality, very few materials satisfy this assumption over the entire range of wavelengths. Nevertheless, for finite range considerations, this will suffice as a good approximation.

2. Diffuse: This denotes directional uniformity. In other words, the intensity of the radiation leaving a surface resulting from emission or reflection is uniform in all angular directions.

3. Local thermodynamic and radiation equilibrium: This assumption allows the existence of a nonisothermal system in which the temperature within the system will be defined unambiguously. Radiation equilibrium requires the amount of radiant energy absorbed per unit time by a given volume to be equal to the amount of radiant energy emitted per unit time by the same volume. According to Kirchhoff's Law, in a particular direction and for each component of polarization, the monochromatic emittance ϵ_λ and the monochromatic absorptance α_λ are equal. That is,

$$\epsilon_\lambda(\theta_1) = \frac{E_\lambda}{E_{\lambda b}} = \alpha_\lambda(\theta_1). \quad (40)$$

As a consequence of the above assumptions, the following equality resulted:

$$\epsilon_\lambda = \alpha_\lambda. \quad (41)$$

With the introduction of these assumptions, radiative heat transfer between surfaces can be described sufficiently accurately without requiring a knowledge of quantum mechanics and electromagnetic theory. Different theories using the above assumptions are discussed briefly below.

Fig. 22. Configuration-shape-factor Notation

Configuration factor is a quantity that represents the fraction of radiant energy leaving one surface and arriving at another surface. It accounts for the geometrical variations between the interacting surfaces that directly affect the amount of radiant-energy transfer. In particular, for diffusely distributed radiant energy leaving surface one and incident upon surface two (as shown in Fig. 22), the configuration factor is given

$$F_{12} = \iint dA_1 dA_2 \frac{\cos \theta_1 \cos \theta_2}{\pi r^2} . \quad (42)$$

This expression is valid under the assumption that the amount of radiant energy leaving either surfaces is uniform over its respective area of consideration. However, this constraint can be relaxed to apply to surfaces with nonuniform radiant-energy flux by considering infinitesimal surface elements.

1. Integral-equation Approach

Consider an enclosure with continuous variation of radiosity, surface temperature, and surface heat flux (as shown in Fig. 23), where radiosity is defined as

$$J(\xi) = \epsilon(\xi) \sigma T^4(\xi) + \rho(\xi) G(\xi), \quad (43)$$

Fig. 23
Enclosure with Continuous Variation
of Radiosity, Surface Temperature,
and Surface Heat Flux

where $G(\xi)$ denotes the incident radiant heat flux upon position ξ , and n is assumed to be unity. The local heat-transfer rate at χ in the enclosure is the difference between the emitted radiant flux and the amount of incident radiation absorbed. That is,

$$q(\chi) = \epsilon(\chi)\sigma T^4(\chi) - \alpha(\chi)G(\chi). \quad (44)$$

With Eq. 43, the radiosity can be reexpressed as

$$J(\xi) = \epsilon(\xi)\sigma T^4(\xi) - \frac{\rho(\xi)}{\alpha(\xi)} [q(\xi) - \epsilon(\xi)\sigma T^4(\xi)]. \quad (45)$$

If the enclosure is opaque, then,

$$\epsilon(\xi) + \rho(\xi) = 1 \quad (46)$$

and

$$J(\xi) = \sigma T^4(\xi) - \frac{1 - \epsilon(\xi)}{\epsilon(\xi)} q(\xi). \quad (47)$$

Since all the radiation incident upon an element of an enclosure comes from other elements that make up the enclosure, the incident radiation heat flux can be written as

$$G(x) = \int J(\xi) dF_{dx-d\xi}. \quad (48)$$

When Eq. 43 is recast into a different form, the radiosity at x becomes

$$J(x) = \epsilon(x)\sigma T^4(x) + 1 - \epsilon(x) \int J(\xi) dF_{dx-d\xi} \quad (49)$$

and combining Eqs. 47 and 49, we obtain

$$\frac{q(x)}{\epsilon(x)} = \sigma T^4(x) - \int \sigma T^4(\xi) dF_{dx-d\xi} + \int [1 - \epsilon(\xi)] \frac{q(\xi)}{\epsilon(\xi)} dF_{dx-d\xi}. \quad (50)$$

If the temperature is prescribed, the heat-flux distribution can be calculated from Eq. 50. Conversely, if the heat flux is provided, the temperature distribution can be determined. This represents two different heat-transfer processes in which the gray-body energy exchange is characterized by the first two terms of the right side of the equation. The last term denotes the energy exchange resulting from geometrical reflections of different elements within the enclosure.

2. Finite-difference Approximation

The integral equation given in Eq. 50 is generally difficult to solve, sometimes even for simple geometries. For this reason a finite-difference approximation is formulated. Instead of examining infinitesimal

elements, we choose a finite number of small subareas such that the radiosity over each subarea is assumed to be constant. The radiosity for subarea i is therefore

$$J_i = \epsilon_i \sigma T_i^4 + \rho_i G_i. \quad (51)$$

From the energy balance between subareas within an enclosure, the total amount of radiant energy arriving at subarea i is

$$G_i = \sum_{j=1}^N J_j F_{i-j}, \quad (52)$$

where N is the number of subareas within the enclosure. The local heat-transfer rate at subarea i is given by

$$q_i = \epsilon_i \sigma T_i^4 - \alpha_i G_i. \quad (53)$$

Combining Eqs. 51 and 52, we obtain

$$J_i = \epsilon_i \sigma T_i^4 + \frac{\rho_i}{\alpha_i} (q_i - \epsilon_i \sigma T_i^4). \quad (54)$$

If the enclosure is opaque, implying that the sum of the emittance and reflectance is unity, then,

$$\epsilon_i + \rho_i = 1 \quad (55)$$

and

$$J_i = \sigma T_i^4 - \frac{1 - \epsilon_i}{\epsilon_i} q_i. \quad (56)$$

Using Eq. 52, we can further eliminate the radiosity to obtain

$$\frac{q_i}{\epsilon_i} = \sigma T_i^4 - \sigma \sum_{j=1}^N T_j^4 F_{i-j} + \sum_{j=1}^N (1 - \epsilon_j) F_{i-j} \frac{q_j}{\epsilon_j}. \quad (57)$$

With the identity

$$\sum_{j=1}^N F_{i-j} = 1,$$

Eq. 57 can be rewritten as

$$\frac{q_i}{\epsilon_i} = \sum_{j=1}^N (\sigma T_i^4 - \sigma T_j^4) F_{i-j} + \sum_{j=1}^N (1 - \epsilon_j) F_{i-j} \frac{q_j}{\epsilon_j}. \quad (58)$$

The first term clearly shows the gray-body energy exchange; the second term describes the geometrical reflection between surfaces. Equation 58, in essence, is the finite-difference representation of Eq. 50, for which a numerical solution can be obtained, provided that a sufficiently large number of mesh points are used. Various similar approximation methods, namely, Hottel's method, Gebhart's method, and the Network method, can all be derived from these finite-difference approximation equations.

3. Network Method

It is recalled from the definitions of radiosity (Eq. 51) and local heat-transfer rate (Eq. 53) that, if the incident radiant heat flux is eliminated from these equations, the result is

$$q_i = \frac{\epsilon_i}{1 - \epsilon_i} (\sigma T_i^4 - J_i), \quad (59)$$

Fig. 24. Equivalent Networks for Energy Exchange between Surfaces

where analogy can be drawn between the above equation and the Ohm's Law expression. The quantities $(1 - \epsilon_i)/\epsilon_i$, q_i , and $(\sigma T_i^4 - J_i)$ can be interpreted as the "resistance" to radiant heat flow, the "current" of heat transfer, and the "potential difference" between two imaginary surfaces, respectively. Equation 59 and its electronic-circuit equivalent are depicted in Fig. 24a. When the emittance ϵ_i approaches zero, regardless of the surface temperature, there is no heat transfer away from surface i . As ϵ_i becomes unity, the thermal resistance, $(1 - \epsilon_i)/\epsilon_i$, vanishes and consequently, Eq. 59 reduces to the simple fact that the radiosity is the black-body emissive power.

Using Eq. 52 and the local heat exchange between surface i and its surroundings given by the difference between radiosity and irradiation,

$$q_i = J_i - G_i, \quad (60)$$

we can write the heat-transfer rate as a function of radiosities from different surfaces and their respective configuration factors.

$$q_i = J_i - \sum_{j=1}^n J_j F_{ij} = \sum_{j=1}^n (J_i - J_j) F_{ij}. \quad (61)$$

The identity $\sum_{j=1}^n F_{ij} = 1$

has been used to arrive at Eq. 61. If the radiosities J_i and J_j are interpreted as thermal potential nodes, Eq. 61 becomes a summation of these thermal potential differences between surfaces within the enclosure where the resistance between nodes is dependent upon the surface areas and their respective configuration factors. This is also illustrated in Fig. 24b.

Authors such as Sparrow (1965) and Kreith (1976) extended this concept and arrived at a network model of radiant energy exchange as depicted in Fig. 24a in which each black-body emitter is denoted by a battery acting as a driving force. The black bodies are typically connected together through two resistors, each representing the gray-body approximation and the geometrical resistance in energy transfer. This electric-network analogy allows the application of linear rules of electrical circuit, namely, Kirchhoff's Law, to many body-energy exchanges in which simplicity and linearity of the equations are preserved.

This basic approach is commonly used by others in formulating their solutions, such as Tien and Cunnington (1969), Fletcher (1970). However, note that the network model given in Fig. 24 is not the result of direct derivation from fundamental equations. The two equations that form the basis of the network model and are deduced from first principles are Eqs. 59 and 61. Each of them describes precisely only the energy-exchange relationship between itself and its immediate surroundings, as depicted in Fig. 24. When the system includes only two surfaces, attempts in connecting the two surfaces results in a network shown in Fig. 24d.

Further efforts in connecting multiple surfaces together, e.g., Fig. 24c, exemplified an inherent assumption. Energy transfer from one black surface to another follows the most direct path, i.e., the connection along which there are the least number of resistors. Since, for the two-surface system, the shortest path between the surfaces happens to be also the only path, the above assumption seems to be superfluous. But as one examines a three-body system (Fig. 24c), energy leaving surface 1 traverses to surfaces 2 and 3 via path 12 and 13, and if the three surfaces are assumed to be of a black-body nature, radiant energy arriving at surface 3 by path 23 is totally independent of surface 1. This implies that, once radiant energy impinges upon any of these surfaces, its earlier history of where the energy originates becomes irrelevant to any subsequent energy-transfer calculation. The energy exchange between surfaces 1 and 2 is

$$q_{12} = A_1 F_{12} (E_{b1} - E_{b2}), \quad (62)$$

where $1/A_1 F_{12}$ is the thermal resistance. This differs from results obtained by evaluating the effective thermal resistance of the three resistors based on the theory of electric circuits, namely,

$$\frac{1}{R} = \frac{1}{\frac{1}{A_2 F_{23}} + \frac{1}{A_1 F_{13}}} + A_1 F_{12}. \quad (63)$$

For a two-body gray-surface system, the network representation that is a direct reduction from the basic equations accurately models the energy exchange between the surfaces. However, similar network representation for three-body interaction poses a few difficulties if caution is not exercised in reducing solution from the model. Figure 24c illustrates the fact that energy exchanges between each pair of radiosity nodes, viz., J_1 and J_2 , J_3 and J_2 , and J_1 and J_3 , satisfy the equation

$$q_{12} = \frac{1}{R_{12}} (J_1 - J_2), \quad (64)$$

where q_{12} is the energy exchange between the radiosity nodes J_1 and J_2 and F_{12} is the configuration-factor resistance between those two surfaces. But in order to ascertain the net heat-transfer rate from surface 1 to surface 2, the network-resistance equivalent between the two points of interest yields

$$F_{ij} = \frac{1 - \epsilon_1}{A_1 \epsilon_1} + \frac{1 - \epsilon_2}{A_2 \epsilon_2} + \frac{F_{12} F_{13} + F_{23}}{F_{12} F_{32} + F_{13}}, \quad (65)$$

with F_{ij} , $i, j = 1, 2, 3$, denoting the configuration-factor resistance between surfaces i and j . The network equivalent model further predicts that the radiant-energy exchange between surfaces 1 and 2 is inversely proportional to the effective thermal resistance (Eq. 64) multiplied by the difference in thermal potential, given by the black emissive power between the said surfaces. That is,

$$q = \frac{1}{R} (E_{b1} - E_{b2}). \quad (66)$$

As the emittance of surface 3, ϵ_3 , approaches zero, implying that it is a perfect reflector ($\rho_3 = 1$), there is a maximum amount of energy transfer between surfaces 1 and 2. Conversely, when surface 3 is a black body, energy transfer is minimum. Careful examination of Eq. 64 shows that the effective thermal resistance is only dependent upon the configuration factors, emittance, and areas of respective surfaces. Consequently, changes in the radiative properties of surface 3 have little effect on the overall energy exchange between the other two surfaces. Similar difficulty also occurs when the model is extended to many-body exchanges.

4. Gebhart Method

The basic assumptions of this method entails the gray-body approximations of interaction and diffuse surfaces. Gebhart, who introduced

Fig. 25. Enclosure to Illustrate the Gebhart Method

this method, denoted the fraction of radiant energy absorbed by surface j to that which is emitted by surface i as G_{ij} . This fraction of radiant energy includes the direct radiant energy transfer and any other transfer through multiple reflections within the enclosure from surfaces i to j . For an enclosure with n surface elements as depicted in Fig. 25, the energy balance between surface i and surface j can be summarized as

$$G_{ij} = \epsilon_j F_{ij} + \sum_{k=1}^n \rho_k F_{ik} G_{kj}, \quad (67)$$

where F_{ij} denotes the fraction of the amount of total direct energy emitted by surface element i that is absorbed by element j and is the definition of

configuration factor. The summation represents the exchanges through reflections. For example, when $k = 1$, the term $\rho_1 F_{i1}$ describes the fraction of total direct energy from surface i that is reflected from element 1. However, this quantity, when multiplied by G_{1j} , signifies the fraction of reflected radiation from surface element i reflected by element 1 that would be absorbed by element j . This is a direct consequence of the assumption that the surfaces are diffuse.

Equation 67 can be rewritten without the summation notation as

$$\left. \begin{aligned} G_{1j} &= \epsilon_j F_{1j} + \rho_1 F_{11} G_{1j} + \rho_2 F_{12} G_{2j} + \dots + \rho_n F_{1n} G_{nj}; \\ G_{2j} &= \epsilon_j F_{2j} + \rho_2 F_{21} G_{1j} + \rho_2 F_{22} G_{2j} + \dots + \rho_n F_{2n} G_{nj}; \\ G_{nj} &= \epsilon_j F_{nj} + \rho_n F_{n1} G_{1j} + \rho_2 F_{n2} G_{2j} + \dots + \rho_n F_{nn} G_{nj}; \end{aligned} \right\} \quad (68)$$

where it can be easily recast in a matrix equation as

$$\begin{pmatrix} \rho_1 F_{11}^{-1} & \rho_2 F_{12} & \rho_n F_{1n} \\ \rho_1 F_{21} & \rho_2 F_{22}^{-1} & \rho_n F_{2n} \\ \rho_1 F_{n1} & \rho_2 F_{n2} & \rho_n F_{nn}^{-1} \end{pmatrix} \begin{pmatrix} G_{1j} \\ G_{2j} \\ G_{nj} \end{pmatrix} = \epsilon_j \begin{pmatrix} F_{1j} \\ F_{2j} \\ F_{nj} \end{pmatrix}, \quad (69)$$

or simply

$$\underline{\underline{AG}} = \underline{\underline{F}}. \quad (70)$$

The solution to Eq. 70 can be realized by multiplying each side with the inverse of A, denoted by $\underline{\underline{A}}^{-1}$.

$$\underline{\underline{A}}^{-1} \underline{\underline{AG}} = \underline{\underline{A}}^{-1} \underline{\underline{F}}. \quad (71)$$

Using the identity of

$$\underline{\underline{A}}^{-1} \underline{\underline{A}} = \underline{\underline{I}}, \quad (72)$$

where $\underline{\underline{I}}$ is defined as the identity matrix satisfying

$$(I)_{ij} = \delta_{ij} = \begin{cases} 0 & i \neq j \\ 1 & i = j, \end{cases} \quad (73)$$

Equation 71 becomes

$$\underline{\underline{G}} = \underline{\underline{A}}^{-1} \underline{\underline{F}}. \quad (74)$$

Therefore, provided the inverse to the matrix A exists, all the components of the vector G can be evaluated. Since the inverse of a matrix is defined as the quotient of the transposed cofactor matrix of A and the determinant of A

$$\underline{\underline{A}}^{-1} = \frac{(\underline{\underline{C}})^T}{|\underline{\underline{A}}|}, \quad (75)$$

Eq. 75 stipulates that $\underline{\underline{A}}^{-1}$ is nonsingular if the determinant of A is nonzero. The existence of A and the uniqueness of its solution is assured if and only if $\underline{\underline{A}}\underline{\underline{x}} = 0$ implies that $\underline{\underline{x}} = 0$, as specified by Stakgold (1967). Physically, this can be easily demonstrated by allowing ϵ_j to go to zero, thereby creating a perfectly reflective surface j, which would render all G_{nj} zero. See Appendix A.

Consequently, the net energy transfer by a surface element j can be written as

$$q_{\text{net } j} = \sum_{\ell=1}^n G_{\ell j} \epsilon_{\ell}^A E_{b\ell} - \epsilon_j^A E_{bj}. \quad (76)$$

The first term with the summation sign incorporates all the energy transfer to element j by absorption through either direct exchange or multiple reflections. The second term expresses the emission loss from the j surface. A reciprocity relation, proved by Gebhart, reduces Eq. 76 to

$$G_{ij}\epsilon_i A_i = G_{ji}\epsilon_j A_j \quad (77)$$

and

$$q_{\text{net } j} = \epsilon_j A_j \left(\sum_{\ell=1}^n G_{j\ell} - E_{bj} \right). \quad (78)$$

5. Hottel's Method

This method was proposed by Hottel, and described by McAdams (1954), where the author introduced the following definition for the energy transfer from surfaces 1 and 2:

$$q_{1-2} = F_{12} A_1 E_{b1}, \quad (79)$$

where F_{12} is the fraction of energy emitted by surface 1 and absorbed by surface 2 assuming black-body exchanges. It is a function of the geometrical orientation of the two surfaces and of the emittance of the respective surface. The net energy exchange between surface 1 and the enclosure is given by

$$q_{\text{net } 1} = \sum_{i=1}^n q_{i-1}, \quad (80)$$

where q_{i-1} is the net flux between surfaces i and 1 and can be expressed as

$$q_{i-1} = q_{1-i} - q_{i-1}. \quad (81)$$

Within an enclosure of n surfaces, except for surface 1, all surfaces are assumed to be at 0 K. Consequently, any energy that arrives at a surface ℓ will have to originate from surface 1 and the energy that is absorbed can then be written as

$$q_{\text{net } \ell} = q_{1-\ell} = (G_{\ell} - J_{\ell}) A_{\ell}, \quad (82)$$

where G_{ℓ} and J_{ℓ} are the irradiation and the radiosity of ℓ , respectively, and A_{ℓ} is the surface area of surface element ℓ . The amount of reflected radiation at surface ℓ will be exactly equal to the radiosity of the surface:

$$\rho_{\ell} G_{\ell} = J_{\ell}. \quad (83)$$

On the other hand, the irradiation of surface element ℓ can also be written as

$$G_{\ell} A_{\ell} = (\epsilon_1 E_{b1} + J_1) A_1 F_{1\ell} + J_2 A_2 F_{2\ell} + \dots + J_n A_n F_{n\ell}, \quad (84)$$

in which the radiosity is defined in a slightly different manner than previously noted. Radiosity of the i th surface, J_i , is defined as the amount of energy that leaves surface i exclusively due to reflections from surface 1; for a black-body enclosure, the above definition reduces to zero. It is also for this reason that the radiant energy that arrives at surface ℓ from surface 1 includes both the radiosity J_1 term and the gray-body emissive power $\epsilon_1 E_{b1}$. Based on Eqs. 84 and 83, a matrix equation describing the radiant-energy interaction results:

$$\begin{pmatrix} A_1 F_{11} - A_1 / \rho_1 & A_2 F_{21} & A_3 F_{31} & \dots & A_n F_{n1} \\ \vdots & \vdots & \vdots & \vdots & \vdots \\ A_1 F_{12} & A_2 F_{22} - A_2 / \rho_2 & A_3 F_{32} & \dots & A_n F_{n2} \\ \vdots & \vdots & \vdots & \vdots & \vdots \\ A_1 F_{1n} & \dots & A_n F_{nn} - A_n / \rho_n \end{pmatrix} \begin{pmatrix} J_1 \\ \vdots \\ J_2 \\ \vdots \\ J_n \end{pmatrix} = - \begin{pmatrix} A_1 F_{11} \epsilon_1 E_{b1} \\ \vdots \\ A_1 F_{12} \epsilon_1 E_{b1} \\ \vdots \\ A_1 F_{1n} \epsilon_1 E_{b1} \end{pmatrix}. \quad (85)$$

It is again obvious that if surface 1 assumes a 0 K temperature, Eq. 85 would become a homogeneous matrix equation to which only the trivial solution exists (see Appendix A).

Based on Eqs. 82 and 83, the radiant-energy transfer from surface 1 to surface ℓ is expressed as

$$q_{1-\ell} = J_{\ell} A_{\ell} \left(\frac{1}{\rho_{\ell}} - 1 \right). \quad (86)$$

By comparing Eqs. 86 and 79, we see that

$$F_{i\ell} = \frac{\epsilon_{\ell}}{\rho_{\ell}} \frac{J_{\ell}}{E_{b1}} \frac{A_{\ell}}{A_1}. \quad (87)$$

It can be further demonstrated that the reciprocity identity is valid, so that

$$F_{ij} A_j = F_{ji} A_i. \quad (88)$$

Finally, from Eq. 79 it is expected that F_{12} would have to converge to the well-known gray-body approximation, viz., $q_{1-\text{enclosure}} = \epsilon_1 A_1 E_{b1}$. Since the summation of energy flow from surface 1 is equal to the gray-body approximation

$$\sum_{i=1}^n q_{1-i} = \epsilon_1 A_1 E_{b1}, \quad (89)$$

it follows that

$$\sum_{j=1}^n F_{1j} = \epsilon. \quad (90)$$

6. Modified Gebhart Method

Recall that Gebhart first defined G_{ij} as the fraction of total radiant energy emitted by surface i that is absorbed by surface j . Consequently, multiple reflections within the enclosure are also meticulously included. By the assumption that the energy from the i th surface is diffusely reflected, the fraction absorbed by the j th surface is G_{ij} . Based on this fact, the following equations result:

$$\left. \begin{aligned} G_{1j} &= \epsilon_j F_{1j} + \rho_1 F_{11} G_{1j} + \rho_2 F_{12} G_{2j} + \dots + \rho_n F_{1n} G_{nj}; \\ G_{2j} &= \epsilon_j F_{2j} + \rho_1 F_{21} G_{1j} + \rho_2 F_{22} G_{2j} + \dots + \rho_n F_{2n} G_{nj}; \\ &\vdots \\ G_{nj} &= \epsilon_j F_{nj} + \rho_1 F_{n1} G_{1j} + \rho_2 F_{n2} G_{2j} + \dots + \rho_n F_{nn} G_{nj}. \end{aligned} \right\} \quad (91)$$

EACH FACE 1 SQ. FT.

