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DEVELOPMENT OF ADVANCED INSULATION FOR APPLIANCES TASK I

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TASK I REPORT

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CHAPTER 1

SUMMARY

1.1 Background

Today, residential and commercial appliances consume a significant amount of energy despite efficiency improvements that have been made over the last 5 years. For some of these appliances, a major portion of the energy goes to offset conduction of heat through insulation. Appliances for which these losses are particularly important are residential and commercial, gas and electric water heaters, refrigerators, freezers, and gas and electric ovens. Currently, the best insulation systems used are polyurethane based foam systems for the lower temperature applications and fiberglass for the higher temperature applications. Typical values of thermal resistivity for these insulations are 7 and 2.7 hr ft^2°F/Btu·in respectively. To reduce the energy consumption of these appliances, ORNL identified the need to develop insulation systems having thermal resistance per inch values in excess of 20 hr ft^2°F/Btu·in (R/in = 20).*

1.2 Objective

The objective of this project is to develop and demonstrate promising insulation systems with thermal resistance/in values greater than 20 hr ft^2°F/Btu·in. In this first task of the project, a preliminary evaluation was made of the techncial feasibility, manufacturing cost, energy savings potential and economic feasibility of a number of candidate systems.

1.3 General Approach

The general approach, summarized in Figure 1.1, involved two basic steps to produce the data which could be used to evaluate the technical and economic feasibility of different insulation systems. First, the energy savings which could be obtained by significantly improving the insulation used in present day appliances were evaluated. This was done by partitioning the energy flow through appliances and evaluating the energy savings which would accrue from arbitrary increases in the thermal resistance of the insulations employed.

Second, a number of different advanced insulation systems were evaluated conceptually to estimate the R/in values that could be achieved. This included both evacuated and nonevacuated systems. Initially, this work was restricted to conceptual designs and theoretical analysis, however, some experimental work was done on air-filled powder systems.

* R/in is the thermal resistivity (hr ft^2°F/Btu·in) of a material and is equal to the inverse of the thermal conductivity.