SURFACES:

- 1: T_1
- 2: T_2
- 3: insulated

Fig. 26

A Cubical Enclosure

These equations essentially describe the energy balance between a surface and its environment. For example, G_{1j} , the amount of radiant energy emitted by surface 1 and absorbed by surface j , is equal to the summation of the fraction j and the fractions that are reflected from each surface.

A cubical enclosure is depicted in Fig. 26 in which the base of the cube is maintained at 1500°F (816°C), one side of the enclosure is heated to 800°F (427°C) and the remaining surfaces are well insulated. It is further assumed that each surface of this cubical enclosure is of unit area and the emittance is 0.7. Insulated surfaces would imply that the amounts of energy incident upon and radiated from such surfaces are equal. Therefore, emittances for these surfaces are arbitrarily chosen to be zero. The configuration factors between the components

within the enclosure are

$$\left. \begin{aligned} F_{11} &= 0 & F_{12} &= 0.2 & F_{13} &= 0.8 \\ F_{21} &= 0.2 & F_{22} &= 0 & F_{23} &= 0.8 \\ F_{31} &= 0.2 & F_{32} &= 0.2 & F_{33} &= 0.6 \end{aligned} \right\}, \quad (92)$$

where surfaces 1, 2, and 3 are labeled according to Fig. 26. The resultant matrix for Eq. 91 can be written as

$$\begin{pmatrix} -1 & 0.06 & 0.8 \\ 0.06 & -1 & 0.8 \\ 0.06 & 0.06 & 0.8 \end{pmatrix}. \quad (93)$$

It can be seen from Eq. 69 that G_{13} vanishes due to the fact that surface 3 is insulated, and ϵ_3 is chosen to be zero. Similarly, based on this assumption and on the definition of G_{ij} , G_{31} or G_{32} is also expected to vanish, for the amount of emitted energy leaving surface 3 and arriving at either surface 1 or 2, which is absorbed, is zero. This is not the case, as is demonstrated in the example. However, this does not imply that $q_{\text{net}3}$, the net energy leaving surface 3, is in error, for it indeed vanishes. So, to circumvent the aforementioned difficulty, G_{ij} can be redefined as follows: G_{ij} represents the fraction of total black-body radiant energy emitted from surface i that is absorbed by surface j . Consequently, Eq. 91 can be rewritten as

$$\left. \begin{aligned} G_{1j} &= \epsilon_1 \epsilon_j F_{1j} + \frac{\rho_1 \epsilon_1 F_{11} G_{1j}}{\epsilon_1} + \frac{\rho_2 \epsilon_1 F_{12} G_{2j}}{\epsilon_2} + \dots + \frac{\rho_n \epsilon_1 F_{1n} G_{nj}}{\epsilon_n}; \\ G_{2j} &= \epsilon_2 \epsilon_j F_{2j} + \frac{\rho_1 \epsilon_2 F_{21} G_{1j}}{\epsilon_1} + \frac{\rho_2 \epsilon_2 F_{22} G_{2j}}{\epsilon_2} + \dots + \frac{\rho_n \epsilon_2 F_{2n} G_{nj}}{\epsilon_n}; \\ \vdots & \\ G_{nj} &= \epsilon_n \epsilon_j F_{nj} + \frac{\rho_1 \epsilon_n F_{n1} G_{1j}}{\epsilon_1} + \frac{\rho_2 \epsilon_n F_{n2} G_{2j}}{\epsilon_2} + \dots + \frac{\rho_n \epsilon_n F_{nn} G_{nj}}{\epsilon_n}. \end{aligned} \right\} \quad (94)$$

These equations describe G_{ij} as the sum of the fraction of black-body radiation from i arriving directly at j that is absorbed and the multiple-reflection terms. Thus the matrix equation becomes

$$\begin{pmatrix} \frac{\rho_1 \epsilon_1 F_{11}^{-1}}{\epsilon_1} & \frac{\rho_2 \epsilon_1 F_{12}}{\epsilon_2} & \frac{\rho_3 \epsilon_1 F_{13}}{\epsilon_3} & \dots & \frac{\rho_n \epsilon_1 F_{1n}}{\epsilon_n} \\ \frac{\epsilon_2 \rho_1 F_{21}}{\epsilon_1} & \frac{\rho_2 \epsilon_2 F_{22}^{-1}}{\epsilon_2} & \frac{\rho_3 \epsilon_2 F_{23}}{\epsilon_3} & \dots & \frac{\rho_n \epsilon_2 F_{2n}}{\epsilon_n} \\ \frac{\rho_1 \epsilon_n F_{n1}}{\epsilon_1} & \frac{\rho_2 \epsilon_n F_{n2}}{\epsilon_2} & \dots & \dots & \frac{\rho_n \epsilon_n F_{nn}^{-1}}{\epsilon_n} \end{pmatrix} \begin{pmatrix} G_{1j} \\ G_{2j} \\ G_{nj} \end{pmatrix} = \begin{pmatrix} -\epsilon_1 F_{1j} \\ -\epsilon_2 F_{2j} \\ -\epsilon_n F_{nj} \end{pmatrix} \epsilon_j.$$

(95)

For $n = 3$, the above equation becomes

$$\begin{pmatrix} -1 & 0.06 & 560 \\ 0.06 & -1 & 560 \\ 8.57 \times 10^{-5} & 8.5 \times 10^{-5} & -0.4 \end{pmatrix} \begin{pmatrix} G_{1j} \\ G_{2j} \\ G_{3j} \end{pmatrix} = -\epsilon_j \begin{pmatrix} \epsilon_1 F_{1j} \\ \epsilon_2 F_{2j} \\ \epsilon_3 F_{3j} \end{pmatrix}. \quad (96)$$

For $j = 2$, the inverse matrix is

$$\frac{1}{0.2968} \begin{pmatrix} 0.352 & 0.072 & 593.6 \\ 0.072 & 0.352 & 593.6 \\ 9.08 \times 10^{-5} & 9.08 \times 10^{-5} & 0.9904 \end{pmatrix} \begin{pmatrix} G_{12} \\ G_{22} \\ G_{23} \end{pmatrix} = -0.7 \begin{pmatrix} 0.14 \\ 0 \\ 2 \times 10^{-4} \end{pmatrix}, \quad (97)$$

which shows that only G_{12} is nonvanishing.

The exchange between surfaces 1 and 2 is given by

$$F_{12} = G_{12} A_1 E_{b1} - G_{21} A_2 E_{b2}. \quad (98)$$

The net energy gain or loss of a surface is

$$F_{\text{net}_1} = \sum_{j=1}^m G_{j1} A_j E_{bj} - \epsilon_1 A_1 E_{b1}. \quad (99)$$

It can be seen from Eq. 98 that if the black-body emissive powers of surfaces 1 and 2 are the same, then the heat-exchange rate between the two surfaces will be zero. Since $G_{12} A_1$ and $G_{21} A_2$ are both independent of temperature, this demonstrates the reciprocity relation

$$G_{12} A_1 = G_{21} A_2. \quad (100)$$

Furthermore, Gebhart has already established that the sum of the fractions from one surface to all other surfaces is always unity. Therefore, it follows that

$$\sum_{j=1}^m \frac{1}{\epsilon_i} G_{ij} = 1. \quad (101)$$

An example of the Gebhart method and the modified version along with a proof of identity is given in Appendix B.

C. Contact Resistance

In general, two phenomena in conduction can be ascribed to the study of contact resistance: constriction thermal resistance and surface interface resistance. Both of these areas have received considerable attention, as indicated in Chapter II. Their importance with regard to engineering heat-transfer calculations cannot be overemphasized. Despite these facts and despite extensive studies in search of a comprehensive, analytical model describing the phenomena, available models all require rather laborious computation and detailed information on statistical surface conditions. Of the three types of insulation of interest, only the honeycomb and the screen-type insulations demand attention in this area. The effects of contact resistance on the supports of parallel-plate insulation are not addressed, since the plates are assumed to be separated by a perfect insulator.

Yovanovich and Tuarze (1969) reported an experimental investigation of thermal-contact resistance at soldered joints in which a survey of leading heat-transfer texts led the authors to believe that a soldered joint exhibited negligible thermal resistance. However, it was demonstrated that experimental measured data on soldered joints showed a ten- to hundredfold increase in thermal resistance over the theoretical predicted value. The method through which the solder joint is prepared also greatly affects the magnitude of the measured values. Figure 27 shows the reported results, where the curves clearly indicate an independence of load pressure. The top curve represents a joint with no oxide layer, but yet containing dispersed cavities throughout the solder. The average thickness of the solder used in these experiments is 15 μm . A complete heat-transfer model describing the honeycomb insulation will have to include such a contribution.

Fig. 27
Contact Resistance vs Load
Pressure for Soldered Joint

The study of thermal-contact resistance of screen materials has received relatively little attention. Smuda and Gyorog (1969) and Gyorog (1970) presented experimental results on the study of various types of interstitial materials. Stainless steel screens were investigated, and the thermal-contact resistance as a function of load is depicted in Fig. 2b. Within the same figure, data of an aluminum screen are also shown for comparison. Since there are thousands of contacts per screen layer, the error involved could be larger than given by individual contacts, but nonetheless, it provides a reasonable estimate of the screen composite.

Fig. 28

Thermal Resistance per Contact vs Load. Conversion factors: 1 Btu/h·°F = 0.528 W/K; 1 lb = 0.138 N.

D. Comprehensive Models

Based on the theories proposed or developed in previous sections, it is possible to combine these theories so that comprehensive models can be formulated to describe the three different kinds of thermal insulation. The total amount of heat transfer through the insulation is assumed to be the sum of the individual heat-transfer components.

$$Q_{\text{Total}} = Q_{\text{conv}} + Q_{\text{rad}} + Q_{\text{cond.}} \quad (102)$$

Note that the gaseous-conduction contribution to heat transfer is included in the Q_{conv} term. The computer code that calculates the thermal conductivity by an iterative method is listed in Appendix F.

1. Multiplates

Whether it be 12 or 24 plates, the thermal-radiation heat transfer satisfies the Stefan-Boltzmann equation. The modified Gebhart method can be used to adequately describe the energy exchanges between surfaces. Due to the simplicity of the geometry, for large plates where edge effects can be ignored, the network model can also be a very precise model. Thermal convection between parallel plates can be of significant importance if the product of the dimension between individual plates and the temperature difference they sustain exceeds a certain value. The Rayleigh number is plotted against $\Delta T L^3$ in Fig. 18. By carefully choosing the constant, one can maintain thermal stability where only conduction will be the major concern.

This model is applied to the experimental results, including both the evacuated and the nonevacuated data. Details are presented in Chapter VI. For 12 multiplates, convection is not expected to play a significant role in overall heat transfer, just as predicted from thermal-stability considerations.

2. Honeycombs

Based on the model presented in Sec. III.A.1 for cellular geometries, natural-convection onset conditions can be easily evaluated. For the type of honeycomb material used in the experiments (aspect ratio 1.33), the theory and experimental results ensure a conduction-dominated heat-transfer regime. To predict the experimental data collected in this study, thermal convection is neglected. The manufacturer of the brazed honeycomb material used a vacuum-brazing technique where the chamber in which the brazing was performed was evacuated. A possibility of 30% of brazed defects was reported. Therefore, even with 30% of the cells gas-filled, the heat-transfer contribution is minimal.

The metallic conduction through the honeycomb-cell wall can be evaluated by assuming a modified cross-sectional area that satisfies the Fourier heat-conduction equation. The modified area is essentially the total of all the cross-sectional areas of the cell walls after discounting the percentage of bad contacts.

Using the modified Gebhart method, one is able to express the net heat loss or heat gain of a surface (Eq. 99). The honeycomb-cell wall is assumed to be insulated; this implies that the irradiation is equal to the radiosity. This assumption is well justified by the fact that each surrounding cell is exposed to similar conditions as the one under consideration. If there is a net heat flow from one cell to another, it would mean that there is a temperature gradient across the cell; this violates the assumption that the heat source is from below. This calculation can easily be extended to include any number of honeycomb cells by simply modifying the area involved. Results are compared with experimental data in Chapter VI, where good agreement is evident.

3. Screens

Similar to the multiplates, both the 12 and the 24 layer screens can be treated by the same model. Section III.A.2 discussed the topic of the onset of natural convection within a porous medium. Based on the formulation and the application of Eq. 38, the thermal stability inside layers of screens can be evaluated. As for the screen materials selected for the experiment, the Rayleigh number falls in the transition regime. However, even though the Rayleigh number is in the neighborhood of the critical value, consideration of only the thermal-conduction aspect will suffice, since the Nusselt-number deviation from unity is still not substantial.

Thermal-radiation heat transfer is modeled by means of the network approach. The screen layers are characterized by a transmissivity τ , which is defined as the fraction of radiation that is transmitted through the medium or as the fraction of open area, as discussed by Tien and Cunningham (1973). The derivation is given in Appendix E, in which two cases are examined: $\epsilon + \rho + r = 1$ and $\epsilon + r = 1$. The results are given in Figs. E.3-E.5. Comparisons are made with data presented by Kreith (1976) and Tien and Cunningham. Based on information from Fig. E.5, results of thermal-radiation heat transfer of screen-type materials can be evaluated, as presented by Shiu and Jones (1978).

The application of these methods to describe the experimental results is presented in Chapter VI.

The three models proposed exhibit very good agreement with measured data. Therefore they can be used to describe similar insulations with different parameters, e.g., τ , ϵ , and ρ .

E. Influence of Sodium and Sodium Oxide on Models

1. Sodium

Studies of the heat-transfer effects due to the presence of sodium deposits are rather limited. Interest is primarily confined to different

Fig. 29. Thermal Conductivity of Sodium vs Temperature. Conversion factors:
 1 Btu/h-ft. °F = 1.73 W/m-K;
 $C = (^{\circ}F - 32)/1.8$.

thermal-hydraulic aspects of the liquid state of sodium and its application in breeder-reactor designs. Sodium is an exceptionally good heat conductor (as shown in Fig. 29), about 100 times better than water. It melts at about 208°F (98°C), and its thermal conductivity maintains almost a linear relation with increasing temperature. Relatively pure sodium exhibits a very low emittance and thus appears to be highly reflective. Without consideration of the nature of the sodium deposits, the aforementioned thermal properties of sodium will not suffice in providing a realistic estimation of the thermal impact of sodium de-

posits on surfaces. Thermal convection within the deposit layer is neglected, due to the reduced thickness of the layer and the unusually good thermal conductivity.

The factors that affect the formation of sodium deposits on a surface can be identified as (1) sodium-vapor pressure, (2) thermal gradients and temperature, (3) mass gradients, and (4) surface condition and wettability. Figure 16 depicts the data obtained by Atomics International on the wetting time versus temperature, with the assumption that the surface is immersed into the liquid sodium. The condition of the surface is further taken to be clean and to have a nominal thin film of oxide due to its contact with air. Figure 11 depicts the relationship between sodium vapor and temperature. Since the condensation can be either of homogeneous or heterogeneous nucleation in nature, strong temperature dependence of the vapor pressure will ensure rapid and abundant condensation on cooler surfaces. However, depending on the conditions of the surface and the environment when liquid sodium condenses on a metallic surface, it will eventually either wet the surface or remain to be a sphere on the surface. Heat-transfer characteristics resulting from either of these situations will differ greatly. In Fig. 30, the aforementioned situations are depicted in which the wetting angle is defined as the angle precluded by the surface and the tangent of the liquid at the point of contact. It can be seen that small wetting angles imply nonwetting, and vice versa. Besides the strong dependence of the wetting time on temperature, wetting further

depends on the material of the surface; the wetting fluid, which in the case of our interest is sodium, and its oxygen content within the fluid; and the surface conditions. Figure 31 shows the wetting angle of stainless steel material under various temperatures. Surfaces were prepared with silicon carbide paper with a lubricant and then polished with 14-grit diamond dust. At about 300°C wetting occurs, and above 400°C significant wetting is reported, as described by Hodkin and Nicholas (1976).

Fig. 30. Definition of Wetting Angles

Fig. 31. Wetting Angles of Different Materials vs Temperature

Figure 32 shows the effects on wetting behavior with different surface preparation using different abrasives. The change of surface roughness is no greater than 0.279 μm when using abrasives ranging from 0.25- μm diamond dust to 180 silicon carbide grit. At low temperatures, the wetting angle is larger for rough surfaces and comparatively smaller as temperature increases. Figure 33 depicts the effects of oxidation on wetting resulting from the increasing thickness of the oxide film on the material. The material tested is preoxidized from a few minutes up to 10 h at 700°C. An elevated temperature is needed before a similar type of wetting angle is observed. However, for other similar material tests, the change is minimal. Figure 34 shows the wetting behavior as a function of time at different temperatures of a preoxidized sample. Finally, Fig. 35 illustrates the effects of wetting due to the presence of different amounts of oxygen in the sodium. Therefore, in order to accurately describe the heat-transfer aspects of a component, these attributing causes must be considered.

Fig. 32

Wetting Angles of Stainless Steel with
Different Surface Finish vs Temperature

Fig. 33
Effects of Oxidation on Wetting Angles
of Stainless Steel vs Temperature

Fig. 34

Effects of Preoxidized Stainless Steel
on Wetting Angles vs Time

Fig. 35

Effects of Oxygen Content in Sodium on
Wetting Angles at Different Temperatures

Due to the complexity of the situation and the interdependence of various parameters, exact analytical solution is supplanted by two cases in which the overall heat-transfer characteristics are bonded and adequately described. The first case involves a situation in which the material surface is perfectly wetted and the sodium is uniformly (in thickness) distributed over the surface. Consequently, this situation is valid only when the temperature is significantly above 300°F (149°C). Due to the low emittance values of sodium, which is in the order of 0.1-0.23 depending on its purity, radiant energy exchange will be affected markedly. The second case attempts to incorporate the situation in which condensation occurs without wetting, i.e., wetting angle greater than 90° , by introducing the effective emittance value which is equivalent to the weighted sums of the products of different emittance and their respective areas normalized to unit areas. The effective emissivity can be written

$$\epsilon_{\text{eff}} = \frac{\epsilon_s A_s + \epsilon_m A_m}{A_s + A_m}, \quad (103)$$

where ϵ_s and ϵ_m represent the emittance values of sodium and the material on which the vapor is deposited, and A_m is the surface area of the material that is not covered with sodium. This implies that A_s is the cross-sectional area of all the sodium droplets. Equation 103 can further be written in fractions of the total area as

$$\epsilon_{\text{eff}} = \epsilon_s \frac{A_s}{A} + \epsilon_m \frac{1 - A_s}{A}. \quad (104)$$

Any metallic or alloy surface covered with an oxide coating causes an acute increase in its emittance, which will significantly affect the characteristics of the radiant-energy exchange of that surface. In most cases, the emittance of the composite essentially exhibits the same emittance value as the

coating. Wade (1959) has shown that the stably oxidized specimens of stainless steel emit diffusely over a range of 600-2000°F (316-1093°C) and a range of incident angles. This implies that the measured value of total normal emissivity is the same as the total hemispherical emissivity.

2. Sodium Oxide

Under conditions of different temperature gradients, sodium vapors will condense on cooler surfaces of the thermal insulation. Regardless of the final form of the vapor deposit, sodium droplets on the insulation surfaces will be exposed to a significant amount of oxygen, which enters the system through the cover-gas supply, outgassing of different components, and leakage caused by routine maintenance. This will eventually oxidize all the exposed sodium surfaces. Once an outer oxidized layer is formed, it will prevent any further contact of oxygen with sodium, thereby stopping the oxidation process. Despite the fact that various arguments have been raised concerning the drainage of sodium on the insulation surface, sodium can only be drained when the surface is wetted by it. Consequently, even if there is adequate solution drainage, there will still be a thin film of sodium the thickness of which depends on the drainage capability of the surface that will be susceptible to oxidation. Effective drainage of the RSB test facility at elevated temperatures was reported by Jansing et al. (1977). Heat transfer due to the presence of the oxide layer is significantly affected. This is attributed to the increase in emittance of the oxide. In order to model the heat-transfer effects of sodium oxide contamination, it is assumed that, at elevated temperatures, radiant heat transfer between surfaces is entirely dominated by the oxide layer. For temperatures that do not allow perfect wetting of the insulation surface, so that oxidation only results in localized areas, the effective emittance of the surface can be represented by

$$\epsilon_{\text{eff}} = \epsilon_s \frac{A_s}{A} + \epsilon_o \frac{1 - A_s}{A}, \quad (105)$$

where ϵ_o is the emittance of the oxide.

The perfectly wetted model is tested only against experimental results in which good agreement is observed. The detail comparison is provided in Chapter VI. The model used is given in Sec. VI.B. To permit an accurate description of either the sodium-wetted or sodium oxide conditions, the emissivity value of stainless steel is altered accordingly. Metallic conduction along the screen wire is also changed. A thin film of sodium 1 mil (0.002 mm) thick is assumed, and thermal conduction through this film layer is calculated. This film thickness was confirmed by chemical analysis performed by Jensen (1978). Despite the fact that the partially wetted models are not tested, the perfectly wetted ones serve to bound the cases.

IV. EXPERIMENTAL SYSTEMS

A. Design and Setup

Experimental investigations were conducted to determine the apparent thermal conductance of various types of proposed thermal insulation. Two types of experimental environment, under which the insulation would be examined, were vital to the understanding of their properties and applications. The first involved a study of the insulation under various temperatures and pressures. Data collected under this condition would provide information that would lead to a better understanding of that type of test sample. A more focused study involved an examination of the insulation heat-transfer properties under sodium environment that resembles reactor conditions. Due to the use of liquid sodium in the study, experiments were performed at the Liquid Metal Experimental Modeling Facility of the Reactor Analysis and Safety Division at Argonne National Laboratory. It was also for this reason that special attention was required for the experimental design and setup of the test section.

A test vessel made up of two sections was conceived, as shown in Fig. 36. The upper section is slightly larger than the lower part, forming an edge upon which the test samples can be supported. Both sections are accessible through a covered flange. The bottom of the vessel is made up of an explosive bonded copper-cladded stainless steel disk, whose purpose is to distribute heat uniformly throughout the 18-in. (457.2-mm) base. A 0.5-in. (12.7-mm)-pipe-size drain pipe leading to an exit valve is centered at the bottom surface. In addition to providing drainage, it also serves as a cold trap where impurities are precipitated and collected due to the lower temperature. A transfer line penetrates the side wall at 5.5 in. (139.7 mm) above the base to allow sodium to be transferred in and out of the vessel. The inlet inside the vessel is 1.5 in. (38.1 mm) above the bottom. A thermocouple well is located on the wall 7 in. (177.8 mm) from the bottom edge, and the tip of the well is 2.58 in. (65.6 mm) from the bottom of the vessel. The container, a shroud welded to a 0.25-in. (6.35-mm) disk, inside which is located the test-insulation samples, is 1 ft (0.3 m) long and 17 in. (0.43 m) in diameter. It rests by its own weight on the machined edge. Threaded rods screwed into the studs on the top side of the container allow it to be transferred in and out of the vessel. Three 0.5-in. (12.7-mm) standard-size pipe, two of which are in the upper section, are provided through the side of the vessel for system-evacuation and gas-filling purposes. A 0.5-in. (12.7-mm) pipe, welded to 0.25-in. (6.35-mm) Swagelok coupling, is connected to each pipe. Neoprene is the gasket material for the flange cover. The disk of the container is always maintained below 150°F (66°C), thereby providing a freezing seal for the sodium vapor that might have traversed up the gap between the shroud and the vessel. Therefore, sodium is not expected in the upper portion of the vessel, and even if, for some unforeseen reason, sodium escapes into the upper section, the flange cover will provide adequate protection against sodium vapor leaking into the atmosphere.

Fig. 36. Design Drawing of Test Vessel

This test vessel was specially designed at Argonne National Laboratory for this series of experiments. Special caution has to be exercised to ensure the containment of sodium within the test vessel. Despite the high cost, Type 304 stainless steel, due to its high compatibility with sodium, is chosen over carbon steel. All components within the test vessel are fabricated with Type 304 stainless steel. Note from Fig. 36 that pipe-thread connections are deliberately avoided in the design, for they are most susceptible to sodium leaks, especially at high temperatures. Joints are either welded or connected together by Swagelok-type fittings. The drain valve used is a Nupro stainless steel valve. The test vessel is fabricated by Huron Fabricators, Inc., of Ohio according to a set of fabrication requirements and procedures, as listed in Table III. Radiography is performed only on the major weld that goes from the top to the bottom of the vessel. A helium leak test done by a separate agency demonstrates that no leaks are detected at a sensitivity of 4.2×10^{-10} std cm³/s.

TABLE III. Fabrication and Inspection Requirements for Test Vessel

Fabrication Requirement

Construction

All component parts shall be machined, formed, and welded to the dimensions and tolerances specified on the drawings.

Welding

Welding procedures and qualified welders as specified in paragraphs UW-47 and -48 of the ASME Boiler and Pressure Vessel Code (1974, Section IX, Welding Qualifications) shall be used in performing the welding operations.

Shielding arc-welding processes shall be used. Backup rings are not allowed. The root pass shall be effectively shielded when possible so an oxide-free interior surface is achieved.

Inspection and Tests

Liquid Penetrant

The root and final pass of all pressure-boundary welds as indicated on drawings shall be given liquid-penetrant inspection as per Appendix VIII, Division I, Section VIII, of the 1974 ASME Boiler and Pressure Vessel Code. The liquid penetrant shall be completely removed immediately after inspection.

Radiography

Welds shall be radiographed as noted on drawings, per Division I, Section VIII, paragraph UW-51, of the 1974 ASME Boiler and Pressure Vessel Code.

Helium Leak Test

Weld areas shall be mass-spectrometer vacuum-leak-tested with helium as noted on drawing. A leak detected at a sensitivity of 1×10^{-9} std cm³/s shall be cause for rejection.

Two thermocouples are located inside the thermocouple well. One is at the bottom of the well, which is 2.58 in. (65.6 mm) from the bottom of the vessel; the other is at the entrance of the well, about 5 in. (127 mm) from the base. These two thermocouples are silver-soldered onto a copper disk to fit inside of the wall. Nine other thermocouples, made from No. 20 gauge (0.8-mm) thermocouple wires with silicate-impregnated insulation to form bare junctions, were welded 180° apart onto the outside of the vessel wall with four on one

side and five on the other, as shown in Fig. 37. Similar thermocouples are located on the drain line, the fill line, the three 0.5-in. (12.7-mm) pipes, and the outside wall of the upper section, as indicated in Fig. 37.

Fig. 37. Locations of Thermocouples
Exterior to Test Vessel

Thermocouples inside the vessel, due to the possibility of their contacting sodium, are stainless steel sheathed. Twenty of them are brought through the flange cover using four Conax fittings, which provide excellent sealing under various conditions. To make provision for repeated removal of the container from the test vessel, a set of thermocouple connections is made inside the upper section. This allows the flange cover to be removed independently of the container. Of the 20 aforementioned thermocouples, two are used to record inlet and outlet coolant temperature; two others are tack-welded on the top side of the container disk in different radial locations; and the remaining 16 are instrumented inside the container

in the lower section of the vessel. Three of the 16 thermocouples are located at different axial locations at the perimeter of a test sample. The rest are all tack-welded onto the test sample under one of the three arrangements depicted in Fig. 38. The thermocouples selected for this experiment, whether sheathed or bare junction, are Type K Chromel-Alumel wires, calibrated to within 1°F (5/9°C) at 500°F (260°C).

Heater wires are tack-welded along the outside vessel wall to provide adequate heating. They are instrumented in sections so that temperature along the wall can be controlled by merely adjusting electric current to different

X Thermocouple location

Fig. 38. Arrangement of Thermocouples on Test Sample

sections. Two layers of Wetpak,* each $\frac{3}{4}$ in. (19.05 mm) thick, are placed around the outside of the vessel to reduce heat losses. Four thermocouples are located between the Wetpak layers to permit the heat losses through the side of the vessel to be estimated. To further keep the heat dissipation to a minimum, the vessel is surrounded by a 2-in. (50.3-mm) layer of fiber-glass insulation.

An air cooling system is used inside the upper section of the vessel. The cooling coil is located on the top surface of the container and is covered with a heat-transfer cement, Thermon, which has a thermal conductivity comparable with stainless steel. Both the inlet and the outlet of the cooling coil are connected to the outside by 0.25-in. (6.35-mm) Conax fittings through the side wall of the vessel, as seen in Fig. 36. Swagelok fittings located immediately before and after the Conax fittings allow easy disconnection that would facilitate the removal of the container. Aluminum foils are placed in a multilayered fashion over the cooling coil to prevent heat loss through the flange cover and the vessel wall of the upper section. The inlet and outlet coolant temperature is recorded by two thermocouples located in the midstream of the coolant flow. Gas flow rate is measured by a flowmeter by Schuttler and Koerting, which is calibrated to $\pm 1\%$ at full scale.

Another cooling system is located on the outside ledge of the vessel where the upper and lower sections meet. Its primary purpose is to maintain the upper section of the vessel around room temperature. Consequently, coupled with the air cooling system inside, the lip on which the container rests will always be maintained below the melting point of sodium, 208°F (98°C). Laboratory water is chosen as the heat coolant to be used inside the 0.25-in. (6.35-mm) copper tubing covered with Thermon. Thermocouples and a flowmeter are used to monitor the inlet and outlet temperatures and the flow rate of the coolant, respectively. The general contention is that water should not be used concurrently with sodium in any sodium system. Part of the motivation behind this belief stems from the fact that sodium, whether it is a liquid or a solid, reacts violently with water, resulting in possible explosions and sodium fires. However, in recent years, as the understanding of sodium properties progresses, water has been proposed, in different applications and with satisfactory precautionary measures, to be used within sodium systems. The water-cooling system used in this experiment satisfied the double-barrier requirement imposed, for example, on the CRBR inert-cell cooling system reported by Brubaker et al., 1978.

*Wetpak is an alumina silica fibrous thermal insulation, which is wet before application. After it has been applied and let dry, water evaporates from the material, leaving behind pockets of air spaces. It has a thermal conductivity of 0.3 Btu-in./h-ft²·°F (0.04 W/m·K) at 350°F (177°C).

Heating capabilities of the vessel are provided by two sets of coaxial stainless steel-sheathed heating cables of diameter 0.016 in. (0.41 mm), manufactured by ARI industries. They are located at the base and at the side of the lower section of the vessel. The set of 15 cables at the bottom is attached on a circular plate that is tightly held against the copper-clad base. All the cables used are 3 ft (0.91 m) long. As for the perimeter of the vessel, there are 18 cables, eight of which are concentrated on the lower 6 in. (152 mm) in order to provide more efficient heating in the area where there is sodium. The remaining heating cables are distributed uniformly along the side up to the ledge of the vessel. Trace heating for sodium-transfer lines for areas in which sodium might be present is furnished by both the coaxial stainless steel-sheathed heating cables and the heavy-duty heater cables, manufactured by General Electric, of which the latter one is easily removable.

All the coaxial heating cables are connected to a power cabinet where 408-V three-phase power supply is transformed by three transformers to 120-V ac, which subsequently supplied the cables. The electrical circuit is shown in Fig. 39. Variable transformers provide control over the heating rate of each cable. If temperature levels have to be maintained over a period of time, temperature controllers by Weathermeasure are used.

Fig. 39. Electrical Circuit of Power Supply

With sodium inside the test vessel and prolonged periods of experimentation, several precautionary measures are implemented to ensure safe operation. The portion of the vessel that contains sodium is placed inside a modified 55-gal (0.208-m³) steel drum where the vessel is supported by firebricks. A sheet of screen mesh is embedded inside a 2-in. (50.8-mm) layer of pea gravel

placed at the bottom of the 55-gal drum, shown in Fig. 40. It is electrically connected so that if there is a sodium leak from the test vessel, by the nature of its property, the open circuit between the screen mesh and the drum closes and sends out an alarm signal. The remaining cavity of the 55-gal (0.208-m^3) drum is filled with vermiculite, which serves as a buffer to keep air from the region. In the event of sodium leakage, sodium reaction with surrounding oxygen is therefore minimized.

Fig. 40

Test Vessel within 55-gal Drum
Filled with Vermiculite

B. Test Samples

Three types of metallic thermal insulation were studied: (a) the multi-layer stainless steel plates or, simply, multiplates, (b) the honeycomb, and (c) the multilayers of stainless steel plates with layers of screens in between, or multiscreen plates. The common characteristics of all these types of insulation is that they are basically composed of layers of metallic plates of which the multiplate is the fundamental concept and the others are all derivatives of it. The honeycomb and the multiscreen plates can be seen as variations of the multiplate concept in which different kinds of spacers are chosen, for example, the honeycomb with vertical metallic strips and the multiscreen plates with screens.

All three types of insulation tested in the experiments are 1 ft (0.3048 m) thick and consist of 1/32-in. (0.79-mm)-thick cold-rolled 2B finish stainless steel plates. The multiplates assembly is supported by passing three equally spaced threaded rods through the perimeter of the assembly with each plate separated by oversized nuts on the three rods, as shown in Fig. 41. For a diameter of 16 in. (0.41 m), the type of plate selected shows adequate strength to support its own weight. Two geometric configurations are studied: the 0.5-in. (12.7-mm) and the 1-in. (25.4-mm) separation. The outside perimeter of the assembly is shielded with a 1-mil (2.54×10^{-2} -mm)-thick stainless steel foil as a radiation barrier and a means of isolating the insulation from the remaining environment.

Fig. 41

Twenty-four Multiplates with Supports.
ANL Neg. No. 900-78-1234 #1.

However, only the 0.5-in. (12.7-mm) separation is tested for the honeycomb insulation. This implies that the honeycomb core used is 0.5 in. (12.7 mm) high. The core cell size is chosen to be 3/8-in. (9.53-mm) formed 2-mil (5.08×10^{-2} -mm) stainless steel strips. The test assembly is made up of five panels, each consisting of four honeycomb cores vacuum-brazed to five 1/32-in. (0.79-mm) face sheets, i.e., the stainless steel plates. Both the production and brazing of the honeycomb are done by Rohr, Inc. It is reported that quality assurance on the brazing process indicates a 70°F (21°C) complete brazing. Each panel of the assembly is separated from the other panels by a similar honeycomb core. In this case, no shield is needed on the perimeter.

Incomplete information available from French patents by Lemerrier on the French-type insulation indicates that a range of screen and wire sizes has been considered for the LMFBR application. They vary from using a 0.4-mm wire with 0.4-mm spacing to 83-mil (2.11-mm) wire and a four-mesh arrangement. General Electric, recently, in their pool-type PLBR studies proposed to use a screen that employs 80-mil (2.03-mm) wire and four-mesh spacing. The importance in the careful selection of the wire size and the mesh size of the screens is reflected in being able to minimize conduction through the wires and radiation through the screens. Forty-seven-mil (1.19-mm) wire, four-mesh Type 304 stainless steel screen is chosen for this study. Similar to the multiplates, two configurations are studied: those with 1-in. (25.4-mm)-thick screens between plates and 0.5-in. (12.7-mm)-thick screens between plates. Consideration of the fact that these types of screen insulation will be used inside a reactor vessel leads to the conclusion that fiber materials should be avoided. Remote as it might seem, any loose or broken wires, if falling into the sodium pool, could result in irreversible dampers to components of the reactor system. To assemble the 0.5-in. (12.7-mm) screen composite, five screen layers are used. Each layer is oriented with a rotation angle from the previous one. A five-layer, 0.5-in. (12.7-mm) screen is shown in Fig. 42.

Fig. 42. Screen Composites

C. Experimental Procedures

This section illustrates the procedures used during the experiment. The discussion will be confined to the heat-transfer measurements of different types of insulation. Procedures pertaining to the infrared spectroscopy study are left to the next section, where a comprehensive treatment of the entire work is given.

The experiment can be divided into three parts according to the environments in which the tests are performed: (a) nonevacuated, (b) evacuated, and (c) sodium. However, due to the similar nature of procedures in both setup and data acquisition of tests in air and in vacuum, they will be treated simultaneously. Differences and variations between tests in air and those in vacuum will be pointed out whenever they arise.

Some of the precautionary measures to ensure a leakproof system during the fabrication process of the vessel were discussed in Sec. IV.A. Minor alterations were performed at the test facility of Argonne National Laboratory. Fittings and valves were added to the test vessel, which was then subjected to another helium leak test to maintain its overall integrity.

1. Evacuated and Nonevacuated Environments

A schematic diagram of both types of environment is shown in Fig. 43 in which the only difference between them is characterized by the presence of a vacuum pumping system for the evacuated case. All the insulation test samples are degreased by either Freon-II or Freon-TF, and then washed with acetone to ensure clean test surfaces before the experiment. Thermocouples, the inside of the test vessel, and any related materials within the test vessel are also subjected to the same cleaning procedure previously mentioned. Each thermocouple is fastened to the test insulation by several pieces of 2-mil (5.08×10^{-2} -mm)-thick stainless steel shim stock. The tip of the thermocouple is tack-welded onto the insulation material. The fusing of the stainless steel on which the thermocouple junction is located and the thermocouple sheath provides an exceedingly temperature-sensitive junction, increasing the accuracy of the thermocouple. Two holes, through which 16 thermocouples from the test sample extend into the upper section, are located on the perimeter of the container disk. Through a series of thermocouple connectors, lead wires, and Conax fittings, these thermocouples are connected to a 24-point Honeywell Strip Chart Recorder. The recorder is calibrated to within 1°F (0.6°C) over the range of $32\text{--}100^{\circ}\text{F}$ ($0\text{--}38^{\circ}\text{C}$). Calibration procedures are detailed in Appendix D. Each thermocouple is disconnected from its connector whenever the flange cover is removed and is checked each time there is a change of test sample. The upper section of the test vessel is shown in Fig. 44. To maintain a good vacuum within the system, the sealant silicone rubber is applied at the ends of each sheath thermocouple lead wire that passes through one of the Conax fittings, to prevent possible leakage within the stainless steel sheath. Figure 45 is a schematic of the thermocouple connections.

Fig. 43. Schematic Diagram of Test Facility

Fig. 44. Upper Section of Test Vessel

Fig. 45

Schematic Diagram of Thermocouple Connections

After the test sample is placed inside the container supported by three angle irons that are attached to the shroud, and the thermocouples are fitted with connectors, the test sample is lowered into the vessel by a pulley. Except for the multiscreen plate, which uses a set of six spring-loaded angle irons, all other test samples use three plain angle irons for support. Subsequently, two 0.25-in. (6.35-mm) tubing connections are made with Swagelok fittings to the air-cooling coil. A positive pressure test at each connection along the coil is then conducted, and with negative results it will be followed by the insulation of the upper section with aluminum foils and the connection of the thermocouples to the flange cover. After the neoprene gasket is in place and the cover lowered, the vessel is sealed by fastening the cover to the flange with 20 bolts. Only coaxial heater wires at the bottom of the vessel are used to provide necessary heating for the vessel at various temperatures. Both the current consumed by, and the voltage sustained across, each wire are recorded after each operating temperature has attained equilibrium. The temperatures along the sidewall of the vessel as well as other temperature monitors outside the test vessel are displayed on an Omega Engineering Model 250 Chromel-Alumel digital readout. The unit is factory-calibrated and is accurate to within 1.3°F (0.72°C), as discussed in Appendix D. After the system has come to an equilibrium, each channel of the digital readout and of the strip-chart recorder is also noted. Flow rates and temperatures of both coolants are also recorded at that time.

Equilibrium of the system is determined by closely monitoring the temperature change within the system. When the change of system temperature within a few hours falls within the error limits of the system, it can be safely assumed that the system is at equilibrium. Certainly, the error limits invoked include the fluctuations of the ambient temperature as well as the

coolant pressures and the resolution of temperature signals by recorders. The average time required to attain an equilibrium for various temperatures is 16-24 h. In addition to thermal equilibrium, the evacuated system also has to attain a low enough gas pressure so that the gas contribution to heat transfer due to convection is negligible. Normally, it takes a slightly longer time to attain a thermal equilibrium and a sufficiently low gas pressure when the system is first evacuated. This is primarily due to the outgassing effects of materials, which can be alleviated by operating the system at a higher temperature, thereby enhancing the outgassing process.

2. Sodium and Sodium Oxide Environment

Because of the extreme sensitivity of sodium to water, all the insulation test samples, the test vessel, and the container were given a final cleaning with alcohol in addition to the degreasing process by either Freon-II or Freon-TF and acetone. Water vapor is carried away from the samples by the evaporating process of the alcohol. Only one test sample is investigated in sodium and sodium oxide environment.

A schematic of the operating system is shown in Fig. 43. Connected to the vessel are an argon system, a dry-air supply, and an evacuating system. The argon system provides an adequate cover-gas supply and purging capability. The dry-air supply, coupled with a liquid-nitrogen cold trap, ensures extremely low-moisture air (dew point at -196.16°C) when bled into the system. Sodium* used in this experiment is transferred initially from a 400-lb drum, which when placed inside the heating facility discourages further movement, to a working pig, which holds about 12 gal (0.454 m^3). This pig is equipped with wheels and can be transported easily. A schematic and an actual picture are shown in Figs. 46 and 47. Before the actual transfer, the sodium within the drum, the transfer line, and the pig are heated above 208°F (98°C), the melting point of sodium. Then the drum is pressurized with argon to about 2 psig (13.79 kPa) when the pig is evacuated to about 3 psig (20.69 kPa) vacuum. The two valves on the transfer line are then opened, and sodium flow is indicated by temperature changes at various locations on the transfer line due to the temperature difference between the line and the drum. The purity of the sodium is ensured by maintaining the drum at slightly higher than melting temperature [$\sim 250^{\circ}\text{F}$ ($\sim 121^{\circ}\text{C}$)] for an extended period of time, allowing impurities to precipitate on colder regions, e.g., the surface of the sodium pool. This principle of cold trapping is clearly demonstrated by Fig. 48 in which the oxygen content is plotted as a function of temperature. The above technique is used on the drum as well as on the working pig, ensuring that minimum impurity within the working sodium is maintained.

*According to the manufacturer, U. S. Industrial Chemicals Co., Ohio, impurities contained in the sodium are (1) calcium less than 400 ppm, (2) chloride less than 0.005%, and (3) potassium less than 100 ppm.

Fig. 46. Mobile Sodium Pig

Fig. 47. Schematic Diagram of Mobile Sodium Pig. Conversion factors: 1 ft = 0.3048 m; 1 in = 25.4 mm.

Fig. 48

Oxygen Content in Sodium vs Temperature.
Conversion factor: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$.

Before the sodium is further transferred into the test vessel, which is maintained at 300°F (149°C), its temperature is raised to about 450°F (232°C) and the test vessel is purged repeatedly with argon. During the transfer, the level of sodium within the test vessel is monitored by two thermocouples, located inside the thermocouple well and separated by an axial distance of 2.17 in. (55.1 mm). When the 450°F (232°C) sodium from the working pig renders a temperature rise of the lower thermocouple, transfer is terminated. During all transfer processes, pressures in both components are noted and adjusted constantly where extensive pressure buildup is deliberately avoided. A relatively low pressure required during transfer can be explained by its low viscosity depicted in Fig. 49.

Fig. 49. Relation of Sodium Viscosity and Temperature

When sodium is needed to wet a designated thickness of insulation, the test vessel is again connected to the working pig, where the inventory sodium is brought inside to raise the sodium level to the required level. To assume a complete wetting of the material, the sodium is kept at about 650°F (343°C) and is maintained at the required level overnight. Experimental data indicate that at that temperature good wetting can be assured, as can be seen from Fig. 16. Afterward, the insulation is drained slowly at the same temperature by removing sodium from the vessel and transferring it back to the pig. High-temperature draining permits only a very thin film to be left on the material. A sodium pool 5 in. (127 mm) deep is retained inside the test vessel to provide a heat source for conductance measurements.

In view of the need to create oxide deposits on the insulation sample, the method chosen to provide enough oxygen for such reaction is by repetitively filling and evacuating the system with dry air. The manufacturer's specification of cylinder dry air has a dew point of -76°F (-60°C). Nonetheless, an extra cold trap using liquid nitrogen is employed to safeguard any influx of moisture into the vessel. During the oxidation process, the sodium pool is maintained at a very low temperature to reduce the solubility of oxide in sodium. Excessive oxidation of the pool is also prevented due to the formation of an oxide layer, which inhibits extensive penetration of oxygen into the sodium. The process is concluded with the purging of the system with argon. Two levels of immersion of the test sample are performed: 5 and 8 in. (127 and 203.2 mm). The amount of dry air needed for the reaction increases proportionally with the increase in sodium-covered surface area.