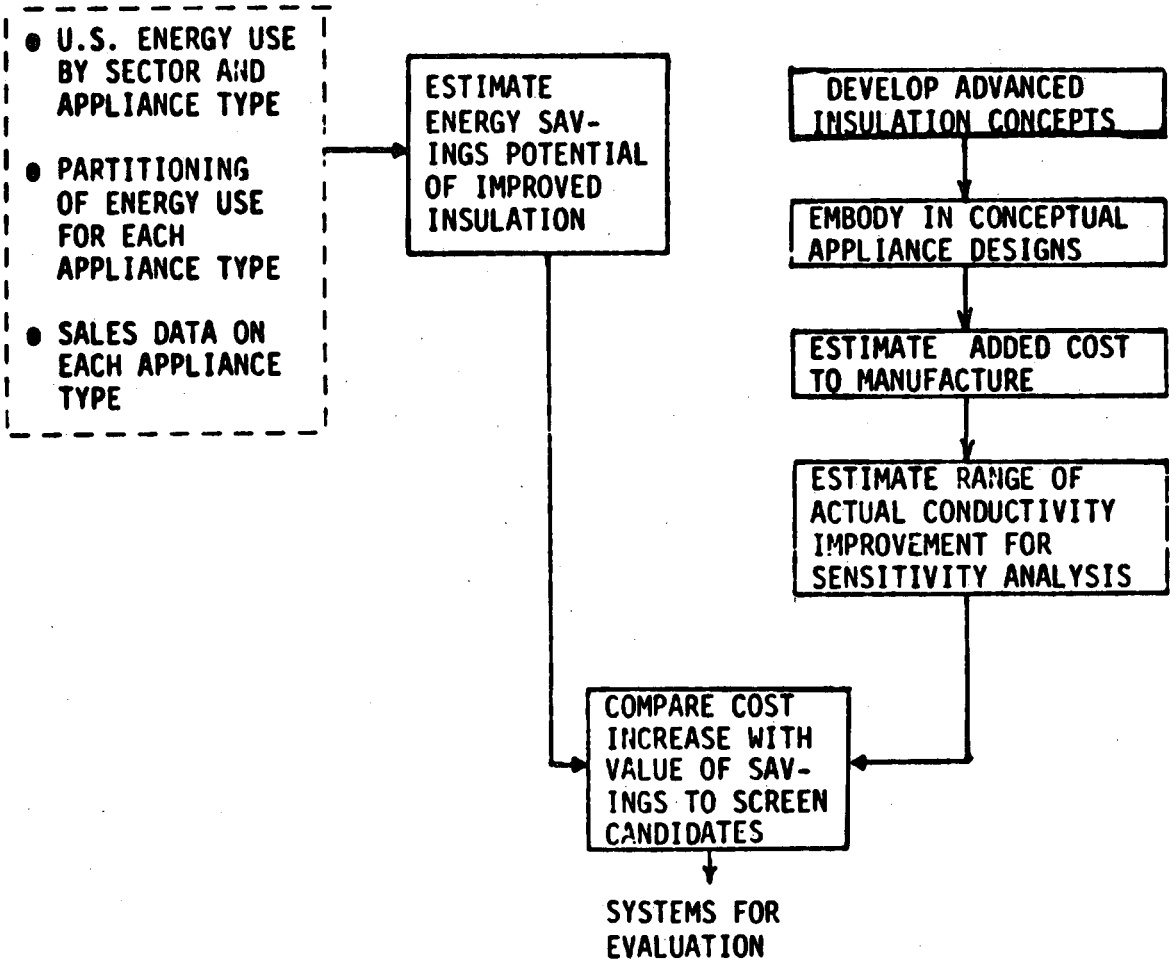

FIGURE 1.1 SUMMARY OF TASK I APPROACH

For the more promising systems, estimates of the cost to incorporate these improved insulations in appliances were made.

Finally results of these two analyses were compared to identify the more promising systems for further development. This was done by calculating simple years-to-payback based on estimated energy savings and the estimated added cost using 1979 fuel prices and costs. Those systems which have a potential years-to-payback of less than 7 years for at least one application, have been selected as candidates for additional development work. The results of this analysis are summarized in Section 1.8.

1.4 General Summary

This analysis identified two types of insulation systems which could meet the threshold value of R/in = 20. These are evacuated systems and gas-filled powders. The evacuated systems fall into two categories:

- those which use an essentially incompressable material to support atmospheric pressure, and
- Those in which the filler material is substantially compressed (thickness reduced 50% or more) when subjected to atmospheric pressure.

Of a number of systems considered, those in the first category using a powder or rigid foam had the best payback periods.

Only one application for a gas-filled powder insulation system was found which met the 7 years-to-payback criteria. This system used air-filled fumed silica. However, thermal properties were not available for a number of lower cost materials which, based on theoretical analysis, would provide acceptable performance.

An important area not covered relates to shipping, maintenance, and customer acceptance questions. Evacuated insulation systems will have a finite life. The systems costed were configured to have a 20 year life span by utilizing a gettering material to remove air which leaks into the evacuated panels during their normal service life. However, some percentage would develop leaks which were beyond the capability of the gettering material. These leaks could deteriorate the performance of the insulation system to the point where replacement is required. Gas-filled powder systems would be much heavier than present systems since densities between 10-20 lb/ft^3 are required to achieve the desired thermal resistivity. This added weight could impact manufacturing, packaging, shipping and installation cost and customer acceptance.

These uncertainties should be addressed in subsequent phases of the project. Specific recommendations are given in Section 1.7.

1.5 Specific Findings

The following major findings were made during this project:

1. Availability of Candidate Insulations:

Two basic types of insulation systems are available which could have R/in values greater than 20 hr ft^2°F/Btu·in. (See Section 2.6) These are:

- Air-filled powders using very small particles (~.005 microns) to reduce gas conduction and convection heat transfer and

- Evacuated panel systems using a variety of load bearing materials contained in an air tight impermeable encapsulation material.

2. Cost Effectiveness of Candidate Systems:

Based on preliminary estimates of manufactured cost and potential energy savings, both of these types may be cost effective in some appliance applications. (See Section 1.8). Air-filled powders using fumed silica are only marginally acceptable, and lower cost materials must be found if they are to have general use.

3. Thermal Property Data

Many of the thermal properties needed to accurately predict the performance of these insulation systems are not available. (See Section 2.2.3) Properties of various powders which may be suitable for air filled systems and properties of evacuated materials under compression at temperatures appropriate for appliances are needed.

4. Status of Commercialization (See Section 2.2.2)

a) Evacuated Systems:

- At least one major appliance manufacturer is currently conducting laboratory tests on evacuated panels.

- Linde has tested 4 systems which they believe can be used in appliances.

b) Air-filled Powder Systems:

- A number of manufacturers are conducting laboratory tests, producing pilot lot quantities, or selling air-filled powder systems.

- These systems are based on fumed silica with a base price of approximately $2.00/lb.