After the experiment is concluded, the test sample is disassembled for examination. The test sample is transferred from the vessel by first freezing whatever sodium is left in the vessel, and, with a continual supply of argon to the test vessel, a plastic bag is placed over the cover-flange area after the flange has been removed. The plastic bag is then purged several times before the test sample is raised slowly from the vessel into the bag, as shown in Fig. 50. During the transfer process, argon is being replenished continually in the test vessel as the test sample is raised. A separate argon supply is also furnished at the top of the plastic bag. The heavier nature of argon as compared to air ensures a sustained argon atmosphere inside the bag and the test vessel. When the bottom of the bag is sealed off, the test sample is then inside an argon-filled containment, where it can be transported safely to the sodium scrubber for rinsing. The sodium and sodium oxide on the surface of the test sample is then dissolved in 15 gal (0.0568 m³) of 100% anhydrous ethyl alcohol.

A second rinse uses another 15 gal (0.0568 m³) of 100% anhydrous ethyl alcohol to remove the remaining sodium on the insulation. Fifteen gallons (0.0568 m³) of water are used as a final rinsing agent to clean the test sample before the test sample is disassembled. Chemical samples are collected for these rinses for analysis to reveal the sodium content within the solutions. Finally, the residual sodium left at the bottom of the test vessel is disposed of by burning in the scrubber. The combustion products are adequately scrubbed and disposed of.

Fig. 50. Transfer of Sodium-wetted Test Sample

D. Infrared Measurements and System

To be able to apply the analytical model to predict the experimental results, the emissivity of stainless steel used has to be known. Infrared technology is chosen to determine the emissivity value of stainless steel as a function of temperature. The experimental system comprises an infrared camera unit, a black and white display unit, and a color monitor. A schematic of the experimental setup is shown in Fig. 51. Styrofoam slabs were used around the samples and the camera setup to prevent thermal-radiation reflection from warm objects in the vicinity of the system. Figure 52 illustrates the arrangement of various equipment and test samples. The camera unit is a lightweight, hand-held, real-time electro-optical prism scanning system with interchangeable lenses and various spectral-range filters. Indium antimonide (InSb) which has a spectral range of 2-5.6 μm under liquid-nitrogen temperature, is used as a photovoltaic infrared detector. The thermal-display unit has a scanning-line frequency of 2500 per second and 280 lines per frame (1:4 interlaced). With the interchangeable gray filter, temperature measurements range from -20°C (-4°F) to 2000°C (3632°F), where the minimum detectable temperature difference at 30°C is 0.2°C . Objects with different temperature characteristics are displayed with different gray tones. The color monitor displays various gray tones in different colors to allow easy distinction.

Fig. 51. Schematic Diagram of Infrared System

Fig. 52. Experimental Setup of Infrared Measurement

Test samples are cut into rectangular plates and then cleansed with a degreasing agent, acetone, and alcohol. A Type K Chromel-Alumel thermocouple is spot-welded on the central portion of each plate. A black-body reference is provided by depositing candle soot, which has an emissivity value of 0.95 according to Touloukian (1970), onto one of the stainless steel plates. Then each test sample and a black-body reference are attached to asbestos paper, where slightly undersize holes on the paper allow better heating from a heater placed behind the asbestos paper. Temperature signals from the thermocouples are displayed on a digital temperature readout.

A portion of the object is scanned sequentially by the vertical and horizontal scanning unit of the camera, where emitted infrared radiation is detected

by the indium antimonide crystal. By the photovoltaic process a small electric current is generated and further amplified, and when this current is coupled with the triggering generators of the camera, a thermal picture as described earlier is produced on a cathode-ray tube. Areas that are detected as emitting the same amount of radiation are characterized by the same shade of gray or the same color. Isotherm units are defined by the manufacturer of the detector to characterize different layers of radiation intensity, whereby, with the help of the calibration graphs, the corresponding temperature can be evaluated. With a standard black-body reference, a difference in isotherm unit ΔI measured between the object and the reference would enable the evaluation of the emissivity of the object by the energy-balance equation

$$\epsilon_o(I_o - I_a) = \Delta I + \epsilon_r(I_r - I_a), \quad (106)$$

where ϵ_o and ϵ_r denote the emissivities of the object and the reference, respectively, and I_o , I_a , and I_r represent the absolute isotherm level for the objective, the ambient, and the reference temperatures, T_o , T_a , and T_r , obtained from the calibration charts.

Two different samples, characterized by different surface conditions, are studied. They are both Type 304 stainless steel material. The first kind, used exclusively in construction of the test vessel as well as the insulation samples, except for the face sheets of the honeycomb insulation, has a 2B cold-rolled surface finish. The other kind is a nickel-based brazed alloy, vacuum-plated on the surface of the first kind. The emissivity data for both samples are presented as a function of temperature in Figs. 53 and 54. The plain stainless steel exhibits a very low emissivity value, 0.05 at about 38°C (100°F), but when the temperature is increased to slightly above 427°C (800°F), a darkening of the surface to bluish brown is observed. A similar appearance is also noted for portions of different insulation test samples which have been exposed to elevated temperatures. The brazed-alloy sample shows a somewhat different temperature dependence. From room temperature up

Fig. 53. Emissivity of Stainless Steel with 2B Cold-rolled Finish at Different Temperatures

Fig. 54. Emissivity of Nickel-based Brazed Alloy vs Temperature

to about 260°C (500°F), the emissivity maintains a fairly constant value of 0.24. Then it exhibits a gradual increase to 0.35 at about 460°C (860°F). This indicates that the presence of oxidation of the surface is an important condition and can strongly influence energy transport.

Both samples are subjected to high temperature for an extended period of time before another set of emissivity measurements are made. A 13% increase is noted for the plain stainless steel sample as depicted by Fig. 55 where emissivity is plotted against time. A similar increase of 12% is observed in Fig. 54 for the brazed-alloy sample. Afterward, the emissivity of the plain stainless steel sample is measured as its temperature is being reduced. The results are shown in Fig. 56. Little change in emissivity value is observed as temperature decreases from 427 to 177°C (800 to 350°F).

Fig. 55. Emissivity vs Time
at 427°C for 2B Finish
Stainless Steel

Fig. 56. Emissivity vs Tem-
perature after 45 Hours
at 427°C

The error involved in this measurement stems from (a) the accuracy of the calibration curve and (b) the systematic and random error of the camera and the display unit. The manufacturer AGA has reported that a $\pm 3\%$ error is associated with the calibration of a large number of these units. The remaining contribution of error is estimated by having, besides the object sample, two reference black bodies in each emissivity measurement, allowing comparison among the three samples. The total error is estimated to range from about 6 to 10%.

E. Energy Balance

Two types of energy balance are used to accurately assess the data collected by the experimental setup. The first is a global-energy balance of the system, as shown in Fig. 57, in which the sum of energy input is balanced with the amount lost through the system boundary. Any significant deviation from this global balance would automatically imply that there are possible instrumentation error and/or equipment failure. This balance is performed

and checked each time for consistency before data are acquired. The energy balance can be expressed as

$$Q_{\text{input}} = Q_{\text{loss}}, \quad (107)$$

$$Q_{\text{input}} = \sum_i \dot{Q}_i = \sum_i \dot{V}_i I_i, \quad (108)$$

and

$$Q_{\text{loss}} = Q_B + Q_W + Q_G + Q_S. \quad (109)$$

The heat-loss terms can be further written as

$$Q_B = h_B A_B \Delta T_B, \quad (110)$$

$$Q_W = \dot{m}_w C_{p_w} \Delta T_w, \quad (111)$$

$$Q_G = \dot{m}_G C_{p_G} \Delta T_G, \quad (112)$$

and

$$Q_S = \frac{2\pi k s}{\ln \frac{r_o}{r_i}} \int \Delta T(z) dz. \quad (113)$$

Fig. 57

Global-energy Balance

Q_a = Heat Transfer To Air
 Q_w = Heat Transfer To Water
 Q_h = Heat Transfer From Heater
 Q_s = Heat Transfer To Side Wall
 Q_b = Heat Transfer To Bottom

$$Q_h = Q_s + Q_a + Q_w + Q_b$$

Equations 107-109 express the relation that heat loss through the base, the water-cooling coil, the gas-cooling coil, and the side wall is equal to heat gain from the sum of the heating power generated by all the electric heating cables. In Eq. 108, the subscript i is the index of summation and P , V , and I represent power of, the voltage across, and the current within each heating cable. The subscripts B , W , G , and s denote quantities that are related to the base of the vessel, the water coil, the air coil, and the side of the vessel, respectively; ΔT , $\dot{m}$, A , C_p , and h represent the temperature difference, mass flow rate, area, heat capacity, and heat-transfer coefficient, respectively. Except for h , A , and C_p , the quantities are experimentally measured. The thermal-conductance dependence on temperature of Wet Pak insulation is shown in Fig. 58 from information released by Refractory Products Co. The ceramic insulation is used to insulate the side as well as the base of the experimental vessel. Moreover, the ways in which the thermocouples are instrumented permit the axial dependent ΔT function of Eq. 113 to be derived from an Nth-order polynomial fit of the experimental data. Thus, the equation can be evaluated accurately. For all the experiments performed, there is global energy balance to within 8-10% accuracy.

Fig. 58

Thermal Conductance of Wet Pak vs Temperature.
Conversion factors: $^\circ\text{C} = (^\circ\text{F} - 32)/1.8$; $1 \text{ Btu}\cdot\text{in.}/\text{h}\cdot\text{ft}^2\cdot^\circ\text{F} = 0.144 \text{ W}/\text{m}\cdot\text{K}$.

The second type of energy balance involves examining the energy gained and lost specifically for each layer of the multiplate insulation, as shown in Fig. 59. From thermocouples located at different axial positions within the insulation, a temperature profile pertaining to insulation is obtained by interpolation. Temperature distributions of the vessel wall as well as of the radiation shield are also experimentally measured. An energy balance is set

up between the amount of energy transfer from one layer to the next layer and the amount that leaves the periphery of the first layer. Mathematically, it can be represented as

$$Q_x(i+1) = Q_x(i) + Q_s(i+1), \quad (114)$$

where $Q_x(i)$ represents the net amount of heat energy that leaves or arrives at the i th surface; and $Q_s(i+1)$ is the amount of heat transfer from the $(i+1)$ surface to the periphery. We can write $Q_x(i)$ for multilayers of reflective plates, assuming a gray-body limit, as

$$Q_x(i+1) = Q_x(i) + Q_s(i+1)$$

Fig. 59. Energy Balance within Insulation Test Sample. Conversion factor: 1 in. = 25.4 mm.

$$Q_x(i) = \frac{1}{R_i} (\sigma T_{i+1}^4 - \sigma T_1^4), \quad (115)$$

where R_i is the thermal resistance between the i th plate and the $(i+1)$ th plate and is given by

$$R_i = \frac{1}{A_i F_{i,i+1}} + \frac{1 - \epsilon_i}{\epsilon_i A_i} + \frac{1 - \epsilon_{i+1}}{\epsilon_{i+1} A_{i+1}}, \quad (116)$$

with A_i , ϵ_i , and $F_{i,i+1}$ representing the area, emissivity, and configuration factor from surface i to surface $i+1$, respectively. We can evaluate $Q_s(i)$ by simply comparing the temperature distribution between the radiation shield and multilayer plates as

$$Q_s(i) = \frac{1}{R_{si}} (\sigma T_1^4 - \sigma T_s^4), \quad (117)$$

where R_{si} , the thermal resistance between the i th surface and the portion of the radiation shield bounded by the i th and the $(i+1)$ th plates, is given by

$$R_{si} = \frac{1 - \epsilon_i}{\epsilon_i A_i} + \frac{1 - \epsilon_{si}}{\epsilon_{si} A_{si}} + \frac{1}{A_s F_{si}}. \quad (118)$$

Here, ϵ_{si} denotes the emissivity of the radiation shield between the i th and the $(i+1)$ th plates, and A_{si} is the associated area. Therefore, from Eqs. 114-118, the net amount of energy exchange can be estimated. A few Btu/h are attributed to the loss through the periphery. When this balance is applied to each layer of the multiplate configuration, the energy obtained for the last plate (namely, the coldest surface), compared with the heat flux measured by the air-cooling coil, indicates good agreement.

V. DISCUSSION OF EXPERIMENTAL RESULTS

A. Multilayer Stainless Steel Plates

Multilayer stainless steel plates of 12 and 24 layers comprising a total insulation thickness of 12 in. (305 mm) were experimentally studied. The insulation with different geometries was subjected to atmospheric and evacuated conditions under which the data were collected. The temperature at the hot face of the insulation was also varied.

1. Twelve Plates

Figure 60 shows the axial temperature distribution versus the axial distance. The gas pressure inside the test section when the measurement was made was atmospheric. For the first 7 in. (177 mm) the temperature drop is less than 100°F (56°C); in the remaining 5 in. (127 mm) there is a temperature reduction in excess of 100°F (56°C). Figure 61 shows the temperature profiles of the test vessel. Except for the first inch at the colder face, the test-section temperature is unequivocally higher than that of the insulation. Despite the presence of the radial thermal shield, a small amount of heat is transferred from the insulation to the vessel wall. This is demonstrated by the energy balance between this heat outflow versus heat through-put across the insulation. The difference in temperatures recorded by thermocouples

Fig. 60. Temperature Distribution of 12 Multiplates, Nonevacuated, with Hot-face Temperature at 340°F (171°C). Conversion factor: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$.

Fig. 61. Temperature Distribution of Test-vessel Wall Concurrent to Fig. 60 Measurements. Conversion factor: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$.

located in the midsection versus the perimeter of the insulation were comparatively substantial, in the order of 15-20°F (8-12°C). This is a direct consequence of the fact that convection prevails in these situations. Analytically, under idealized conditions,

$$q_{\text{total}} = h_c A \Delta T + h_R A \Delta T \quad (119)$$

(see Chapter III discussion), the existing configuration of a 1-in. (25-mm) gap and the given temperature difference do not lend themselves to the onset of thermal convection. Observation of convection can be ascribed to the fact that perturbation occurred in the axial direction near the edge and in the azimuthal direction. Another major reason that supports the conclusion of thermal-convection occurrence stems from the evacuated data taken at the same temperature. The described feature vanished accordingly.

Fig. 62. Temperature Distribution of 12 Multi-plates, Nonevacuated, with Hot-face Temperature at 615°F (324°C). Conversion factors: °C = (°F - 32)/1.8; 1 in. = 25.4 mm.

The temperature distribution of a similar test but at a higher temperature is shown in Fig. 62. The feature signifying thermal convection is not as pronounced as in the last test, but it is evident. Close examination shows that the inherent characteristics of the curve depicted in Fig. 62 is identical to what is shown in Fig. 60.

When the test vessel is evacuated, heat transfer due to thermal convection can be eliminated. This further confines the energy-transfer processes to only thermal radiation and residual gas conduction. Figures 63-65 depict the temperature distribution within the insulation in different temperature domains. A least-squares fit applied to each of these profiles indicates its linear dependency over axial distance. The surrounding wall temperatures for these profiles are presented in Figs. 66 and 67. Due to the sensitive dependency of radiation heat transfer on temperature, energy throughput of the test insulation is significantly less at a lower

mean temperature. (Mean temperature is defined to be the average temperature between the hot and cold faces.) Despite the fact that comparatively large thermal resistance for radial energy transfer from the test sample to the vessel wall exists, small losses (in the order of a few Btu/h) can sustain a deviation from the anticipated results.

Fig. 63. Temperature Distribution of 12 Multi-plates, Evacuated, with Hot-face Temperature at 390°F (199°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Fig. 64. Temperature Distribution of 12 Multi-plates, Evacuated, with Hot-face Temperature at 548°F (287°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Fig. 65. Temperature Distribution of 12 Multi-plates, Evacuated with Hot-face Temperature at 690°F (366°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Fig. 66. Temperature Distribution of Test-vessel Wall Concurrent to Fig. 63 Measurements. Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

2. Twenty-four Plates

Fig. 67

Temperature Distribution of Test-vessel Wall Concurrent to Fig. 64 Measurements. Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

The test results presented in Figs. 68 and 69 are of a 24-plate thermal insulation exposed to different temperatures. Data were collected in an atmospheric condition. Both curves display a very linear behavior over the axial distance of the insulation. This contrasts greatly with the results for the 12 plates obtained under similar conditions. The linearity of these results shows that a minimal amount of heat transfer can be attributed to natural convection. This can be understood analytically from the fact that the separation distance between any two horizontal surfaces is related to the cube root of the Rayleigh number. Therefore, despite small perturbation in temperature distribution within the test vessel, a reduction in the plate-separation distance can significantly increase the thermal stability within the region.

Fig. 68. Temperature Distribution of 24 Multiplates, Nonevacuated, with Hot-face Temperature at 442°F (228°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Fig. 69. Temperature Distribution of 24 Multiplates, Nonevacuated, with Hot-face Temperature at 543°F (284°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

When the test vessel was evacuated and the same insulation was subjected to a range of temperatures, results obtained are as plotted in Figs 70-72. The linear dependency on the axial distance is again evident for all three hot-face temperatures.

Fig. 70. Temperature Distribution of 24 Multiplates, Evacuated, with Hotface Temperatures at 350°F (177°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Fig. 71. Temperature Distribution of 24 Multiplates, Evacuated, with Hotface Temperature 440°F (227°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Fig. 72

Temperature Distribution of 24 Multiplates, Evacuated, with Hot-face Temperature at 550°F (288°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

For all the evacuated cases, the maximum temperature difference between the axial temperature of the insulation and the test-vessel wall is not in excess of 50°F (28°C) for the nonevacuated cases, the temperature difference is about 100°F (56°C). This is a direct consequence of the presence of natural convection within the insulation and of the increased thermal-radiation heat transfer through the wider gap which reduces the temperature difference between plates.

B. Honeycombs

Due to the composition of the honeycomb insulation (detailed description is given in Sec. IV.A), in which honeycomb cores are placed between layers of stainless steel plates, the gaps normally existing between multiple-layer plates are now being compartmentalized into various cells 3/8 in. (9.5 mm) wide. Coupled with the 0.5-in. (12.7-mm) cell height, this cell arrangement further minimizes the possibility of natural-convection occurrence within the cell. From an experimental point of view, the presence of the honeycomb core also enhances the isolation of the test vessel with the test sample. Moreover, to facilitate comparison among various types of insulation, the same types of stainless steel plates were used to form the honeycomb insulation face sheets. Therefore, the honeycomb results are obtained under the best-controlled conditions.

The temperature distributions for atmospheric conditions over the total axial length are given in Figs. 73 and 74. A linear behavior is observed for

Fig. 73. Temperature Distribution of Honeycomb, Nonevacuated, with Hot-face Temperature at 310°F (154°C). Conversion factors: °C = (°F - 32)/1.8; 1 in. = 25.4 mm.

Fig. 74. Temperature Distribution of Honeycomb, Nonevacuated, with Hot-face Temperature at 480°F (249°C). Conversion factors: °C = (°F - 32)/1.8; 1 in. = 25.4 mm.

both cases. The temperature difference between the axial temperature and test-vessel wall temperature is about 100°F (56°C). Even with the presence of air, results confirm the lack of thermal convection within the insulation assembly. Deviation from linearity diminishes further for temperature distribution under the evacuated conditions depicted in Figs. 75-78. The temperature difference between the wall and the insulation is substantially larger than any encountered earlier. The biggest difference is almost 200°F (111°C) shown in Fig 77. This clearly shows that, due to the high thermal resistance of the insulation, the temperature within the composite experiences a more severe drop than that of the wall. This effect is not as pronounced in the previous cases.

Fig. 75

Temperature Distribution of Honeycomb, Evacuated, with Hot-face Temperature at 320°F (160°C). Conversion factors: °C = (°F - 32)/1.8; 1 in. = 25.4 mm.

Fig. 76

Temperature Distribution of Honeycomb, Evacuated, with Hot-face Temperature at 430°F (221°C). Conversion factors: °C = (°F - 32)/1.8; 1 in. = 25.4 mm.

Fig. 77

Temperature Distribution of Honeycomb, Evacuated, with Hot-face Temperature at 670°F (354°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Fig. 78

Temperature Distribution of Honeycomb, Evacuated, with Hot-face Temperature at 695°F (368°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

C. Screen-type Insulations

1. With 12 Reflective Plates

In this arrangement, reflective stainless steel plates of the same type were placed between every inch of screens. Figures 79-81 depict the temperature profiles of the insulation in different temperature ranges. These data were collected under atmospheric conditions when the test vessel was filled with air. A notable feature that resembles the 12 and 24 plates is the slight deviation from the linear behavior at higher temperatures. This is evident in all three temperature ranges. The effect can possibly be attributed to the influence from thermal convection and test-vessel wall effects. The presence of the screen material serves to divide the 1-in. (25.4-mm) gap into various irregular-size compartments with the intention of confining thermal convection to a localized region if not totally eliminating it. Significant improvements in thermal stability are noted when compared to the 12 plates. Detailed comparison and discussion of different types of insulation are reserved until the next chapter.

Fig. 79. Temperature Distribution of 12 Screen Plates, Non-evacuated, with Hot-face Temperature at 300°F (149°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Fig. 80. Temperature Distribution of 12 Screen Plates, Non-evacuated, with Hot-face Temperature at 365°F (185°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Fig. 81. Temperature Distribution of 12 Screen Plates, Non-evacuated, with Hot-face Temperature at 470°F (243°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Similar tests were also performed for this insulation under evacuated conditions. Results over a range of hot-face temperature are given in Figs. 82-85. The temperature profiles exhibit a very linear behavior over the length of the insulation. Except for the data point at 12 in. (305 mm), an essentially perfect linear fit can be obtained with the remaining data points. The variance between that point and the linear fit is about 10°F (5.6°C) over the temperature range measured.

Fig. 82. Temperature Distribution of 12 Screen Plates, Evacuated, with Hot-face Temperature at 350°F (177°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Fig. 83. Temperature Distribution of 12 Screen Plates, Evacuated, with Hot-face Temperature at 490°F (254°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Fig. 84. Temperature Distribution of 12 Screen Plates, Evacuated, with Hot-face Temperature at 620°F (327°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Fig. 85. Temperature Distribution of 12 Screen Plates, Evacuated, with Hot-face Temperature at 580°F (304°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

2. With 24 Reflective Plates

For the same overall insulation thickness, instead of using 12 reflective plates, we used 24, reducing the gap distance between plates to 0.5 in. (13 mm). The gaps were then filled with stainless steel screens. Figures 86-88 depict the temperature distributions of the insulation over a range of temperatures. Similar profiles are noted when compared to corresponding 12-screen-plate and honeycomb tests. This indicates the comparable influence due to natural convection or the lack thereof within the screens. Figures 89-92 show the temperature profiles of this insulation recorded under evacuated conditions. The linear dependency for all these curves further accents the slight possibility of thermal-convection occurrence inside the screen layers when filled with air. Detailed treatment and discussion are refrained until the next chapter.