- Reported R/in values range from 7 to 36 $\frac{hr\ ft^2 °F}{Btu \cdot in}$ at 200°F and -120°F respectively.

- Density is typically 15 lb/ft^3.

5. Edge Effects

Heat flow through the edges of evacuated panels can reduce or eliminate potential savings depending on the configuration used. (See Section 2.5.4)

6. Impact of Other Cost Elements:

Additional costs elements (not included in the preliminary analysis) such as shipping containers, warranty service, maintenance and appliance life may have a major impact. (See Section 2.1.3).

1.6 Conclusions

1. Evacuated and air-filled insulations show sufficient promise so that further research and development work is justified to resolve performance, cost, life, and weight penalty uncertainties.

2. For both types of systems, the state-of-the-art is such that additional development work is required before a demonstration program with manufacturers participation can be initiated.

1.7 Recommendations

We recommend that further work be undertaken to obtain the additional information necessary to complete the concept evaluation prior to the initiation of a demonstration program. The general objective of this phase of the project would be to quantify the impacts on costs and customer utility of the factors identified in Section 1.5 for both air-filled powder and evacuated systems. Specific tasks are listed below.

1.7.1 Evacuated Panel

Preliminary feasibility analyses have shown that several evacuated panel configurations, particularly those employing perlite (Concepts VI and X),* may offer a potential for significant energy conservation with acceptable consumer costs. In order to establish the commercial practicality of these concepts and obtain the participation of a manufacturing partner we recommend that the following tasks be undertaken:

1. Make a preliminary evaluation of practicality by:

   - Developing several possible configuration concepts which edge losses.

   - Performing failure analysis of these concepts to predict failure rates and types.

*See Section 1.8

- Quantifying additional costs (manufacturing, shipping, warranty, life, etc.)
- Re-evaluating feasibility based on energy savings potential and total added cost.

2. Measure properties of candidate materials under appropriate temperatures and pressures.

3. Fabricate and test evacuated panels in the laboratory.

4. Refine data from Task 1 based on laboratory tests.

5. Locate manufacturing partner for demonstration project.

6. Prepare demonstration plan.

1.7.2 Air-Filled Powder

The preliminary feasibility analysis indicated that air-filled powder insulations using fumed silica as the base material may be cost effective in a few applications. However, for many applications, the added weight due to their high density (e.g., 15 lb/ft^3) may impact the utility of the appliance for the user. The weight will also add costs for shipping and possibly for installation. In addition, although the use of fumed silica at two dollar per pound was found to be marginally acceptable, considerably less expensive materials are required if air-filled powders are to have wide spread use.

To address these considerations we recommend that the following tasks be undertaken to further evaluate the feasibility of air-filled powders:

1. Evaluate impact of added weight from viewpoint of:

   - Customer utility
   - Cost increase due to packaging, shipping and installation difficulties.

2. Estimate cost-per-pound that candidate materials should have to be viable.

3. Identify materials with acceptable particle size and cost.

4. Select most promising materials and perform laboratory tests to determine R/in value at different densities.

5. Locate manufacturing partner for demonstration project.

6. Prepare demonstration plan.

1.7.3 Discussion:

This recommended work should clearly indicate whether any of the candidate systems are economically and technically feasible. In both cases, each of the first four tasks represent logical milestones at which a decision about proceding can be made. If the steps are conducted in parallel, the data developed for one type will in many cases be useful in the evaluation of the other.

We believe that the results to date indicate that one or both of these types of systems will prove economically and technically feasible and that significant energy conservation would result from their use.

1.8 Summary of Payback Analysis Results

A number of potential systems for achieving high performance insulations were identified. Table 1.1 describes these systems briefly. All but one of these (Concept V) are evacuated. Concept V is a gas-filled powder. The last four concepts are based on evacuated systems proposed by Linde.* In addition to these systems which were found to be cost effective in some applications, other systems which were not found to be cost effective are described in Chapter 2.

For each of these systems, estimates were made of the manufactured cost including materials of using them in the appliances considered in this project. In addition, estimates of the energy savings and the value of this energy savings on an annual basis were made for each of the appliances. These energy savings estimates are summarized in Tables 1.2 and 1.3 for different values of insulation resistivity. Tables 1.4 and 1.5 summarize the results of payback calculations made utilizing these costs and energy savings estimates. For residential appliances, evacuated insulation systems appear to offer promise for refrigerators, freezers and both gas and electric water heaters. In some cases, quite reasonable payback periods are projected. For commercial appliances the most promising application appears to be ovens. Commercial freezers and commercial water heaters also have good payback periods for a few of the evacuated systems. The air filled system, using relatively expensive Cab-O-Sil is marginally acceptable only for commercial electric ovens.

*Letter to ADL dated October 18, 1977

SPECIFICATION NUMBER	CONCEPT*	THERMAL RESISTIVITY $\left(\frac{hr\ ft^2 °F}{Btu \cdot in}\right)$
IV	Evacuated Open Cell Foam (Phenolic with Steel Encapsulation)	15
V-2	Air Filled Fumed Silica (Cab-O-Sil)	20
VI	Evacuated Powder (Perlite with Steel Encapsulation)	20
IX	Hollow Glass Spheres in Vacuum (Glass Spheres with Steel Encapsulation)	15
X	Evacuated Powder with Rigid Foam Supports (Perlite 67% and Urethane with Steel Encapsulation)	20**
XI	Evacuated Multilayer (5%) and Urethane (Aluminized Mylar and Silk with Steel Encapsulation)	20**
XII	Evacuated Powder with Honeycomb Support (Perlite with Fiberglass Epoxy Honeycomb and Steel Encapsulation	20**
A***	Linde A (49%) and Urethane	20**
B	Linde B (32%) and Urethane	20**
C	Linde C (19%) Urethane	20**
D	Linde D (8%) and Urethane	20**

* Other concepts considered had pay backs in excess of 7 years for all applications evaluated.

** To minimize cost and achieve R/in = 20, these systems were based on sandwiching the evacuated panel between two layers of polyurethane (See Concept X Chapter 2 for detailed description.) Percentage indicates the percent by volume of the evacuated insulation component.

*** All Linde Concepts are evacuated.

TABLE 1.1 SUMMARY OF INSULATION CONCEPTS INCLUDED IN TABLES 1.2 and 1.3

Appliance	Baseline Energy Consumption (10^6 Btu/year)	SAVINGS USING INSULATION WITH					
		R/in = 10		R/in = 15		R/in = 20	
		(10^6 Btu/yr)	($/yr)	(10^6 Btu/yr)	($/yr)	(10^6 Btu/yr)	($/yr)
Refrigerator	3.7	0.77	11.28	1.53	22.41	1.88	27.54
Freezer (Chest)	3.9	0.78	11.43	1.54	22.56	1.86	27.25
Water Heater							
Gas-	30.4	2.43	9.23	2.86	10.87	3.07	11.67
Electric-	20.5	1.52	22.27	1.66	24.32	1.75	25.64
Oven							
Gas-	2.82	0.17	0.64	0.20	0.77	0.22	0.84
Electric-	1.23	0.17	2.46	0.20	2.97	0.22	3.24

NOTE:
- All savings at point of use
- Electricity @ $0.05/Kwh
- Gas @ $3.80/$10^6$ Btu

TABLE 1.2: SAVINGS FROM USE OF IMPROVED INSULATION IN RESIDENTIAL APPLIANCES

Appliance	Baseline Losses through Insulation (10^6 Btu/year)	SAVINGS USING INSULATION WITH					
		R/in = 10		R/in = 15		R/in =20	
		(10^6 Btu/yr)	($/yr)	(10^6 Btu/yr)	($/yr)	(10^6 Btu/yr)	($/yr)
Freezer*	2.30	0.73	10.69	1.59	23.29	1.96	28.71
Water Heater - Gas	3.61	2.35	8.93	2.76	10.49	2.97	11.29
Electric	3.89	2.54	37.21	2.98	43.66	3.21	47.03
Oven** - Gas	4.2	2.60	9.87	3.04	11.56	3.30	12.54
Electric	4.2	2.60	38.07	3.04	44.56	3.30	48.35

* Per 3 foot section of display freezer

** Single free standing oven

NOTE:
- All savings at point of use
- Electricity @ $0.05/Kwh
- Gas @ $3.80/$10^6$ Btu

TABLE 1.3: SAVINGS FROM USE OF IMPROVED INSULATION IN COMMERCIAL APPLIANCES

	YEARS-TO-PAYBACK FOR CONCEPTS EVALUATED									
R/in	15	20	15	20	20	20	20	20	20	20
Concept / Appliance	IV	VI	IX	X	XI	XII	A	B	C	D
Refrigerator	6.4	2.4	--	2.8	--	--	--	--	--	--
Freezers	5.9	2.6	--	2.9	--	6.1	5.7	--	--	--
Water Heaters										
Gas	7.0	4.9	--	5.1	--	--	4.7	5.0	4.8	4.7
Electric	4.2	2.4	5.4	2.6	4.8	4.4	4.2	--	5.2	3.0
Ovens										
Gas	--	--	--	N.A.	N.A.	N.A.	N.A.	N.A.	--	--
Electric	--	--	--	N.A.	N.A.	N.A.	N.A.	N.A.	--	--

NOTE:
- Blanks signify that years-to-payback were greater than 7 for the application and concept.
- N.A. signifies temperature range of insulation not acceptable.
- All concepts are evacuated.