Fig. 86

Temperature Distribution of 24 Screen Plates, Nonevacuated, with Hot-face Temperature at 407°F (208°C). Conversion factors: °C = (°F - 32)/1.8; 1 in. = 25.4 mm.

Fig. 87

Temperature Distribution of 24 Screen Plates, Nonevacuated, with Hot-face Temperature at 475°F (246°C). Conversion factors: °C = (°F - 32)/1.8; 1 in. = 25.4 mm.

Fig. 88. Temperature Distribution of 24 Screen Plates, Non-evacuated, with Hot-face Temperature at 556°F (291°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Fig. 89. Temperature Distribution of 24 Screen Plates, Evacuated, with Hot-face Temperature at 345°F (174°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Fig. 90. Temperature Distribution of 24 Screen Plates, Evacuated, with Hot-face Temperature at 480°F (249°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Fig. 91. Temperature Distribution of 24 Screen Plates, Evacuated, with Hot-face Temperature at 596°F (313°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Fig. 92. Temperature Distribution of 24 Screen Plates, Evacuated, with Hot-face Temperature at 650°F (343°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

3. Sodium Immersion and Oxidation

The thermal insulation described in Sec. 2 above was further tested by using sodium as a heat source. This essentially will confirm the data collected earlier by using either radiative or convective means of heating the test sample.

Fig. 93. Temperature Distribution of 24 Screen Plates with Sodium as a Heat Source. Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Figure 93 indicates the temperature profile versus the axial distance. The fact that a smooth curve is obtained can be ascribed to the change of the inside atmosphere of the vessel from air to argon, as plotted in Fig. 18. The inert gas argon has properties that give rise to less stable conditions thermodynamically. This shows that, if there is natural convection in tests described in Sec. 2 above, it would be initiated under onset conditions. For $\Delta T L^3$ that is in the order of 10^{-4} , a slight increase in the Rayleigh number renders the conditions thermally less stable. Conclusively speaking the systems to be presented, which are argon-filled and use liquid sodium as a heat source are more susceptible to the onset of natural convection than those tested in air.

The insulation composite was immersed in hot sodium for an extended period to allow wetting. Between 5 and 5.5 in. (127 and 140 mm) of the insulation was covered with

sodium. After a prescribed amount of time as determined by the wetting property of stainless steel (refer to Chapter IV for details), excess sodium was drained away from the insulation. Only a few inches of sodium are retained in the test vessel to serve as a heat source as described in the previous paragraph. Temperatures recorded by thermocouples located inside the insulation are shown in Figs. 94-96. A distinct feature can be noted in which the temperature profiles for either of these three tests exhibit a discontinuity of the first derivative at the midsection. The location at which the continuity occurs agrees with the height of sodium immersion. Due to the presence of the liquid sodium, the capacity to transfer heat through the composite is significantly enhanced, causing the reduction in temperature difference between plates. When temperature is increased, the temperature difference in the sodium-wetted region is also increased accordingly. This indicates that, at elevated temperatures, thermal radiation becomes a more pronounced means of heat transfer than conduction and, therefore, the discontinuity of the profile is less obvious, as seen in Fig. 96.

Fig. 94. Temperature Distribution of 24 Screen Plates Wetted with 5 in. (127 mm) of Sodium, Hot-face Temperature at 390°F (199°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Fig. 95. Temperature Distribution of 24 Screen Plates Wetted with 5 in. (127 mm) of Sodium, Hot-face Temperature at 455°F (235°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Fig. 96
Temperature Distribution of 24 Screen Plates Wetted with 5 in. (127 mm) of Sodium, Hot-face Temperature at 840°F (449°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

To measure the effects that sodium oxide have on the thermal physical properties of the insulation, repetitive purging and filling of the test vessel with dry oxygen were performed to provide an adequate oxygen supply to allow oxidation of the sodium layers. Tests similar to those aforementioned were undertaken and results are shown in Figs. 97-100. The presence of liquid sodium within the few inches of the insulation is certainly evident. Nevertheless, after the oxidation process, the effects observed are indistinguishable from sodium-wetted tests.

Fig. 97

Temperature Distribution of 24 Screen Plates Wetted with 5 in. (127 mm) of Sodium, Hot-face Temperature at 845°F (452°C). Conversion factors: °C = (°F - 32)/1.8; 1 in. = 25.4 mm.

Fig. 98

Temperature Distribution of 24 Screen Plates Wetted with 5 in. (127 mm) of Sodium Oxidized, Hot-face Temperature at 698°F (370°C). Conversion factors: °C = (°F - 32)/1.8; 1 in. = 25.4 mm.

Fig. 99

Temperature Distribution of 24 Screen Plates Wetted with 5 in. (127 mm) of Sodium, Oxidized, Hot-face Temperature at 560°F (293°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Fig. 100

Temperature Distribution of 24 Screen Plates Wetted with 5 in. (127 mm) of Sodium, Oxidized, Hot-face Temperature at 430°F (221°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Another set of tests, similar in nature to those presented earlier, was conducted, except that the insulation had been immersed in 8 in. (203 mm) of liquid sodium. Long-time immersion was allowed to permit a thorough removal of the oxide layer by the inventory sodium through solubility. Figures 101-103 depict the results with 8-in. (203-mm) immersion. The discontinuity is less obvious because most of the data points are clustered within the wetted region.

However, the sodium elevation was confirmed when the test assembly was disassembled. A similar behavior was also observed for temperature distribution of 8-in. (203-mm) sodium immersion after oxidation. The results are presented in Figs. 104-106.

Fig. 101

Temperature Distribution of 24 Screen Plates Wetted with 8 in. (203 mm) of Sodium, Hot-face Temperature at 857°F (458°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Fig. 102

Temperature Distribution of 24 Screen Plates Wetted with 8 in. (203 mm) of Sodium, Hot-face Temperature at 690°F (366°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Fig. 103

Temperature Distribution of 24 Screen Plates Wetted with 8 in. (203 mm) of Sodium, Hot-face Temperature at 520°F (271°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Fig. 104. Temperature Distribution of 24 Screen Plates Wetted with 8 in. (203 mm) of Sodium, Oxidized, Hot-face Temperature at 448°F (231°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Fig. 105. Temperature Distribution of 24 Screen Plates Wetted with 8 in. (203 mm) of Sodium, Oxidized, Hot-face Temperature at 615°F (324°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Fig. 106

Temperature Distribution of 24 Screen Plates Wetted with 8 in. (203 mm) of Sodium. Oxidized. Hot-face Temperature at 820°F (438°C). Conversion factors: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$; 1 in. = 25.4 mm.

Figures 107-109 show traces of sodium and sodium oxide adhering to the screens and plates. These photographs were taken during the final disassembly process, when sample solutions were collected for chemical analysis. Drain holes can be seen in Fig. 107. They are 5/8 in. (15.9 mm) in diameter and 4 in. (102 mm) from the perimeter. These drain holes are presented in alternating layers for the first 6 in. (152 mm). Chemical analysis of the various solution rinses by Jensen (1978) confirmed a film thickness within the screen to be about 1 mil (0.025 mm). Jensen used two methods (titration and atomic absorption) to accurately ascertain the amount of sodium present in the solution.

Fig. 107. Sodium Oxide Deposits on Plate. ANL Neg. No. 900-78-1179 #17A.

Fig. 108

Example of Sodium Oxide Deposits
on First Few Screens. ANL Neg.
No. 900-78-1179 #13A.

Fig. 109

Example of Sodium Oxide Deposits
on Last Few Screens. ANL Neg.
No. 900-78-1179 #12.

VI. COMPARISONS BETWEEN DATA AND THEORY

A. Discussion of Reduced Experimental Measurements1. Honeycombs

To be able to use the data presented in Sec. V.B requires that they be further reduced. Based on the heat transfer measured experimentally, the effective thermal conductivity of the honeycomb insulation is calculated for results obtained under evacuated and atmospheric conditions. The ordinate is normalized to the maximum ratio of Q to ΔT . Figure 110 depicts a very weak dependency, about 20% of the $Q/\Delta T$ ratio, over the range of temperatures considered. For all practical purposes, there is essentially no distinction between the evacuated and the atmospheric systems. Therefore, coupled with the fact about the identical behavior of temperature profiles presented in Chapter V, the data clearly indicate natural convection within each honeycomb cell is negligible. Figure 110 assumes a constant area A and a constant axial spatial difference ΔX . Consequently, the ratio $Q/\Delta T$ is henceforth directly proportional to the effective thermal conductivity.

Fig. 110

Normalized Q/DT vs Temperature for Honeycomb. Conversion factor: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$.

The effective thermal conductivity is calculated based on the equation

$$KA = \frac{\Delta T}{\Delta X} \quad (120)$$

or

$$K = \frac{Q}{\Delta T} \frac{\Delta x}{A}, \quad (121)$$

where all variables on the right-hand side of Eq. 121 are measurable. The results are depicted in Fig. 111, with solid symbols denoting atmospheric conditions. The temperature indicated by the abscissa is the mean temperature between the prescribed axial distance difference. The results presented in Fig. 111 exhibit the same dependency in temperature as those depicted in Fig. 110. For the type of honeycomb used in the experiment [0.5-in. (12.7-mm) height and 3/8-in. (9.53-mm) cell diameter], the reduced data indicate no observable difference between evacuated and atmospheric conditions. The data

Fig. 111

Effective Thermal Conductivity vs Temperature for Honeycomb. Conversion factors: 1 Btu/h·ft·°F = 1.73 W/m·K; °C = (°F - 32)/1.8.

point representing the highest mean temperature is around 420°F (216°C), which is equivalent to a hot-face temperature of about 700°F (371°C) if the cold face is kept to 150°F (66°C). Small extrapolation to a higher mean temperature of 525°F (274°C) would be tantamount to having a hot-face temperature of 900°F (482°C). Such extrapolation is acceptable because the second derivative of the linear curve fitting of these data points is very stable.

For comparison purposes, the theoretical results for honeycomb insulation are also plotted in Fig. 111. Radiation transfer assumes

$$Q_3 = \sum_{i=1}^3 G_{i3} A_i E_{bi} - \epsilon_3 A_3 E_{b3}, \quad (122)$$

which is derived in Chapter III.

The emissivity of stainless steel is chosen to be 0.3 (see Chapter IV), and the configuration factor hidden in G_{ij} is deduced from literature published by Hottel on radiation exchange between parallel disks and squares. Heat transfer due to conduction through the honeycomb core is calculated analytically based on the equivalent area of the experimental test sample according to the Fourier equation of conduction. Gaseous conduction is neglected. Good agreement is evident. As temperature decreases, the theoretical curve is going to asymptotically approach a limit which signifies the metallic conduction limit. At the high-temperature regime, the dominance of thermal radiation is succinctly displayed through progressively nonlinear characteristics. Also depicted in Fig. 112 is the thermal conductivity of argon and air as a function of temperature.

2. Twelve-plate Results

As described in Chapter IV, the 12-plate configuration is one of the most simple geometries. Twelve layers of stainless steel plates were spaced 1 in. (25.4 mm) apart to form a composite. In Fig. 112, the values of the heat flux to the temperature-difference ratio normalized to the maximum are plotted. The abscissa represents the hot-face temperature of the test sample. Data were collected in both evacuated and atmospheric conditions. Both sets of

Fig. 112. Normalized Q/DT vs Temperature for 12 Multiplates. Conversion factor: °C = (°F - 32)/1.8.

data exhibit a linear characteristic over the temperature range between 300 and 700°F (150 and 370°C). The presence of air in the sample caused a substantial increase in the normalized $Q/\Delta T$ ratio. Moreover, there is a proportional increase in the nonevacuated condition throughout the temperature range of concern. This increase results notably from the dominant convective heat transfer with larger ΔT at higher temperatures. As pointed out in Chapter V, analytical prediction precludes any thermal convective heat transfer under the above conditions; nevertheless, perturbations in temperature contribute to the onset condition of thermal instability.

The steady increasing $Q/\Delta T$ value indicates the possibility of a diminishing temperature difference, a growing heat transmission through the sample, or both. For the evacuation condition, the sole heat-transfer mechanism is thermal radiation, and its effects upon the thermal properties of the sample are evident. However, black-body thermal radiation shows a highly nonlinear dependence on temperature. Such dependence is not most apparent here because only a small temperature range and only reasonably low temperatures are considered. Contributions that can be ascribed to the presence of air are obvious.

Fig. 113

Effective Thermal Conductivity vs Temperature for 12 Multiplates.
Conversion factors: 1 Btu/h·ft·°F = 1.73 W/m·K; °C = (°F - 32)/1.8.

The substantial difference cannot be solely attributed to the conductive heat transfer of the gas; therefore convection would also have to prevail within the insulation. Figure 113 shows the effective thermal conductivity as a function of the average temperature. The smooth curve in the figure represents the results obtained based on the analytical approach given in Chapter III. There is fairly good agreement between theory and the experimental results. The analytical results derived from the theory of thermal radiation can be approximated by a straight line over the temperature range of interest. Except for about a threefold increase in the effective thermal conductivity, the nonevacuated and evacuated cases exhibit a similar dependence on temperature resulting from thermal radiation. As concluded earlier, the threefold difference results from the thermal convection. The equivalent Nusselt number is about 7, which corresponds to a Rayleigh number of about 10^5 .

Therefore, despite the fact that theory predicts a Nusselt number of unity for the condition that existed in the test sample, instability and perturbation could cause and perpetuate thermal convection within the described insulation.

3. Twenty-four-plate Results

This configuration is identical to the one described in Sec. 2 above with variance in the total number of stainless steel plates used. Consequently, increasing the number of plates to 24 reduces the interstitial gap to

0.5 in. (12.7 mm). The normalized ratio of heat flux to temperature difference versus the hot-face temperature is plotted in Fig. 114. Two notable features

Fig. 114. Normalized Q/DT vs Temperature for 24 Multiplates. Conversion factor: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$.

0.5 in. (12.7 mm), thermal convection is reduced to only gaseous conduction, characterized by a unity Nusselt number. Reasonable agreement is shown between analytical and experimental results. The effective thermal conductivity almost doubles its values as the average temperature is increased from 100 to 400 $^{\circ}\text{F}$ (38 to 204 $^{\circ}\text{C}$). Both analytical and experimental results can be accurately represented by linear approximations.

4. Twelve-screen Results

The 12-screen-plate insulation evolves from the 12-plate configuration. Layers of screen materials were used to fill the gaps between plates. Results of the ratio of normalized heat flux to temperature as a function of hot-face temperature are plotted in Fig. 116. Similar to the results already been discussed, the evacuated and nonevacuated data both display the same sort of linear dependence on temperature. However, there is a small difference, about 10%, between data collected under different conditions. This persists uniformly over the range of temperature that is of interest.

The effective thermal conductivity is plotted against average temperature in Fig. 117. The 10% disparity between the two sets of data suggests that the presence of the screens between layers of metallic plates does indeed provide a more stable configuration in which thermal convection is reduced. However, the screen layers stop short of completely eliminating convection.

distinguish this set of data from previously mentioned results. In the present case, a slight distinction can be made between the evacuated and the nonevacuated data. Second, the gradient of these data points reflects a more sensitive dependence on temperature.

In Fig. 115, the effective thermal conductivity is plotted against the average temperature. The smooth curve depicts the theoretical predictions over the temperature range. There is essentially only a 10% difference between both the evacuated and the nonevacuated data.

This implies that, for a gap size of

Fig. 115. Effective Thermal Conductivity vs Temperature for 24 Multiplates. Conversion factors: 1 Btu/h·ft· $^{\circ}\text{F}$ = 1.73 W/m·K; $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$.

The theoretical prediction labeled as $\tau = 1 - \epsilon$ in Fig. 117 is based on the assumption that the screen material uniformly fills the gaps between the plates, and its reflectivity vanishes. Reasonable agreement is obtained between theory and data.

Fig. 116. Normalized Q/DT vs Temperature for 12 Screen Plates. Conversion factor: $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$.

Fig. 117. Effective Thermal Conductivity vs Temperature for 12 Screen Plates. Conversion factors: $1 \text{ Btu/h}\cdot\text{ft}\cdot^{\circ}\text{F} = 1.73 \text{ W/m}\cdot\text{K}$; $^{\circ}\text{C} = (^{\circ}\text{F} - 32)/1.8$.

A more realistic approach is to characterize the screen as a semi-transparent material with the transmissivity τ and a nonvanishing reflectivity ρ . Analytical development was presented in Chapter III. Results arrived at using this model are also shown in Fig. 117. The curve represents the effective thermal conductivity of an insulation with 1-in. (25.4-mm) gap size filled with screen material of four-mesh 0.047-in. (1.19-mm)-dia wire. Conduction and thermal effects related to the presence of the gas within the test sample are neglected. The emissivity of the screen material is assumed to be 0.3, and the transmissivity is 0.66. The latter constant corresponds to the fraction of open area as supplied by the manufacturer (Cambridge Wire Cloth). Good agreement can be noted. The equivalent heat-transfer coefficient for the nonevacuated case is about 0.22, which gives rise to a Nusselt number slightly above unity. Consequently, for the 12-screen nonevacuated geometry, there is only a very limited degree of natural-convection heat transfer within the insulation if it is at all present. This again confirms analytical prediction in the onset of natural convection for screen materials (see Chapter III).

The choice of the transmissivity is based on the fact that for one layer of four-mesh; 0.047-in. (1.19-mm)-dia screen, the fraction of open area is 0.66. If another layer is placed over it, the total projected open area is reduced, but the effective open area is still 0.66. Since the nature of the thermal radiation is assumed to be diffused, the transmissivity is therefore chosen to be that of the fraction of the effective open area.

5. Twenty-four-screen Results

The 24-screen geometry is the extended configuration to the 24 plates as the 12 screens are to the 12 plates. The major differences between the two screen-filled systems are (1) the number of plates and the number of screen layers employed, and (2) the gap size between successive plates. Instead of the layers of screen material, five sheets were used to comprise the interstitial material. Results of the ratio of normalized heat flux to temperature difference are plotted as a function of hot-face temperature in Fig. 118. In

Fig. 118. Normalized Q/DT vs Temperature for 24 Screen Plates. Conversion factor: $^{\circ}C = (^{\circ}F - 32)/1.8$.

addition to the evacuated and nonevacuated data, in which the nonevacuated data imply an air-filled atmospheric environment, test results of the sample obtained in an argon atmosphere are also presented. Unlike the test mentioned earlier, the argon-filled test utilized 5 in. (127 mm) of liquid sodium as a heat source. Analytically (as confirmed by results plotted in Fig. 118), aside from the negligible difference between the two gases, alteration in the heat source should not have rendered any changes in the thermal-physical properties of the sample.

Several of the familiar features mentioned above recurred in this set of data, of which the linear characteristic of both cases is obvious. The data points seem to be scattered a little more than any of the ones mentioned. The value of the slope shared by these cases indicated their dependence on thermal radiation. Figure 119 shows the effective thermal conductivity as a function

Fig. 119
Effective Thermal Conductivity vs Temperature for 24 Screen Plates. Conversion factors: $1 \text{ Btu/h}\cdot\text{ft}\cdot^{\circ}F = 1.73 \text{ W/m}\cdot\text{K}$; $^{\circ}C = (^{\circ}F - 32)/1.8$.

of the average temperature. The disparity between the evacuated and the non-evacuated results lies within the estimate prediction equivalent to a Nusselt number of unity. Therefore, this further demonstrates that screen-type interstitial materials are capable of providing thermally stable conditions under which convective heat transfer can be ensured to be only of the conductive nature. Another evidence that supports the above conclusion can be derived from the results of the argon filled test sample. The Rayleigh number is defined as

$$Ra = \frac{L^3 \rho^2 C_p \beta \Delta T}{k \mu} \quad (123)$$

The ratio of the Rayleigh number of air to that of argon is given by

$$\eta = \frac{Ra_{air}}{Ra_{Ar}} = \frac{L_{air}^3 \rho_{air}^2 C_{p, air}}{L_{Ar}^3 \rho_{Ar}^2 C_{p, Ar}} \frac{K_{Ar} \mu_{Ar}}{K_{air} \mu_{air}}, \quad (124)$$

assuming the product $\beta \Delta T$ to be the same for both gases. At 100°F (38°C), for η to be unity, the ratio of characteristic lengths is

$$\frac{L_{Ar}}{L_{air}} = 0.94. \quad (125)$$

This implies that, if argon is used, a 6% reduction of the characteristic length is needed to ensure the equivalent Rayleigh number as in an air environment. Conversely, if L_{Ar}/L_{air} is maintained constant, the Rayleigh-number ratio will be affected by 17%. Therefore, results obtained under the argon atmosphere will indicate that the same configuration is now more susceptible to the onset of natural convection as characterized by the 17% increase in Rayleigh number. Little difference is noted for the argon-related test result.

6. Twenty-four Screens with Sodium and with Oxide

After the tests of which the results are presented in the section above, the same test samples were immersed in hot sodium. The assembly has then drained, and related measurements were made. The detailed experimental procedures are discussed in Chapter IV. Two different levels of immersion were performed. Oxidation of the sodium adhering to the test sample was made possible by purging the initially argon-filled test vessel with dry oxygen with a dew point below the temperature of liquid nitrogen. Results of these tests are shown in Fig. 120, where the effective thermal conductivity of both the wetted and oxidized configurations is given as a function of the average temperature.

Little difference can be ascribed to the presence of either sodium or oxidized sodium. Data also indicate a less sensitive dependence on temperature, as can be noted from the relatively small gradient. This comparatively

Fig. 120. Effective Thermal Conductivity vs Temperature for 24 Screen Plates with Sodium and Sodium Oxide. Conversion factors: 1 Btu/h·ft·°F = 1.73 W/m·K; °C = (°F - 32)/1.8.

mild dependence implies the insignificance of thermal-radiation contribution to the overall heat transfer of the insulation. It further exemplifies the increased role of thermal conduction as the thermal resistance of the metallic screens is diminished. Consequently, although data are available only between 400 and 700°F (204 and 371°C), extrapolation can easily be made to include the freezing of sodium [208°F (98°C)] and the maximum temperature expected of the insulation.

The lack of distinctive features between data collected from screens covered with liquid sodium and with

oxidized sodium is truly interesting. Previously, little was known quantitatively of the impacts of the presence of sodium and sodium oxide on thermal-physical properties. Given either of the above circumstances, heat transfer will be enhanced. When sodium, which is well recognized for its superb heat-transfer capability, thoroughly covers the screens, wetting of the screen surfaces would occur. The aspects of wetting were addressed when the experimental procedure of wetting was discussed in Chapter IV. Once the screens are wetted, and if sodium drainage is at substantially high temperatures, only a very thin film of sodium, ~0.001 in. (0.025 mm), will be left adhering to the metal surface and thus contributing to the conductive heat transfer of the screens. The ratio of thermal conductivity of stainless steel to sodium at 400°F (204°C) is 0.2. Moreover, sodium reasonably free of oxides exhibits a lower emissivity value, ~0.10-0.15, than stainless steel, which incidentally will affect the emissivity of the metallic composite and cause a reduction in thermal-radiation transfer.