TABLE 1.4: PAYBACK PERIODS FOR INSULATION SYSTEMS IN RESIDENTIAL APPLIANCES

YEARS-TO-PAYBACK FOR CONCEPTS EVALUATED											
R/in	15	20	20	15	20	20	20	20	20	20	20
Concept / Appliance	IV	V-2	VI	IX	X	XI	XII	A	B	C	D
Display Freezer	6.2	—	1.6	--	2.2	--	--	--	--	--	--
Water Heater -											
Gas	--	—	5.9	--	6.4	--	--	--	--	--	--
Electric	3.4	—	1.8	4.7	2.0	4.2	3.8	3.6	6.5	4.5	2.5
Oven -											
Gas	5.8	—	4.3	6.9	N.A.	N.A.	N.A.	N.A.	N.A.	6.9	4.6
Electric	1.5	4.3	1.1	1.8	N.A.	N.A.	N.A.	N.A.	N.A.	1.8	1.2

NOTE:
- Blanks signify that years-to-payback were greater than 7 for the application and concept.
- N.A. signifies temperature range of insulation not acceptable.
- All concepts are evacuated except Concept V-2.

TABLE 1.5: PAYBACK PERIODS FOR INSULATION SYSTEMS IN COMMERCIAL APPLIANCES

CHAPTER 2

INSULATION CONCEPTS

2.1 General Approach

2.1.1 Concept Identification

In this analysis, we combined mathematical analysis, insulation performance data from the literature, contacts with manufacturers and a broad practical experience with high performance insulation materials and systems to develop and evaluate various insulation concepts for application to the energy consuming appliances under consideration. The mathematical analysis served two purposes. First, it provided a basis for understanding the heat transfer process occuring within a thermal insulation. Second, it was used to predict the thermal performance of various concepts. These concepts came from:

- "Blue Sky" meetings in which ADL specialists in materials, manufacturing techniques, insulations and appliances worked together to explore new concepts.
- Data from manufacturers
- Modifications or combinations of insulation systems based on performance data collected from various sources including:
    - manufacturers
    - other researchers
    - patents
    - literature
    - prior ADL work

Section 2.2 describes the various sources of concepts identified in this project. The remainder of the chapter describes the analysis used to develop and evaluate the specific concepts considered and presents detailed descriptions of the concepts.

2.1.2 Concept Evaluation Criteria

The criteria used in the preliminary evaluation and selection of materials and concepts were:

- Thermal performance, i.e., potential steady state R-value
- Temperature limitations
- Relative cost

- Evacuated vs. nonevacuated systems
- Rigid vs. compressible insulation materials
- Leak detection requirements
- Encapsulation requirements
- Technical feasibility
- Cost vs. potential energy savings.

2.1.3 Steering Committee Review

In addition to applying these criteria, preliminary concept descriptions were submitted to a steering committee for review. Companies represented were:

- Carborundum
- Owens Corning
- DuPont

This meeting produced the following considerations:

- Lifetime should be at least 20 years
- Powders tend to settle over time and this would reduce the insulation effectiveness
- Blowing agents in composite insulations (such as Concept X) will defuse and degrade the performance of the evacuated component
- Costs associated with the use of evacuated insulations plus the difficulty in obtaining long lives may make them unacceptable for use in appliances.

In addition to this formal meeting, discussions have been held with Cabot, Calgon, and A. O. Smith concerning the various concepts.

2.2 Development of Preliminary Concepts

2.2.1 Blue Sky Meetings

Two meetings were held with ADL specialists to discuss advanced insulation concepts. Primary attention focussed on reducing the gas conduction component. Ideas included:

- Make the insulation with a high proportion of colloidal particles or small fibers with dimensions equal to that of the mean free path of air. (Concept V, Section 2.6).

- Fill the small particle insulation with a large molecule gas such as krypton, argon, xenon, flourocarbon or sulfur hexafloride (Concept V, Section 2.6).

- Fill the insulation with a gas which would be condensed at the normal operating temperature. (This concept was evaluated but vapor pressures, even in the condensed state, were found to be too large to reduce gas effects).

- Use the thermal cycling of the appliance to pump gas out of the insulation using a gettering material which could be regenerated or valves which opened when the appliance was at it's highest temperature (No practical embodiment was identified).

- Solid conduction could be reduced by orientation of foam cells or fibers, layering of materials or use of hollow (evacuated) spheres or fibers. (Since gas conduction is the most important loss mechanism, these concepts were not evaluated).

- Evacuated panels could be intermittently pumped by employing the motor of the compressor system in a refrigeration system to drive a vacuum pump when the compressor is running.

- Inner and outer appliance walls could be used for encapsulation. These flat surfaces could be supported by Nomex or other load bearing materials (Concept X and XII, Section 2.6).

- Mylar-Aluminum encapsulated evacuated panels could be foamed in place to protect against leaks and punctures. (Concept XI, Section 2.6)

2.2.2 Other Developments

EVACUATED SYSTEMS

Two manufacturers were found to be involved in the development of evacuated panels which might be used in appliances. A major manufacturer* has indicated that they have been evaluating various systems in the laboratory, but have not developed a system suitable for field testing or demonstration.

Linde during the course of other work has developed 4 insulations which might be suitable for appliances. They submitted thermal properties (see Section 2.6, Concepts A,B,C, and D) for the basic insulation; however, they have not attempted to integrate the systems with actual appliances.

AIR-FILLED POWDERS

A number of manufacturers and researchers are selling or developing powder based insulations. Those identified during this project are described below:

Micropore Insulation Ltd., Upton, England:

The firm manufactures an insulation called "Microtherm." It is basically a compacted fumed silica with fiberglass and binders for strength giving a density of about 15 lb/ft^3. It has an R/in value of 7.1 hr ft^2°F/Btu·in at a mean temperature of 120°F, and can be used to hot face temperatures in excess of 1500°F. Primary applications include wood burning stoves and electric heat storage units where high temperatures are encountered.

Grunzweig and Hartmann (G&H), Frankfurt, Germany:

This firm manufactures an insulation called "Minileit" which has properties and applications similar to those of "Microtherm."

L'Air Liquide, Georges Claude, France:

A patent** on compacted powder insulation made from fumed silicas has been issued to L'Air Liquide. The patent shows R/in values of 24 hr ft^2°F/Btu·in at 32°F and 300 mm Hg and 45 hr ft^2°F/Btu·in at -120°F and atmospheric pressure. No properties are given at atmospheric pressure at the temperature of interest in this project. L'Air Liquide has indicated that work on commercializing this product is progressing and more data and samples should be available in late 1980.

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* Confidential disclosure to Arthur D. Little, Inc.

** U.S. Patent # 4,159,359

Vanderbilt University, Nashville, Tennessee:

Results using powders comprised of compacted fumed silica (50%) and carbon black (50%) have been reported* showing R/in values of 13.4 hr ft^2 ^oF/Btu·in with 10 lb/ft^3 density. Further work is continuing.

Gullfiber International A.B., Goteborg, Sweden

This firm is working on the development of an insulation which we believe is similar to that of Micropore, G&H, and L'Air Liquide. No data are currently available.

Materials Technology, Atlanta, Georgia

This firm is working on the development of a proprietory insulation which they expect to have R/in values greater than 20 hr ft^2°F/Btu·in. Details should be available in 6-12 months from the date of this report.

* Rogers, D.B., and Williamson, John. W., "Thermal Characteristics of Gas-Filled Fiber Powder Insulation Systems." 15th International Thermal Conductivity Conference Proceedings, August 24-26, 1977.

2.2.3 Data on Thermal Insulating Materials

As mentioned above, one starting point for this analysis was existing performance data on low conductivity insulations. The applicable data, collected from various sources, are summarized in Table 2.1 For this table the following notes apply:

- Density:

  A range of densities is indicated for several materials. This indicates that the material was listed over a range of densities and the thermal resistivity presented may be an average value, a typical value, or the most common value. Also these data may have been reported by one or many authors. The R/in values associated with the single density values usually represent an average R/in value obtained from many tests with the same or different materials by the same or different authors.

- Particle Size:

  The particle size of the tested materials is an important parameter in evacuated insulation. However, this information was available for only 20 percent of the materials tabulated. In many cases our sources were secondary which may account for the incompleteness of the information.