On the other hand, when the sodium is allowed to be oxidized by oxygen, a coating of principally sodium oxide (Na_2O) and possibly some sodium superoxide (Na_2O_2) will appear. However, this process is self-limiting in that, once an oxide layer is formed, it prevents any further oxidation of the sodium below. It has been established that the emissivity of sodium oxide is in the order of 0.8-0.9, which is substantially higher than that of stainless steel or sodium. However, this change in radiation property is not reflected in data presented in Fig. 120. There is neither an increase of the effective thermal conductivity in the absolute sense nor any change in the gradient as a function of temperature which signifies the nonlinear behavior of thermal radiation. One possible explanation for this is that the sodium oxide film is nonuniform and extremely thin so that its influence on thermal-radiation transfer is diminished greatly. The presence of this sodium oxide layer would have been caused by the insufficient amount of oxygen and the length of time to

which the insulation is exposed. An alternative explanation to this behavior could have stemmed from the fact that a complete sodium oxide layer is formed, but then later, as temperature is elevated, the oxide layer is dissolved by the sodium underneath. Figure 48 shows the solubility curve of sodium monoxide (Na_2O) in sodium as a function of temperature. For a temperature change of 700°F (389°C), the solubility changes by a hundredfold, which could have provided the condition under which the observed results could be explained. All evidence indicates that the latter explanation is much more plausible.

Analytical prediction based on the model proposed in Chapter III is depicted in Fig. 120. It is assumed that sodium displays an emissivity of 0.15 and that a 1-mil (0.001-in.) sodium thickness adheres to the insulation composite. Reasonably good agreement below 750°F (399°C) is realized. The relative ratio between conduction and radiation is found to be one to two. This is unprecedented, even for the honeycomb assembly.

B. Comparisons between Types of Insulation and Published Data

1. Comparison between Various Types of Insulation

In this section, the various proposed design insulations are compared. Different modes of heat transfer are identified, and their relative significance with regard to the overall thermal-physical property is discussed. Direct comparison of this nature is possible because each experimentally tested type of insulation is based on the fundamental concept of either 12 or 24 plates.

Figures 112 and 115 show that there is a major difference in effective thermal conductivity between 24 and 12 plates. This pertains not only to the nonevacuated but also to the evacuated data as well. The 24-plate geometry indicates that the small gap between plates is vital in the suppression of natural convection. A substantial margin will have to be allowed in designing the gap dimension to accommodate perturbations that could render the configuration thermally unstable. Significant consequence is implied from the figures. It can be seen from the 12-plate nonevacuated data, where, despite the analytically proven stable condition, thermal convection increases the effective conductivity almost threefold. The evacuated data provide the means by which the significance of thermal radiation can be isolated and studied. As the number of reflective plates is doubled, both theory and data reveal that almost the same magnitude of increase is realized in the overall conductance. It becomes much more significant at higher temperatures. This evidence lends itself to the conclusion that, as an insulation, the 24-plate arrangement is superior to its 12-plate counterpart.

One major reason why the 12-plate configuration is not as compatible as the 24-plate arrangement is that it is highly susceptible to the onset of thermal convection. However, this can be eliminated through the use of screen material placed between the gaps. Comparisons between Figs. 114 and 118, show

that the presence of the screen layers is able to reduce convection to that of only conduction. At low temperatures, the difference in thermal conductivity is ascribed to the metallic conduction of the screens. At higher temperatures the nonlinear increase is due to thermal radiation of the same material; for example, at 500°F (260°C), a 30% increase is noted. Therefore, it is clear that screens will help to mitigate a thermally unstable condition, and yet, their presence contributes to higher thermal-radiation heat transfer, especially at elevated temperatures.

The logical conclusion thus far would seem to be the 24 screen plates. If comparisons are made between the 24-plate arrangement and its screen counterpart, as depicted in Figs. 116 and 120, the above contention is well supported. Both arrangements show little effects of thermal convection, thereby ensuring a solely conductive and radiative medium. Thermal radiation from the reflective plates as well as from the screen layers of the 24 plates is still higher; at 500°F (260°C), the thermal-conductivity increase is from 0.05 to 0.07, a 40% increase. Metallic screen conduction is also evident at low temperature, despite its relatively minor contribution. Generally speaking, the 24 screen plates have slightly better overall heat-transfer characteristics than the 24 plates.

For a 24- or a 12-screen-plate system, except for the number of reflective plates employed, the number of layers of screens used is identical. Therefore, only two parameters enter into consideration; one is the number of reflective plates used, and the other is the plausible change in thermal properties resulting from changing the transmissivity of the layers. The first parameter is assessed through a comparison of the 12 plates to the 24 plates. Therefore, changes observed when the 12 screen plate is compared with its 24-screen counterpart can only be ascribed to the second parameter. Comparison between the 12-screen plates and the 24-screen plates reveals that the reduction in effective thermal conductivity can be accounted for by the increase in reflective plate. This indicates that the number of screen layers employed under this configuration is quite independent of the transmissivity, which supports the analytical model reported earlier. This implies that the difference in the thermal-radiation contribution to heat transfer from screens between the two arrangements is negligible.

Since the honeycomb composite is based on the concept of 24 plates, comparison must first be made with that configuration. As is apparent from the contrast between Figs. 112 and 116, the honeycomb exhibits a much higher thermal-conductivity value than the 24 plates. Both designs indicate that only conduction and radiation prevail. Significant metallic conduction from the honeycomb cell wall accounts for the almost threefold difference at low temperatures. Major effects from thermal radiation arising from the cell wall are also evident at higher temperatures. At 650°F (343°C), the honeycomb-effect conductivity has doubled in value. Around 500°F (260°C), the ratio of the thermal conductivities of the two geometries is above two. Hence, from a strictly thermal-insulation standpoint, the 24-plate design is far more attractive than the honeycomb composite.

When the 24 screen plates are compared with the honeycomb assembly, the 24 screen plates exhibit a better overall thermal performance. At reduced temperatures, heat transfer by means of metallic conduction through the 2-mil (0.051-mm)-thick honeycomb cell wall surpasses heat transfer through the screen layers. As temperature is increased, a similar nonlinear dependence of thermal conductivity on temperatures is observed that is characteristic of the increasing role of thermal radiation.

Fig. 121. Effective Thermal Conductivity vs Temperature for Screen Plates, Composite Insulation by Lemerrier et al. Conversion factors: 1 Btu/h·ft·°F = 1.73 W/m·K; °C = (°F - 32)/1.8.

The effects of sodium and sodium oxide on thermal insulation can be recognized by the contrasting results depicted in Figs. 120 and 121. Between temperatures of 300 and 500°F (149 and 260°C), the presence of sodium caused the effective thermal conductivity to double its value. Since both sets of data indicate strong dependence on metallic conduction at low temperatures, this twofold increase in conductivity is fairly uniform over the described temperature range. At higher temperatures a more substantial influence of thermal radiation reduces the thermal-conductivity disparity to within 50%. Based on the model proposed, the difference will eventually vanish at temperatures beyond the interest of LMFBR design criteria.

As discussed in the previous section, the sodium oxide formed after the repeated purging of the test vessel

with oxygen appears to have been dissolved by the sodium underneath it. Despite the lack of experimental data on the effects of sodium oxide on thermal conductivity, the theory proposed to describe the 24 screen plate has demonstrated that good agreement can be obtained for sodium-wetted conditions. Hence, if the model is applied to a sodium oxide environment, its impact can be evaluated accordingly.

2. Comparison with Other Results

As mentioned in Chapter II, very little experimental or theoretical data are available for multilayer composite insulation, especially under sodium

environment. For the simple 12- and 24-plate arrangements, analytical methods to evaluate them are well established. Most of the literature today addresses the concept of honeycomb with anticipated solar applications. Little information is known on metallic honeycombs. This is also true for the screen approaches, except for a recent report by Lemerrier et al. (1976) on their measurement of a proposed screen-type insulation considered to be used in the SuperPhenix LMFBR. Results by Lemerrier et al. are shown in Fig. 121. Unfortunately no detailed description was given concerning the insulation composite.

When the results of Lemerrier et al. are compared with those for the 12 and 24 screen plates, the latter configuration shows good agreement. Lemerrier's data indicate principal conductive heat transfer for temperature below 300°F (149°C), and gradually the effects of thermal radiation become prominent. Agreement is noted to be within 5%. Unlike the 24 screens, the 12-screen plates show a stronger dependence on thermal radiation, and metallic-conduction heat transfer is less evident.

Fig. 122

Comparison of Effective Thermal Conductivity of Honeycomb by Rohr and by This Study. Conversion factors: 1 Btu/h·ft·°F = 1.73 W/m·K; °C = (°F - 32)/1.8.

Comparison can be made between the French data and the analytical model developed in Chapter III for screen-type insulations. Theory predicts an asymptotic value for metallic condition of about 0.025 Btu/h·ft·°F (0.043 W/m·K), whereas data show a measured value of 0.035. The rise in thermal conductivity above 300°F (149°C) is more abrupt but gradual for the French data, and it attains a value of 0.1 at ~780°F (416°C). Model prediction from this study shows a thermal-conductivity value of 0.1 at about 700°F (371°C). The lack of detailed information on the French test sample makes further comparison difficult.

Rohr Industries, the manufacturer of the honeycomb core used in this investigation, provides some preliminary estimates of metallic honeycomb thermal properties in their literature. Information is shown in Fig. 122 where the thermal conductivity is plotted as a function of temperatures. The data depicted are for a 3/16-in. (4.76-mm) cell and a 2-mil (0.05-mm) core thickness. Since the cell size is reduced in halves, there is approximately a onefold increase in the linear length in core materials used. At low temperatures, where thermal conduction prevails, this alteration would account for the 100% increase in the reported results. The lack of thermal radiative properties of the Rohr Industries sample renders comparison with the model at elevated temperature difficult.

VII. CONCLUSIONS AND RECOMMENDATIONS

Five different designs of metallic insulation were experimentally studied and respective analytical models proposed. Three sets of experimental conditions were chosen under which measurements were made: evacuated, nonevacuated, and sodium-wetted environments. Because of carefully selected geometries, direct comparison between data is possible. All test samples exhibit negligible effects due to thermal convection. From a purely thermal-physical-property standpoint, the 24 plates are by far the best. Besides its exceedingly simple arrangement, it is also the most economical. Then, there are the 24 screen plates, which exhibit high thermal resistance at low temperatures. The increasing dominance by thermal radiation is very gradual. The honeycomb and the 12 screen plates are equally compatible in the temperature range of interest. The honeycomb is most susceptible to conduction at low temperatures. The 12 plates rank last among all the types of insulation tested. This is primarily due to the fact that natural convection occurred within the insulation.

However, a thermal insulation which possesses superior thermal characteristics as an insulation could become useless if it is highly susceptible to sodium transport and condensation. This is one reason why there are reservations in using multiplates as LMFBR insulation. Both the honeycomb and the screen plates indicate good to excellent thermal properties, and they also provide barriers whereby sodium-aerosol transport is severely limited and localized.

Due to the difficulties involved in studying sodium contamination of the insulation, immersion of a selected thermal insulation, namely the 24 screen plates, in sodium was introduced. Results show that there is a 100% increase in thermal conductivity. Therefore, over the lifetime of the reactor, if the proposed insulation is wetted with sodium, the thermal conductivity can be expected to double its value. However, a potentially more adverse condition, i.e., the sodium oxide contamination, was not thoroughly studied. If the residual sodium adhering to the insulation is oxidized to sodium oxide, the emissivity of the medium will be altered. This could lead to increased heat transfer through the insulation. The important parameters in designing the various types of insulation were identified.

Good agreement was noted between the models proposed and the experimental data. Therefore, the models can subsequently be used not only to predict the thermal properties of the insulation under normal conditions but also to predict when the insulation is contaminated with sodium or sodium oxide. These experimental results can henceforth be used as design guidelines for the thermal insulation required for larger LMFBR systems.

A few areas deserve additional attention, namely, the impact of sodium oxide on the thermal properties of insulation, and the manner in which the insulation is filled with sodium or sodium oxide. Their influence will be

geometry-dependent. Moreover, long-time study on vapor transport within a particular type of insulation is informative with regard to the degradation rate of the thermal insulation over the anticipated lifetime of the reactor system. Finally, based on information collected in this study on the proven ability of honeycomb and screen materials to reduce or eliminate thermal convections, studies can be extended to include the applications of these proposed types of insulation near and around penetrations and components.

APPENDIX A

Uniqueness of the Gebhart and Hottel Solutions

The solutions obtained from both the Gebhart and the Hottel methods are a direct consequence of solving an inhomogeneous matrix equation of the form

$$AX = C. \quad (A.1)$$

It has also been demonstrated that the solution x exists and is unique if and only if the solution to the homogeneous equation is the trivial solution (Stakgold, 1967)

$$AX = 0. \quad (A.2)$$

In other words, if x does have a nontrivial solution, then the determinant of A must be zero.

For the Gebhart method, a homogeneous equation implies that the absorptance of the surface to which all the energy is directed is zero and the set of equations can be written as

$$\left. \begin{aligned} (\rho_1^F - 1)G_{1j} + \rho_2^F G_{2j} + \rho_3^F G_{3j} + \dots + \rho_n^F G_{nj} &= 0; \\ \rho_2^F G_{1j} + (\rho_2^F - 1)G_{2j} + \rho_3^F G_{3j} + \dots + \rho_n^F G_{nj} &= 0; \\ \rho_n^F G_{1j} + \rho_2^F G_{2j} + \rho_3^F G_{3j} + \dots + [\rho_n^F - 1]G_{nj} &= 0. \end{aligned} \right\} \quad (A.3)$$

Since each equation of Eqs. A.3 is linearly independent, it is true if and only if all the coefficients of G_{ij} in that equation vanishes. Conversely, if the coefficients are not zero, then all the G_{ij} 's will have to vanish to satisfy the equality. It is evident that not all the coefficients will vanish for a particular configuration. Therefore, it shows that the Gebhart matrix has indeed only the trivial solution to its homogeneous equation. It also follows that the solution to the inhomogeneous equation exists and is unique. Physically, it is quite obvious that if the absorptance of the surface of concern j vanishes, the fraction of energy emitted by any surface i that is absorbed by j , G_{ij} , should vanish since no energy is reaching surface j . In other words, the vector G_{nj} is identically zero. This further implies the solution to the inhomogeneous equation exists and is unique.

Similarly, the equation used by Hottel, shown in Eqs. A.4, indicates that the homogeneous equation describing a situation in which the emittance of the surface of concern (in this case surface 1) is zero will render the last terms of all the Eqs. A.4 zeros.

$$\left. \begin{aligned} \left(A_1 F_{11} - \frac{A_1}{\rho_1} \right) J_1 + A_2 F_{21} J_2 + A_3 F_{31} J_3 + \dots + A_n F_{n1} J_n &= -A_1 F_{11} \epsilon_1 \rho_1 E_{G_1}; \\ A_1 F_{12} J_1 + \left(A_2 F_{22} - \frac{A_2}{\rho_2} \right) J_2 + A_3 F_{32} J_3 + \dots + A_n F_{n2} J_n &= -A_1 F_{12} \epsilon_1 \rho_1 E_{G_1}; \\ A_1 F_{1n} J_1 + A_2 F_{2n} J_2 + \dots + \left(A_n F_{nn} - \frac{A_n}{\rho_n} \right) J_n &= -A_1 F_{1n} \epsilon_1 \rho_1 E_{G_1}. \end{aligned} \right\} \quad (A.4)$$

The linear independence of these equations assures that all the coefficients of J_i are zero or conversely that all the J_i 's will have to be zero. Consequently, the existence and uniqueness of the inhomogeneous solution follow. The assumption invoked by Hottel (McAdams, 1954) is that all but one surface have 0 K temperature, and that the emittance of that surface vanishes. This implies that that surface is not transferring any energy into the system, and the enclosure is essentially at 0 K temperature equilibrium. This further signifies that there will not be any energy transfer between surfaces, and therefore the solution of J_i to such a system must also vanish.

APPENDIX B

Numerical Application of the Gebhart Method
and the Modified Gebhart Method

A cubical enclosure is depicted in Fig. 26 in which the base of the cube is maintained at about 1500°F (816°C).

$$\left. \begin{array}{lll} F_{11} = 0 & F_{12} = 0.2 & F_{13} = 0.8 \\ F_{21} = 0.2 & F_{22} = 0 & F_{23} = 0.8 \\ F_{31} = 0.2 & F_{32} = 0.2 & F_{33} = 0.6 \\ \epsilon_1 = 0.7 & \epsilon_2 = 0.7 & \epsilon_3 = 0 \\ \rho_1 = 0.3 & \rho_2 = 0.3 & \rho_3 = 0 \end{array} \right\} . \quad (\text{B.1})$$

For the Gebhart Method, the matrix equation becomes

$$\begin{pmatrix} -1 & 0.06 & 0.8 \\ 0.06 & -1 & 0.8 \\ 0.06 & 0.06 & -0.4 \end{pmatrix} \begin{pmatrix} G_{1j} \\ G_{2j} \\ G_{3j} \end{pmatrix} = -\epsilon_j \begin{pmatrix} F_{1j} \\ F_{2j} \\ F_{3j} \end{pmatrix} . \quad (\text{B.2})$$

The inverse of the matrix can be expressed as

$$A^{-1} = \text{c}\tilde{\text{o}}\text{f}(A), \quad (\text{B.3})$$

where $\text{c}\tilde{\text{o}}\text{f}$ is the transpose of the cofactor. The inverse of the matrix in Eq. B.2 can be written as

$$\begin{pmatrix} G_{1j} \\ G_{2j} \\ G_{3j} \end{pmatrix} = \frac{0.7}{0.297} \begin{pmatrix} 0.352 & 0.072 & 0.848 \\ 0.072 & 0.352 & 0.848 \\ 0.0636 & 0.0636 & 0.9964 \end{pmatrix} \begin{pmatrix} F_{1j} \\ F_{2j} \\ F_{3j} \end{pmatrix} . \quad (\text{B.4})$$

For $j = 1$,

$$A^{-1} \begin{pmatrix} 0 \\ 0.2 \\ 0.2 \end{pmatrix} = \begin{pmatrix} G_{11} \\ G_{21} \\ G_{31} \end{pmatrix} = \begin{pmatrix} 0.434 \\ 0.566 \\ 0.5 \end{pmatrix} ; \quad (\text{B.5})$$

for $j = 2$,

$$A^{-1} \begin{pmatrix} 0.2 \\ 0 \\ 0.2 \end{pmatrix} = \begin{pmatrix} G_{12} \\ G_{22} \\ G_{32} \end{pmatrix} = \begin{pmatrix} 0.566 \\ 0.434 \\ 0.5 \end{pmatrix}; \quad (\text{B.6})$$

for $j = 3$,

$$A^{-1} \begin{pmatrix} 0.8 \\ 0.8 \\ 0.6 \end{pmatrix} = \begin{pmatrix} G_{13} \\ G_{23} \\ G_{33} \end{pmatrix} = \begin{pmatrix} 0 \\ 0 \\ 0 \end{pmatrix}; \quad (\text{B.7})$$

$$q_{\text{net}1} = \sum_{i=1}^n G_{i1} \epsilon_i A_i E_{bi} - \epsilon_1 A_1 E_{b1}; \quad (\text{B.8})$$

$$q_{\text{net}1} = G_{11} \epsilon_1 A_1 E_{b1} + G_{21} \epsilon_2 A_2 E_{b2} + G_{31} \epsilon_3 A_3 E_{b3} - \epsilon_1 A_1 E_{b1}; \quad (\text{B.9})$$

$$q_{\text{net}2} = G_{12} \epsilon_1 A_1 E_{b1} + G_{22} \epsilon_2 A_2 E_{b2} + G_{32} \epsilon_3 A_3 E_{b3} - \epsilon_2 A_2 E_{b2}; \quad (\text{B.10})$$

$$q_{\text{net}3} = G_{13} \epsilon_1 A_1 E_{b1} + G_{23} \epsilon_2 A_2 E_{b2} + G_{33} \epsilon_3 A_3 E_{b3} - \epsilon_3 A_3 E_{b3}; \quad (\text{B.11})$$

$$q_{\text{net}1} = \epsilon_1 A_1 E_{b1} (-G_{12}) + G_{12} \epsilon_1 A_1 E_{b2}; \quad (\text{B.12})$$

$$q_{\text{net}1} = G_{12} \epsilon_1 A_1 (E_{b2} - E_{b1}) = 0.3962 A_1 (E_{b2} - E_{b1}); \quad (\text{B.13})$$

where the equalities of reciprocity and identity are used.

$$q_{\text{net}2} = 0.3962 A_2 (E_{b1} - E_{b2}); \quad (\text{B.14})$$

$$q_{\text{net}3} = 0. \quad (\text{B.15})$$

For the Modified Gebhart Method, the matrix equation is

$$\begin{pmatrix} F_{11}\rho_1 - 1 & \frac{\epsilon_1}{\epsilon_2} F_{12}\rho_2 & \frac{\epsilon_1}{\epsilon_3} F_{13}\rho_3 \\ \frac{\epsilon_2}{\epsilon_1} F_{21}\rho_1 & F_{22}\rho_2 - 1 & \frac{\epsilon_2}{\epsilon_3} F_{23}\rho_3 \\ \frac{\epsilon_3}{\epsilon_1} F_{31}\rho_1 & \frac{\epsilon_3}{\epsilon_2} F_{32}\rho_2 & F_{33}\rho_3 - 1 \end{pmatrix} \begin{pmatrix} G_{1j} \\ G_{2j} \\ G_{3j} \end{pmatrix} = -\epsilon_j \begin{pmatrix} \epsilon F_{1j} \\ \epsilon_2 F_{2j} \\ \epsilon_3 F_{3j} \end{pmatrix}. \quad (\text{B.16})$$

If $\epsilon_3 = 0.001$, the above equation can be written as

$$\begin{pmatrix} -1 & 0.06 & 560 \\ 0.06 & -1 & 560 \\ 8 \times 10^{-5} & 8 \times 10^{-5} & -0.4 \end{pmatrix} \begin{pmatrix} G_{1j} \\ G_{2j} \\ G_{3j} \end{pmatrix} = -\epsilon_j \begin{pmatrix} 0.7F_{1j} \\ 0.7F_{2j} \\ 0.001F_{3j} \end{pmatrix}. \quad (\text{B.17})$$

The inverse becomes

$$\frac{-1}{0.2968} \begin{pmatrix} 0.352 & 0.072 & 594 \\ 0.072 & 0.352 & 594 \\ 9 \times 10^{-5} & 9 \times 10^{-5} & 0.9964 \end{pmatrix}. \quad (\text{B.18})$$

If $j = 1$,

$$A^{-1}(-0.7) \begin{pmatrix} 0 \\ 0.14 \\ 2 \times 10^{-4} \end{pmatrix} = \begin{pmatrix} G_{11} \\ G_{21} \\ G_{31} \end{pmatrix} = \begin{pmatrix} 0.304 \\ 0.3968 \\ 5 \times 10^{-4} \end{pmatrix}; \quad (\text{B.19})$$

if $j = 2$,

$$A^{-1}(-7) \begin{pmatrix} 0.14 \\ 0 \\ 2 \times 10^{-4} \end{pmatrix} = \begin{pmatrix} G_{12} \\ G_{22} \\ G_{32} \end{pmatrix} = \begin{pmatrix} 0.3968 \\ 0.304 \\ 5 \times 10^{-4} \end{pmatrix}; \quad (\text{B.20})$$

if $j = 3$,

$$A^{-1}(-0.001) \begin{pmatrix} 0.56 \\ 0.56 \\ 6 \times 10^{-4} \end{pmatrix} = \begin{pmatrix} G_{13} \\ G_{23} \\ G_{33} \end{pmatrix} = \begin{pmatrix} 2 \times 10^{-3} \\ 2 \times 10^{-3} \\ 2 \times 10^{-6} \end{pmatrix}; \quad (\text{B.21})$$

$$q_{\text{net}2} = \sum_{i=1}^n G_{i1} A_i E_{bi} - \epsilon_1 A_1 E_{b1}; \quad (\text{B.22})$$

$$q_{\text{net}1} = 0.396 A_1 (E_{b2} - E_{b1}); \quad (\text{B.23})$$

$$q_{\text{net}2} = 0.396A_2(E_{b1} - E_{b2}); \quad (\text{B.24})$$

$$q_{\text{net}3} = 0. \quad (\text{B.25})$$

To show the identity relation, since

$$G_{ij} = \epsilon_1 \epsilon_j F_{ij} + \sum_{i=1} \epsilon_1 F_{1j} \rho_i \frac{G_{ij}}{\epsilon_i}, \quad (\text{B.26})$$

$$\sum_j \frac{A_k G_{kj}}{\epsilon_k} = \sum_{j=1} \epsilon_j A_k F_{kj} + \sum_{j=1} \sum_{i=1} A_k F_{ki} (1 - \epsilon_i) \frac{G_{ij}}{\epsilon_i}. \quad (\text{B.27})$$

Sum the above equation over k, using

$$\sum_k F_{jk} = 1 \quad (\text{B.28})$$

and

$$\sum_j \sum_i A_i G_{ij} = \sum_j \epsilon_j A_j. \quad (\text{B.29})$$

From reciprocity,

$$\sum_j A_j \sum_i G_{ji} - \epsilon_j = 0 \quad (\text{B.30})$$

or

$$\frac{1}{\epsilon_j} \sum_i G_{ji} = 1. \quad (\text{B.31})$$

APPENDIX C

Error Analysis of Temperature Measurements

Two kinds of thermocouples are used throughout the experiment: stainless steel sheath and asbestos-glass-impregnated wires. All in-vessel temperature measurements are made with stainless steel sheath thermocouples; ex-vessel measurements are made with insulated thermocouple wires, except for the base heating elements. The procedures and methods by which these thermocouples are secured in their respective locations are described in Chapter IV and will not be repeated here.

To estimate the temperature error incurred due to the presence of a thermocouple, different models are established to represent different situations. The first one, shown in Fig. C.1, involves an insulated thermocouple wire welded onto the surface of a massive solid. Heat-conduction theory [Grober (1961)], provides an expression for the heat flow rate passing through an isothermal circular area situated on the otherwise insulated surface of a semi-infinite solid. The heat-transfer rate is

$$Q_1 = 4r_1 K_s (T_{tc} - T_s), \quad (C.1)$$

where T_{tc} is the temperature of the circular area, K_s is the conductivity of the solid, T_s is the temperature of the solid at infinity, is the conductivity of the solid,

$$r_1 = r_w \sqrt{2}, \quad (C.2)$$

and r_w is the radius of the wires and for the type of insulated thermocouple used.

Fig. C.1. Insulated Thermocouple Attached to Massive Solid

After the system has acquired thermal equilibrium, energy conservation requires that the amount of heat conducted through the circular area be equal to the amount conducted into or away from the surface by the thermocouple leads. Based on the fin theory, Eckert and Drake (1959) and Schneider (1955),

$$Q_2 = \frac{KA}{R} (T_\beta - T_{tc}), \quad (C.3)$$

and energy balance,

$$Q_1 = Q_2. \quad (C.4)$$

For the application,

$$KA = (R_{w1} + R_{w2}) A_w. \quad (C.5)$$

Here R_{w1} and R_{w2} are thermal conductivities of the leads, A_w is the cross-sectional area of the wire, T_β is the temperature of the bulk environment, and R is the thermal resistance of heat transfer between the wires and their immediate environment and is expressed as

$$R = \frac{1}{2\pi h R_i} + \frac{\ln \frac{r_i}{r_w}}{2\pi k_i}, \quad (C.6)$$

where k_i is the thermal conductivity of the insulating materials of the wires, h is the heat-transfer coefficient to the fluid environment, and r_i is defined as (see Fig. C.1)

$$r_i = (D_1 + D_2)/4. \quad (C.7)$$

Thus, the result is

$$\frac{T_{tc} - T_s}{T_\beta - T_s} = 1 + \frac{4r_i k_s - 1}{\sqrt{\frac{KA}{R}}}. \quad (C.8)$$

For all the ex-vessel glass-impregnated thermocouples, $T_{tc} - T_s$ and the result is found to be within 1°C. Temperature deviations of such small magnitude are expected, for attempts are made during instrumentation to maintain lead wires at temperatures close to junction temperature. The heat-transfer coefficient used to arrive at the results is based on conduction and radiation effects and their dependence on temperature.

A different model is used to evaluate the temperature error of stainless steel sheath thermocouples welded onto thin members, as shown in Fig. C.2. A detailed treatment of the model is given by Schneider (1955). The temperature error expressed in a nondimensional form is

$$\frac{T_{tc} - T'}{T_\beta - T'} = \left[1 + 2\pi k_s \sigma \epsilon r_1 \frac{K_1(\epsilon r_1)}{K_0(\epsilon r_1)} \frac{1}{C} \right]^{-1}, \quad (C.9)$$

$$C = (2\pi r_{w1} H_{w1} R_{w1} \pi r_{w1}^2)^{1/2} + (2\pi r_{w2} H_{w2} R_{w2} \pi r_{w2}^2)^{1/2}, \quad (C.10)$$

and

$$H_{w1} = \left(\frac{1}{h} + \frac{\delta i}{R_i} \right)^{-1}, \quad (C.11)$$

where K_1 and K_0 denote the modified Bessel functions of the second kind, and h_1 and h_2 represent the heat-transfer coefficients on either side of the thin

plate. The heat-transfer coefficient characterizing heat transfer between the thermocouple wire and its environment is denoted by h ; T' is the equilibrium temperature of the plate in the absence of a thermocouple;

$$T' = \frac{h_1 T_{\beta 1} + h_2 T_{\beta 2}}{h_1 + h_2}. \quad (C.12)$$

Fig. C.2

Insulated Thermocouple Attached to Thin Member

The temperature error is plotted against the ratio of plate to thermocouple conductance in Fig. C.3. However, in order to evaluate the above quantities, various heat-transfer coefficients must be known. Given the configurations in which the sheath thermocouples are instrumented, only conduction and radiation prevail. The situation is further simplified for evacuated systems where the only heat-transfer means is by radiation. For an isothermal wall, thermal-radiation heat transfer within an enclosure can be estimated by writing

$$Q = \epsilon \sigma (T^4 - T_w^4), \quad (C.13)$$

where ϵ is the emissivity of the material, T_w is the enclosure wall temperature, and σ is the Stefan-Boltzmann constant. Recast into a different form,

$$Q = h_{\text{rad}} (T - T_w), \quad (C.14)$$

where

$$h_{\text{rad}} = \frac{\epsilon \sigma (T^4 - T_w^4)}{T - T_w}. \quad (C.15)$$

If the temperature difference $T - T_w$ is significantly smaller than T_w , the equation for h_{rad} can be rewritten as

$$h_{\text{rad}} \approx 4\epsilon \sigma T_w^3 \quad (C.16).$$

for thermocouples monitoring the insulation sample temperatures. The error is limited to within a few degrees Celsius.

Fig. C.3. Temperature Errors vs Thermocouple Conductance

APPENDIX D

Recorder Calibrations

Two different kinds of recorders are used: the Honeywell Model 15 Universal Multipoint stripe chart record, and a digital multiple-channel readout Model 250B made by Omega Engineering at Stamford, Connecticut. The Model 250B was calibrated at the factory with a precision source, and frequent calibration is not necessary due to the stability and internal accuracy of the meter. However, calibration verification over a range of temperature is made to ensure reliable results according to the owner's manual published by the manufacturer. Good agreement is obtained between the meter and the thermocouple reference tables by Powell (1968).

Model 15 is calibrated at 1000°F (538°C) full scale in a similar manner using a voltage generator. Two set points are chosen: cabinet temperature and 900°F (482°C). Cabinet temperature is measured by using an independent temperature-measuring device, and the recorder is adjusted accordingly. Based on the thermocouple reference tables by Powell, a corresponding voltage of 900°F (482°C) is used to provide the high-temperature signal, where further adjustment is made. Then the low set point is checked again to ensure accuracy. After such a calibration, it is found that at 400°F (204°C), the recorder is accurate within $\pm 1.3^\circ\text{F}$ (0.72°C). Model 15 is then checked against Model 250B, and agreement is obtained within the error limits stated earlier.

APPENDIX E

Network-model Derivations

Two cases of the network model are derived: (1) the $\epsilon + \tau = 1$ derivation, given in Fig. E.1, and (2) $\epsilon + \rho + \tau = 1$, given in Fig. E.2, where ϵ , ρ , and τ , denote the emittance, reflectance, and transmittance of the medium, respectively. The radiosities leaving the top part of the bottom part of the porous medium are represented by J_u and J_D ; G_u and G_D are the irradiances from above and below of the medium; E_{bs} is the black-body emissive power of the medium. If the techniques described in Chapter III are applied, the resultant equations can be interpreted as an Ohm's-Law equation where two network models result. The results of the models and their comparisons with other data are presented in Figs. E.3-E.5.

$$\epsilon + \tau = 1$$

$$J_u = \epsilon E_{bs} + \tau G_D$$

$$J_D = \epsilon E_{bs} + \tau G_u$$

$$q_u = J_u - G_u$$

$$= \frac{1}{\tau} [\epsilon (E_{bs} - J_u) + (J_u - J_D)]$$

Fig. E.1

Derivation of $\epsilon + \tau = 1$

$$\epsilon + \rho + \tau = 1$$

$$J_u = \epsilon E_{bs} + \tau G_D + \rho E_{b1}$$

$$J_D = \epsilon E_{bs} + \tau G_u + \rho E_{b2}$$

$$q_u = J_u - G_u$$

$$= \frac{\epsilon}{\rho} (E_{bs} - J_u) + \frac{\tau}{\rho} (E_{b2} - J_u)$$

Fig. E.2

Derivation of $\epsilon + \rho + \tau = 1$

Fig. E.3. Total Thermal Resistance between Two Plates Separated by a Non-reflective and Nonopaque Medium

Fig. E.4. Total Thermal Resistance between a Black Surface and Its Adjacent Non-reflective and Nonopaque Medium

Fig. E.5. Total Thermal Resistance between Two Plates Separated by a Reflective, Transparent, and Emissive Medium

APPENDIX F

Computer Listing

FORTRAN IV G1 RELEASE 2.0

MAIN

DATE = 79335

15/12/34

```

0001      REAL KEFF,KARG,H,HNEW,KNEW
0002      INTEGER OPTION
0003      COMMON T(200),TP,NNAO
0004      COMMON TEMP,KARG,KEFF,FNAS,FSS,ENA,ESS,ENAO,SIGMA,RHO,U,DNAS,DINS
0005      COMMON G,H,M,TT,DELT,METHOD,OPTION

      C
      C      INITIATE THE ARRAY T AND CONSTANTS
      C
      C      TEMP DENOTES THE COLD FACE TEMP OF INSULATION
0006      TEMP=610.
      C      LARG HAD THE VALUE OF AN EVACUATED SYS
0007      KARG=.002
0008      KEFF=.035
0009      FNAS=.88
0010      FSS=.9936
0011      ENA=.11
0012      ESS=.5
0013      ENAO=.9
0014      SIGMA=.173E-8
0015      RHO=.045
0016      U=8.5E-7
0017      DNAS=1.5
0018      DINS=1.
0019      G=32.2
0020      H=.7
0021      H=36
      C      METHOD IS ALWAYS 1
0022      METHOD=1
      C      OPTION PROVIDES CHOICES W/ OR W/O PLATE THICKNESS
0023      OPTION=0
      C
      C      CALCULATE THE TEMPERATURE OF THE FIRST PLATE
      C
0024      14  CALL GAPI(Q)
0025      CALL PLATE(Q)
0026      115 TR=TT-TEMP
0027      IF (DELT.GE.TR ) GO TO 90
0028      IF (KEFF.GE..7) GO TO 999
0029      GO TO 14
0030      90  WRITE (6,97) TT
0031      97  FORMAT (1H0,2X,E10.5)
0032      DO 101 I=1,M
0033      WRITE (6,95) T(I)
0034      95  FORMAT (1H0,5X,E10.5)
0035      101 CONTINUE
0036      WRITE (6,400) NNAO
0037      400 FORMAT (1H0, 2X, ' NUMBER OF PLATES THAT HAVE SODIUM OXIDE ',I2)
0038      TEMP=TEMP+1.
0039      999 STOP
0040      END

```

FORTRAN IV G1 RELEASE 2.0

GAP

DATE = 79335

15/12/34

```

0001      SUBROUTINE GAP(Q)
0002      REAL KEFF,KARG,H,HNEW,KNEW
0003      INTEGER OPTION
0004      COMMON T(200),TP,NNAO
0005      COMMON TEMP,KARG,KEFF,FNAS,FSS,ENA,ESS,ENAO,SIGMA,RHO,U,DNAS,DINS
0006      COMMON G,H,M,TT,DELT,METHOD,OPTION
0007      C      TP AND TT DENOTE THE SODIUM POOL TEMP
0008      14      TP=1460.
0009      TT=1460.
0010      NNAO=0
0011      DO 10 I=1,200
0012      10      T(I)=0.
0013      15      DO 20 I=1,200
0014      TT=TT-.5
0015      IF (METHOD.EQ.1) CALL HEAT(Q)
0016      IF (NNAO.LT.1) GO TO 18
0017      ESS=ENAO
0018      18      QCONV=H*(TP-TT)
0019      QRAD=SIGMA*FNAS*(TP**4-TT**4)/((1/ENA+1/ESS-1))
0020      QT=QCONV+QRAD
0021      IF (Q.GE.1000) GO TO 999
0022      Y=Q-QT
0023      IF (Y) 30,30,20
0024      20      CONTINUE
0025      C
0026      C      CHECK THE VALUE OF H WITH THE CALCULATED DELTA T
0027      C      ADJUST H IF NECESSARY AND RE-CALCULATE TEMP OF 1ST PLATE
0028      C
0029      30      DELT=TP-TT
0030      C=QT/DELT
0031      CALL CONV(HNEW,DELT,DNAS)
0032      S=ABS((H-HNEW)/H)
0033      R=HNEW-H
0034      IF (S.GE.2) GO TO 999
0035      IF (R) 39,39,40
0036      39      IF (S.LE..05*H) GO TO 40
0037      H=(H+HNEW)/2
0038      GO TO 14
0039      40      PQCONV=QCONV/QT*100
0040      PQRAD=QRAD/QT*100
0041      WRITE (6,75) PQCONV,PQRAD,C
0042      75      FORMAT (1H0,2X,'PQCONV=',E10.3,5X,'PQRAD=',E10.3,5X,'CONDUCTANCE 0
0043      IF ARGON GAP=',E10.4)
0044      RETURN
0045      999      STOP
0046      END

```

FORTRAN IV G1 RELEASE 2.0

PLATE

DATE = 79335

15/12/34

```

0001      SUBROUTINE PLATE(Q)
0002      REAL KEFF,KARG,H,MNEH,KNEH
0003      INTEGER OPTION
0004      COMMON T(200),TP,NNAO
0005      COMMON TEMP,KARG,KEFF,FNAS,FSS,ENA,ESS,ENAO,SIGMA,RHO,U,DNAS,DINS
0006      COMMON G,H,M,TT,DELT,METHOD,OPTION

      C
      C      CALCULATE THE TEMP OF EACH PLATE (36)
      C
0007      T(1)=TEMP
0008      DSS=1./12./2.
0009      IF (OPTION.EQ.0) GO TO 13
      C      DSS IS CHANGED TO A NEW VALUE DUE TO FINITE THICKNESS OF SS PLATE
      DSS=DSS-1./32./12.
0010
0011      13  N=M-1
0012          NN=M-NNAO
0013          DO 50 I=1,N
0014              IF ( OPTION.EQ.0) GO TO 11
0015              TS=T(I)
0016              CALL FOIL(Q,TS,X)
0017              WRITE (6,10) X
0018          10  FORMAT (1H , ' TEMP OF SS ',E10.5)
0019              T(I)=X
0020          11  TS=T(I)+1.5
0021              IF (NN.GE.I) GO TO 64
0022              ESS=ENAO
0023              GO TO 66
0024          64  ESS=.5
0025          66  DO 65 J=1,300
      C      QCOND=(TS-T(I))*KARG/DSS*1.08
      QCOND=(TS-T(I))*2.5
0026
      C      QRAD=SIGMA*FSS*(TS**4-T(I)**4)/(1/ESS+1/ESS-1)
      QRAD=SIGMA*(TS**4-T(I)**4)*.011
0027
0028      QRADT=((TS+T(I))/2.)*4-T(I)**4)*SIGMA*.1
0029      QRAD=QRAD+QRADT
0030      Y=Q-QCOND-QRAD
0031      IF (Y) 60,60,70
0032      60  T(I+1)=TS
0033          WRITE (6,119) TS
0034      119  FORMAT (1H ,2X,E10.5)
0035          WRITE (6,109) Q,QCOND,QRAD
0036      109  FORMAT (1H ,2X,'Q=',E10.4,2X,'QCOND=',E10.4,2X,'QRAD=',E10.4)
0037          IF (TS.GE.TT) GO TO 110
0038          GO TO 50
0039      110  WRITE (6,120) M
0040      120  FORMAT (1H0,' NEED LESS THAN ',I3,' PLATES')
0041          GO TO 111
0042      70  TS=TS+.3
0043      65  CONTINUE
0044      50  CONTINUE
0045          GO TO 112

      C
      C      WITH THE TEMP DIFFERENCE OF ALL THE PLATES, FIND DELTA T IF IT
      C      IS LESS THAN 850 CHANGE K AND ITERATE OTHERWISE PRINT
      C

```

FORTRAN IV G1 RELEASE 2.0

PLATE

DATE = 79335

15/12/34

```

0046      111  IF ( OPTION.EQ.0) GO TO 210
0047          CALL FOIL(Q,TS,X)
0048          WRITE (6,10) X
0049      210  DELT=TS-T(1)
0050          CALL CONH(DELT,KEFF,KNEW,Q,DINS)
0051          GO TO 115
0052      112  IF ( OPTION.EQ.0) GO TO 220
0053          CALL FOIL(Q,TS,X)
0054          WRITE (6,10) X
0055      220  DELT=T(M)-T(1)
0056          CALL CONH(DELT,KEFF,KNEW,Q,DINS)
0057      115  RETURN
0058          END

0001          SUBROUTINE CONV(H,DELT,DNAS)
0002          REAL H,NU,KARG
0003          PR=.66
0004          RHO=.045
0005          KARG=.016
0006          G=32.2
0007          U=8.5E-7
0008          B=1460.
0009          GR=DNAS**3*RHO**2*G*DELT/U**2/B/32.2**2
0010          NU=.173*(GR*PR)**.28
0011          H=HU*KARG/DNAS
0012          WRITE(6,10) H,NU,GR,DELT
0013      10  FORMAT (1H0,5X,'H= ',E10.3,5X,'NU= ',E10.3,5X,'GR= ',5X,E10.3,'DELT=
1',E10.5/)
0014          RETURN
0015          END

0001          SUBROUTINE CON(DELT,KEFF,KNEW,Q,DINS)
0002          REAL KEFF,KNEW
0003          REAL KEFF,KNEW
0004      91  FORMAT (1H0,2X,'DELTA T=',2X,E10.5,2X,'KEFF=',E10.4)
0005          KNEW=Q*DINS/DELT
0006          WRITE (6,10) KNEW
0007      10  FORMAT (1H0,2X,'KNEW=',E10.4)
0008          KEFF=KEFF+0.005
0009          WRITE (6,15) KEFF
0010      15  FORMAT (1H0,2X,'NEW KEFF=',E10.4)
0011          RETURN
0012          END

0001          SUBROUTINE HEAT(Q)
0002          REAL KEFF,KARG,H,HNEW,KNEW
0003          INTEGER OPTION
0004          COMMON T(200),TP,NNAO
0005          COMMON TEMP,KARG,KEFF,FNAS,FSS,ENA,ESS,ENAO,SIGMA,RHO,U,DNAS,DINS
0006          CONHO: G,H,M,TT,DELT,METHOD,OPTION
0007          Q=KEFF*(TT-TEMP)/DINS
0008          RETURN
0009          END

0001          SUBROUTINE FOIL(Q,TS,T)
0002          REAL KSS
0003          KSS=9.4
0004          T=TS
0005          DO 10 I=1,50
0006              T=T+0.01
0007              QCOND=KSS*(T-TS)/(1./32./12.)
0008              Y=Q-QCOND
0009              IF (Y) 20,20,10
0010      20  GO TO 25
0011      10  CONTINUE
0012      25  RETURN
0013          END

```

ACKNOWLEDGMENTS

I am indebted to Mr. A. Amorosi and Dr. H. O. Monson of Argonne National Laboratory (ANL) who introduced me to this problem and have provided encouragement and helpful suggestions throughout this study. Special appreciation is extended to my thesis adviser, Professor B. G. Jones, for his valuable inspiration, advice, encouragement, and guidance on the subject. Numerous discussions with E. A. Spleha, D. J. Quinn, E. S. Sowa, and F. A. Smith on different aspects of the experimental setup and design are duly acknowledged. I am also grateful to R. M. Hodges and H. J. Shepelak for their assistance in the experimental preparation.

Finally, I acknowledge the support and cooperation of the Reactor Analysis and Safety Division, the Materials Science Division, and the Engineering Division of ANL during this work, as well as the financial support by the Nuclear Engineering Program of the University of Illinois during the initial stages of my study.