- Mean Temperature and Temperature Range:

  The mean temperatures, T_m, and cold and hot plate temperatures, T_c and T_h, are given in columns 5 and 6 respectively for the sample test conditions. It is important also to note that in all cases except one, 87°F was the highest test sample temperature. Generally these data were taken for cryogenic applications, i.e., cold plate temperatures at -320°F (the boiling point for nitrogen) and -423°F (the boiling point for hydrogen) and are, therefore, outside the range of interest in the present study.

- Thermal Resistivity (R/in):

  The data in the literature were generally presented as the apparent thermal conductivity versus gas pressure in a variety of units. These were converted to thermal resistivity (hr ft^2°F/Btu-in) and pressure in mm Hg.

No.	Material Description	Density (lb/ft^3)	Particle Size (Micons)	Tm (°F)	T_c, T_h (°F)	R/in Values at Indicated Cell or Interestitial Gas Pressure (Atmospheres) 10^{-8}	10^{-7}	10^{-6}	10^{-5}	10^{-4}	10^{-3}	10^{-2}	10^{-1}	Atmos	Gas	Reference #	Page
1.	Aluminum Oxide Granuals	125	-	-120	-320,80	83		-	-	-	-	-	-	-	air*	1	28
2.	Calcium Silicate Powder	7-22	.02-.07	-120	-320,80	278	88-278	-	-	-	-	-	-	-	air*	1	64
3.	Glass fiber, Type AA	15	-	-115	-315,84	-	-	-	417	166	42	12	8	8	air*	1	100
4.	Min-K 504	16	-	-110	-320,100	-	-	-	-	-	-	-	-	6	air*	1	103
5.	Magnesium Silicate (Talc Powder)	75	-	-120	-320,80	83	-	-	-	-	-	-	-	-	air*	1	139
6.	Multi-layers, Linde S1-12	2.5-4.7	-	-375	-420,-330	-	2777	1851	555	83	23	14	11	11	air*	1	148
7.	Perlite Granular	8.7	-	-130	-322,80	-	138	138	138	69	13	8	5	4	air*	1	166
8.	Phenolic spheres	12	25-100	-120	-320,80	119	119	111	88	42	10	6	-	-	air*	1	171
9.	Polystyrene Fiber	2.8	-	-265	-320,-203	-	-	119	46	24	15	11	11	9	air*	1	177
10.	Polystyrene granuals (ground styrofoam)	2-3.7	-	-379	-424,-335	833	520	128	20	8	4	4	4	1	air*	1 1	191 191
11.	Silica Aerogel Powder	6	.025	-373	-424,-323	833	694	520	278	151	83	-	-	-	He	1	193
12.	Silica Aerogel Powder	6	-	-117	-322,87	83	83	69	49	33	23	16	12	8	N_2	1	208
13.	Santocel&Copper Flakes(LindeCS-5)	11	-	-375	-420,-330	-	1666	1515	1389	185	64	-	-	-	air*	1	2
14.	Silicon Dioxide & Carbon	3.5-5	-	-130	-320,61	99-277	-	-	-	-	-	-	-	-	air*	1	246
15.	Silicon Dioxide Powder	3.3	-	-263	-319,-207	-	-	-	69	43	28	17	10	7	air*	1	91
16.	Vermiculite	9.4	-	-117	-322,87	-	83	-	-	-	-	-	-	3	N_2	1	70
17.	Diatomaceous Earth Powder	15	1-100	-120	-322,80	-	119	119	111	83	33	-	-	-	N_2	2	24
18.	Carbon Lampblack Powder	12		-120	-322,80	111	111	111	98	56	28	13	8	-	N_2	3	7
19.	ADL 3042 (Al/Mylar & Silk net)	15.8	-	-125	-320,69	556	-	-	-	-	-	-	-	-	-		
20.	ADL 17			365	80,650	-	36	26	13	6	4				air		

NOTES:
- Data taken at $< 5 \times 10^{-5}$ mm Hg
- Tm = Mean Temperature
- T_c T_h are the Cold and Hot plate temperatures during measurement of the apparent thermal conductivity.
- Gas = Refers to interstitial gas: (*) indicates the assumed gas type when not identified in the data.
- Sample 19 is compressed with a mechanical load of 15.6 psi during measurement. All other samples are unloaded.