REFERENCES

- Adrianov, V. N., *Application of the Electric Analogue to the Problem of Radiant Heat Transfer*, Izv. Akad. Nauk SSSR, Otd. Tekh. Nauk, Energ. Avtom. 1 (1959).
- Amorosi, A., et al., *An Overview of Pool-type LMFBRs: General Characteristics*, ANL-76-61 (May 1976).
- Arnold, G. P., et al., *Reactor Development Program Progress Report*, LA-1575, Los Alamos Scientific Laboratory (July 1953).
- Atomics International, *PLBR Pool Report*, TI-707-230-007, Phase A (1978).
- Balkwill, J. K., *Chemical and Physical Testing of Thermal Insulation on Materials for Use at FFTF and SPTF*, LMEC-74-6, LMEC (1974).
- Barrett, W. L., *Design and Construction of the Sodium Natural Convection Experiment*, TRG-Report-2752(R) (Oct 1975).
- Barzelay, M. E., et al., *Thermal Conductance of Contacts in Air Craft Joints*, NACA Tech Note 3167 (Mar 1954).
- Barzelay, M. E., et al., *Effect of Pressure on Thermal Conductance of Contact Joints*, NACA Tech Note 3295 (May 1955).
- Becker, H. B., *A Mathematical Solution for Gas-to-Surface Radiative Exchange Area for a Rectangular Parallelepiped Enclosure Containing a Gray Medium*, J. Heat Transfer 99, 203-207 (1977).
- Bernard, H., *Les Tourbillons Cellulaires Dans une Nappe Liquide*, Rev. Gen. Sci. Pures Appl. 11 (1900).
- Bevans, J. T., and R. V. Dunkle, *Radiant Interchange Within an Enclosure*, J. Heat Transfer 82, 1 (Feb 1960).

- Brannon, R. R., Jr., and R. J. Goldstein, *Emissance of Oxide Layers on a Metal Substrate*, J. Heat Transfer 92, 257 (1970).
- Brubaker, R. C., et al., ANL, personal communication (Feb 1978).
- Brunot, A. W., and F. F. Buckland, *Thermal Contact Resistance of Laminated and Mechanical Joints*, Trans. ASME 71 (1949).
- Buchberg, H., I. Catton, and D. K. Edwards, *Natural Convection in Enclosed Spaces--A Review of Application to Solar Energy Collection*, J. Heat Transfer 98, 182 (May 1976).
- Cassidy, J. F., and M. Herman, *Thermal Contact Resistance Measurements at Ambient Pressures of One Atmosphere to 3×10^{12} mm Hg and Comparison with Theoretical Predictions*, Prog. Astronaut. Aeronaut. 23, 23-24 (1969).
- Castelman, A. W., et al., "On the Behavior of Aerosols under Fast Reactor Accident Conditions," *International Congress on the Diffusion of Products of Fission*, Saclay (1969).
- Catton, I., *Effects of Lateral Boundaries on Natural Convection between Horizontal Surfaces*, Ph.D. thesis, UCLA (1966).
- Catton, I., et al., *Natural Convection Flow in a Finite, Rectangular Slot Arbitrarily Oriented with Respect to the Gravity Factor*, Int. J. Heat Mass Transfer 17 (1974).
- Catton, I., and D. K. Edwards, *Effects of Side Walls on Natural Convection between Horizontal Plates Heated from Below*, J. Heat Transfer, Paper No. 67-HT-4 (Nov 1967).
- Cess, R. D., *The Interaction of Thermal Radiation with Conduction and Convection*, Heat Transfer, Adv. Heat Transfer 1, 1 (1964).
- Chandrasekhar, S., *Hydrodynamic and Hydromagnetic Stability*, Oxford Press (1971).
- Chellew, N. R., et al., "Installation of Prototype Detection Systems on EBR-II," *Sodium Technology Quarterly Report, April-June 1972*, ANL-7957, pp. 33-36 (Nov 1972).
- Chu, T. Y., and R. J. Goldstein, *Turbulent Convection in a Horizontal Layer of Water*, J. Fluid Mech. 60(1), 141-159 (1973).
- Chung, B. T. F., and P. S. Sumitra, *Radiation Shape Factors from Plane Point*, J. Heat Transfer (Aug 1972).
- Clark, J. A., *Cryogenic Heat Transfer*, Adv. Heat Transfer 5, Academic Press (1969).
- Clausing, A. M., *An Experimental and Theoretical Investigation of the Thermal Contact Resistance*, ME Technical Report 242, U. of Illinois (1966).
- Clausing, A. M., *Heat Transfer at the Interface of Dissimilar Metals--The Influence of Thermal Strain*, Int. J. Heat Mass Transfer 9, 791-801 (1966).
- Clausing, A. M., and B. T. Chao, *Thermal Contact Resistance in a Vacuum Environment*, J. Heat Transfer 87, 243 (May 1965).

- Collins, G. D., G. E., personal communication (1978).
- Cooper, L. P., B. C. Cerutti, and D. W. Cissel, *Operating Experience with the EBR-II Rotating-plug Freeze Seals*, ANL-7617 (Dec 1969).
- Cooper, M. G., et al., *Thermal Contact Conductance*, Int. J. Heat Transfer 12, 279-300 (1969).
- Cravalho, E. G., et al., *Effect of Thin Surface Films on the Radiative Properties of Metal Surfaces*, Prog. Astronaut. Aeronaut. 23 (1969).
- Crosbie, A. L., and D. C. Look, *Shorter Communications--A Closed-form Solution to Radiant Interchange between Nonisothermal Plates*, Int. J. Heat Transfer 17, 955-956 (1974).
- DeGraaf, J., and E. Van Der Held, *The Relation between the Heat Transfer and the Convection Phenomena in Enclosed Plane Air Layers*, Appl. Sci. Res. 3 (1952).
- Delisle, J. P., et al., "Operating Experience of Rapsodie and Phenix Relating to Sodium Aerosols and Vapors," IAEA Conf., IWGFR-15, Cadarache, France (Jan 1977).
- De Vahl Davis, G., *Laminar Nature Convection in a Rectangular Cavity*, Int. J. Heat Mass Transfer 11 (1968).
- Drakhlin, E., Zh. Tekh. Fiz. 22 (1952).
- Eckert, E. R. G., and R. M. Drake, Jr., *Heat and Mass Transfer*, McGraw-Hill, NY (1959).
- Eckert, E. R. G., et al., *Total Normal Emissivity Measurements for Porous Materials Used for Mass-transfer Cooling*, ASME Paper No. 57-F-8 (1957).
- Edwards, D. K., et al., *Prediction of Heat Transfer by Natural Convection in Closed Cylinders Heated from Below*, Int. J. Heat Mass Transfer 12, 23 (1969).
- Edwards, D. K., *Suppression of Cellular Convection by Lateral Walls*, J. Heat Transfer, Paper No. 68-WA/HT-6 (Feb 1969).
- Fenech, H., and W. M. Rohsenow, *Prediction of Thermal Conductance of Metallic Surfaces in Contact*, J. Heat Transfer (Feb 1963).
- Files, F. J., et al., *High Capacity--High Efficiency Filters for Sodium Aerosol*, NYO-841-10, Harvard Air Cleaning Laboratory (Aug 1967).
- Fink, J. K., et al., *Interactions of Certain Refractory Materials with Sodium*, ANL-75-74 (May 1976).
- Fitzjarrald, D. E., *An Experimental Study of Turbulent Convection in Air*, J. Fluid Mech. 73(4), 693-719 (1976).
- Fletcher, L. S., and D. A. Gyorog, *Prediction of Thermal Contact Conductance between Similar Metal Surfaces*, Prog. Astronaut. Aeronaut. 24 (1970).
- Funk, C. W., *Sodium Frost Technology*, Hanford Engineering Development Laboratory, HEDL-TC-25, Rev. 1 (July 1974).
- Garson, A. M., and R. J. Goldstein, *Velocity and Heat Transfer Measurements in Thermal Convection*, Phys. Fluids 16(11), 1818-1825 (1973).

- Gebhart, B., *Calculations in Radiant Surroundings of Arbitrary Complexity for Gray, Diffuse Radiation*, Int. J. Heat Mass Transfer 3, 341 (1961).
- Gieseke, J. A., et al., *Characteristics of Agglomerates of Sodium Oxide Particles*, Battelle, BMI-NUREG-1977 (Aug 1977).
- Glaser, P. E., et al., *Thermal Insulation Systems*, ANSA SP-5027 (1967).
- Greenfield, M. A., et al., *Characteristics of Aerosols Produced by Sodium Fires*, AI-AEC-12878 (Oct 31, 1969).
- Grober, H., et al., *Fundamentals of Heat Transfer*, McGraw-Hill, NY (1961).
- Gyorog, D. A., *Investigation of Thermal Insulation Materials for Contacting Surfaces*, Prog. Astronaut. Aeronaut. 24 (1970).
- Hamilton, D. C., and W. R. Morgan, *Radiant Interchange Configuration Factors*, NASA TN2836 (1952).
- Hartnett, J. P., and W. E. Welsh, Trans. ASME 79, 1551 (1957).
- Harpole, G. M., and I. Catton, *Laminar Natural Convection about Downward Facing Heated Blunt Bodies to Liquid Metals*, Paper No. 76-HT-HH, J. Heat Transfer 98, 208 (May 1976).
- Henry, J. J., and H. French, *The Use of Analog Computers for Determining Surface Parameters Required for Prediction of Thermal Contact Conductance*, J. Heat Transfer 86, 543 (Nov 1964).
- Himeno, Y., et al., *Sodium Vapor Deposition in Annular within Enclosed Air Cover Gas Space*, JAPFNR-245 (Nov 1976).
- Himeno, Y., et al., *Sodium Vapor Deposition Studies Related to the Development of Components for LMFBR in Japan*, JAPFNR-243 (Nov 1976).
- Hodkin, E. N., and M. G. Nicholas, Atomic Energy Research Establishment, *The Wetting of Cladding Materials and Other Metals and Alloys by Sodium*, AERE-R-7406, Rev. (1976).
- Hollands, K. G. T., et al., *Free Convective Heat Transfer across Inclined Air Layers*, J. Heat Transfer, Paper No. 75-HT-55 (May 1976).
- Hornberger, D. W., et al., "FFTF Fuel Handling," *Proceedings of the 23rd Conference on Remote Systems Tech*, p. 397 (1975).
- Hottel, H. C., and E. S. Cohen, AIChE J. 4(1), 3 (1958).
- International Working Group on Fast Reactors, *Tenth Annual Meeting Summary Report, Part II*, IWGFR-18, International Atomic Energy Agency (1977).
- International Working Group on Fast Reactors, *Aerosol Formation, Vapour Deposits and Sodium Vapour Trapping*, IWGFR-15, International Atomic Energy Agency (Dec 1976).
- Jakob, M., *Heat Transfer*, Vol. 2, Wiley, NY (1957).
- Jansing, W., et al., *Operating Experience on Sodium Deposits in KNK Reactor and RSB Facility*, IWGFR-15, International Atomic Energy Agency (1977).
- Jurney, E. T., et al., TID-10048, Los Alamos Scientific Laboratory (May 1954).

- Jensen, K. J., ANL, personal communication to K. Shiu, ANL (1978).
- Lapwood, E. R., *Convection of a Fluid in a Porous Medium*, Proc. Cambridge Philos. Soc. 44 (1948).
- Katto, Y., and T. Masuoka, *Criterion for the Onset of Convective Flow in a Fluid in a Porous Medium*, Int. J. Heat Mass Transfer 10 (1967).
- Kreith, F., *Principles of Heat Transfer*, 3rd Edition, Intext Educational Publishers, NY (1976).
- Larkin, B. K., and S. W. Churchill, *Heat Transfer by Radiation through Porous Insulations*, AIChE J. 5(4), 467 (1959).
- Lemercier, G., French Patents 7323338 and 7323339 (1973).
- Lemercier, G., et al., "Development Program and Major Test Results for Thermal Insulation Used in Presence of Sodium Vapors and Aerosols," *Inter. Conf. on Liquid Metal Tech. in Eng. Production*, Champion, PA (1976).
- Lighthill, M. J., J. Mech. Appl. Math 6 (1953).
- MacGregor, R. K., et al., *Radiation Interchange Interior to Multilayer Insulation Blankets*, J. Spacecr. Rockets, 7(2), 221-223 (1970).
- McAdams, W. H., *Heat Transmission*, 3rd Edition, McGraw-Hill, NY (1954).
- McKinzie, D. J., *Experimental Confirmation of Cyclic Thermal Joint Conductance*, Prog. Astronaut. Aeronaut. 24, 289-309 (1970).
- McKinzie, D. J., *Plastic Set of Smooth Large Radii of Curvature Thermal Conductance Specimens at Light Loads*, AIAA J. 11(3) (1973).
- McKisson, R. L., *Sodium Technology*, Atomics International, North American Rockwell (1970).
- Malkus, W. V. R., and G. Veronis, *Finite Amplitude Cellular Convection*, J. Fluid Mech. 4, 225 (1958).
- Martin, B. W., Proc. R. Soc. London A230 (1955).
- Mason, E. A., and L. Monshick, *Transport Properties of Polar-gas Mixtures*, J. Chem. Phys. 36(10) (May 1962).
- Mersman, W. A., *Heat Conduction in an Infinite Composite Solid with an Interface Resistance*, Trans. Am. Math. Soc. 53, 14-24 (1943).
- Mikic, B. B., and W. M. Rohsenow, *Thermal Contact Resistance*, NASA Technical Report No. 4542-41 (1966).
- Norden, P. A., and A. G. Usmanov, *The Conception of Convection in Horizontal Fluid Layers*, Heat Transfer Sov. Res. 4 (1972).
- Oppenheim, A. K., *Radiation Analysis by the Network Method*, Trans. ASME 78, 725 (1956).
- Ostrach, S., and D. Pnuelli, *The Thermal Instability of Completely Confined Fluids inside Some Particular Configurations*, J. Heat Transfer, Paper No. 62-WA-140 (Nov 1963).

Ostonmov, G. A., *Free Convection under Conditions of the Internal Problem*, NACA TM 1407 (1958).

Ozoe, H., et al., *Natural Convection Patterns in Long Inclined Rectangular Box Heated from Below*, Part I, Part II, *Int. J. Heat Mass Transfer* 20, 123 (1977).

Paquin, M. L., "High Temperature Multi-Foil Thermal Insulation," *IEEE Inter Society Conversion Eng. Conf. Rec.*, Boulder, CO, Vol. 1 (1968).

Person, R. A., and H. Leuenberger, *Compilation of Radiation Shape Factors for Cylindrical Assemblies*, Paper No. 56-A-144, *Mech. Eng.* 79, 374 (1957).

Pittman, C. M., and A. J. Bushman, *Configuration Factors for Exchange for Radiant Energy between Axisymmetrical Sections of Cylinders, Cones and Hemispheres and their Basis*, NASA, TN D-944 (1961).

Platten, J. K., and G. Chavpeyer, *Nonlinear Two-dimensional Benard Convection with Soret Effect: Free Boundaries*, *Int. J. Heat Mass Transfer* 20 (1977).

Popov, V. M., *Determining Thermal Resistance of Contact between Finished Wavy Metallic Surfaces*, *J. Eng. Phys.* 32(5), 497-502 (1977).

Powell, R., et al., *Thermocouple Reference Tables Based on the IPTS-68*, U. S. Department of Commerce, NBS Monograph 125 (1968).

Primary System Study Group, ANL, *A Study of Basic Approaches to the Design of the Primary System of Loop-type LMFBRs*, ANL/A78-3 (Mar 1978).

Rabl, Ari, *Radiation Transfer through Specular Passages--A Simple Approximation*, *Int. J. Heat Mass Transfer* 20, 323-330 (1976).

Ralph, J. C., and P. Sugarman, *Free Convection Heat Transfer across Gas-filled Enclosed Spaces between a Hot Liquid Surface and a Cooled Roof*, AERE-R-8244 (Jan 1976).

Randall, K. R., et al., *Natural Convection Characteristics of Flat Plate Collectors*, ERDA, COO/2971-1 (Sept 1977).

Rayleigh, Lord, *On Convective Currents in a Horizontal Layer of Fluid When the Higher Temperature is on the Underside*, *Philos. Mag.* 2 (1926).

Refractory Product Co. publication on the specifications of Ceramic Fibre Insulation.

Rhee, S. J., et al., *Natural Convection Heat Transfer in Beds of Inductively Heated Particles*, *J. Heat Transfer* 100, 78 (Feb 1978).

Richmond, J. C., and J. E. Stewart, *Spectral Emittance of Uncoated and Ceramic-coated Inconel and Type 321 Stainless Steel*, NASA, Memo 4-9-59w (1959).

Roca, R. T., and B. B. Mikic, *Thermal Contact Resistance in a Non-ideal Joint*, NASA, Tech. Report No. DSR71821-77 (1971).

Rohr Industries, Inc., *Handbook of Rohrbond Sandwich Design*, Aerospace and Marine Systems Group (1976).

Roux, J. A., and A. M. Smith, *Combined Conductive and Radiative Heat Transfer in a Scattering Infinite Slab*, *J. Heat Transfer* 100 (1978).

- Schaaf, S. A., *On the Superposition of a Heat Source and Contact Resistance*, Q. Appl. Math. 5(1), 107-111 (1974).
- Schmidt, R. J., and S. W. Milverton, *On the Instability of a Fluid When Heated from Below*, Proc. R. Soc. London, A, 152 (1935).
- Schneider, G. E., et al., *Transient Thermal Response of Two Bodies Communicating through a Small Circular Contact Area*, Int. J. Heat Transfer 20 (1977).
- Schneider, G. E., and M. M. Yovanovich, *Correlation of the Over-all Thermal Resistance of Metallic O-Rings Contacting Two Cylinders*, J. Spacecr. Rockets 11(3) (1974).
- Schneider, P. J. *Conduction Heat Transfer*, Addison-Wesley, Reading, MA (1955).
- Sheriff, N., and N. W. Davies, *Review of Liquid Metal Natural Convection Relevant to Fast Reactor Conditions*, UK-AEA (May 1977).
- Sheth, A., *Parametric Study of Sodium Aerosols in the Cover-gas Space of Sodium-cooled Reactors*, ANL-75-11 (Mar 1975).
- Shimazaki, T. T., *Deposition of Sodium Vapor and Aerosols in Vertical Annuli in the Presence of Thermal Convection*, TI-095-22-133 (Aug 1970).
- Shimazaki, T. T., *Transport of Radioactive Gases through the Sodium Dip Seals in the CRBRP Top Head*, TI-707-230-010 (Mar 1975).
- Shiu, K., and B. G. Jones, *Thermal Performance Prediction of Multi-Plate Insulation for LMFBR Systems*, Trans. Am. Nucl. Soc. 26 (1977).
- Shiu, K., and B. G. Jones, *Thermal Radiation Transfer Through LMFBR Deck Insulation*, Trans. Am. Nucl. Soc. 28 (1978).
- Silveston, P. L., Part I, Forsch. Ingenieurwes. 24 (1958).
- Smith, J., and F. Seely, *Advanced Mechanics of Materials*, Second Edition, Wiley (1952).
- Smuda, P. A., and D. A. Gyorog, *Thermal Isolation with Low-conductance Interstitial Materials under Compressive Loads*, Prog. Astronaut. Aeronaut. 23, 3-22 (1969).
- Sparrow, E. M., *Radiation Heat Transfer between Surfaces*, Adv. Heat Transfer 2, 400 (1965).
- Stakgold, I., *Boundary Value Problems of Mathematical Physics*, MacMillan, NY (1967).
- Strong, H. M., et al., *Flat Panel Vacuum Thermal Insulation*, J. Appl. Phys. 31(1) (1960).
- Stuart, J. T., *On the Non-Linear Mechanics of Hydrodynamic Stability*, J. Fluid Mech. 4 (1958).
- Sun, W. M., and D. K. Edwards, "Natural Convection in Cells with Finite Conducting Side Walls Heated from Below," *Fourth International Heat Transfer Conference Paris-Versailles*, Vol. 4 (1970).
- Threlfall, D. C., *Free Convection in Low-temperature Gaseous Helium*, J. Fluid Mech. 67(1), 17-28 (1975).

- Tien, C. L., and G. R. Cunningham, *A Study of Heat-Transfer Processes in Multilayers Insulations*, Prog. Astronaut. Aeronaut. 23 (1969).
- Tien, C. L., and G. R. Cunningham, *Radiation Heat Transfer in Multilayer Insulation Housing Perforated Shields*, Prog. Astronaut. Aeronaut. 35 (1973).
- Touloukian, D., *Thermal Physical Properties of Matter*, Vol 7, IFI/Plenum (1970).
- Tripp, et al., *Radiation Shape Factors for Plane Surfaces and Spheres, Circles or Cylinders*, Kans. State Univ. Bull. 46(4) (1962).
- Veilleux, E., and M. Mark, *Thermal Resistance of Bolted or Screwed Sheet Metal Joints in a Vacuum*, J. Spacecr. Rockets, Engineering Notes 6(3), 339-342 (1969).
- Verschoor, J. D., and P. Greebler, *Heat Transfer by Gas Conduction and Radiation in Fibrous Insulations*, Trans. ASME (Aug 1962).
- Viskanta, R., *Heat Transfer in Thermal Radiation Absorbing and Scattering Media*, ANL-6170 (May 1960).
- Wade, W., *Measurements of Total Hemispherical Emissivity of Several Stably Oxidized Metals and Some Refractory Oxide Coatings*, NASA, Memo 1-20-59L (1959).
- Wade, W. R., and W. S. Slemper, *Measurements of Total Emittance of Several Refractory Oxides, Cermets and Ceramics for Temperatures from 600°-2,000°F*, NASA, Tech. Note D-998 (1962).
- Westinghouse, *PLBR Pool Report* (1978).
- Wilkes, J. O., *The Finite Difference Computation of Natural Convection in an Enclosed Rectangular Cavity*, Ph.D. thesis, U. of Michigan (1963).
- Willis, N. D., and E. A. Ryder, *Thermal Resistance Measurements of Joints Formed between Stationary Metal Surfaces*, Trans. ASME 71 (1949).
- Yih, Chia-Shun, *Thermal Instability of Viscous Fluids*, J. Fluids XVII(1) (1959).
- Yovanovich, M. M., *Thermal Construction Resistance between Contacting Metallic Paraboloids: Application to Instrument Bearings*, Prog. Astronaut. Aeronaut. 24 (1970).
- Yovanovich, M. M., and M. Tuarze, *Experimental Evidence of Thermal Resistance at Soldered Joints*, J. Spacecr. Rockets Eng. Notes 6(7), 855-857 (1969).
- Yovanovich, M. M., *A Correlation of the Minimum Thermal Resistance at Soldered Joints*, J. Spacecr. Rockets Eng. Notes 7(8), 1013-1014 (1970).