1. Loser, J.B., C.E. Moeller and M.B. Thompson "Thermo Physical Properties of Insulating Materials", ML-TOR-64-5, April, 1964.
2. Arthur D. Little, Inc., "Advanced Studies on Multi-Layer Systems for NASA", CR 54924, 1966.
3. Glaser, P.E., "Advances in Thermal Insulating Materials for Thermal Protection Systems", AMER Rocket Society, Paper 2153-61

TABLE 2.1: SUMMARY OF INSULATION MATERIALS DATA

- Interstitial Gas:

  In many of the reported data no reference was made to the gas within the insulations. Except where noted, we have assumed air for the interstitial gas. This is reasonable as most samples are made up in an air environment and have absorbed atmospheric gases on their surfaces.

- Compression: (Not indicated on Table 2.1)

  Compression refers to the mechanical load imposed on the system when the interior is evacuated. Because these data are for insulations applied to cryogenic systems where the boundaries consist of rigid structures that are made to resist atmospheric loads without imposing them on the insulation, the considerations of compression as an active parameter is rarely done. Thus, there are few data which measure the characteristic that is important to insulations encapsulated with flexible or non rigid boundaries.

Conclusions

- Evacuated powders, fibers and multi-layer insulation dominate the literature for resistivities greater than 20. Generally, these data are for evacuated insulations which are contained between structurally rigid boundaries so that the insulations do not need to support the compressive load of the atmosphere. This compressive load may be present in the application of these insulations to the appliances under consideration. Additional data are needed, therefore, to show the effect of compressive load as well as vacuum pressure on resistivities.

- Material No. 19 is a multi-layer insulation that was measured while evacuated and under a compressive load of one atmosphere. This is representative of others measured under similar conditions. However, the vacuum conditions of the sample were never increased above 10^{-4} mm Hg and the expected loss of performance associated with increased gas pressure was not determined.

- New experimental information is required particularly for a variety of powders and fibers (also combinations of these two) in the temperature range of -20 to +600 °F and for a range of compressive loads from 1 to 2 atmospheres.

- Higher R/in values may be achieved with air filled powders but experimental verification at temperatures of interest for appliances is required since the conductivity may increase significantly with temperature.

2.3 Theoretical Analysis

2.3.1 Introduction

Thermal insulation materials are designed to resist the transfer of heat between two boundaries at different temperatures. Whether natural or man made, they are characteristically porous, low in weight and have little or no metallic components. Their high thermal resistance is sometimes accomplished with a low weight material prepared from loosely bonded fibers and powders filled with air. Sometimes, this is accomplished with low weight materials in the form of foams (both closed or open cells) filled with air or other gases that have special thermal properties.

In any insulation there are three basic modes of heat transfer: 1. conduction; 2. convection; and 3. radiation. Conduction occurs in both the solid material and in the gas filling the spaces. Convection takes place in the gas as a result of the unbalanced bouyant forces induced by differences in temperature among the various regions of the insulating material. Radiation takes two forms: 1. direct transmission between the hot and cold boundaries through the interviewing mediums, both gas and solid; 2. absorption within the intervening mediums, mainly the solid material and its reemission. These heat transfer modes are interactive, but are often treated as independent phenomena for simplicity.

Good thermal insulation materials usually are made up with a small amount of solid material that has a high thermal resistance and is distributed in a form which creates a large number of small voids. Solids have thermal resistivities lower than those gases by factors of 1 to 4 orders of magnitude. For this reason, they are used sparingly. While gases have a high resistivity, they are poor thermal insulators when used alone because they are easily convected. Further, thermal radiation is an important heat transfer mechanism when a gas is present alone between two boundaries at different temperatures. Thus, a small amount of solid material in the form of fibers or powders impedes gas convection and interrupts the thermal radiation. The resistance through the solid is made large by using only materials with a large thermal resistivity and by reducing all direct paths through the solid material. Solids with a high thermal resistivity, such as foamed polyurethane entrap gases in small closed cells, thus, preventing convection and intercepting thermal radiation.

All conventional insulations have a resistivity of about 8 hr ft^2°F/Btu-in. or less. Some of these materials can be made to yield higher thermal resistivities by modifying them or their conditions of use or by combining these with other insulating media. In each modification an attempt is made to reduce or eliminate one or more of the dominant heat transfer modes.

The following is a brief discussion of the three modes and how each can be increased or decreased by changes to the basic insulation material.

2.3.2 Gas Conduction

The thermal conductivity of a gas depends in part on the distance that a gas molecule travels before it strikes another molecule (mean free path) and the average distance between the boundaries of any solid material present in the gas (pore size). When the mean free path of the gas is small in comparison to the pore size, its thermal conductivity varies only slightly with pressure. However, when the mean free path of the gas is greater than or approximately equal to the pore size, its thermal conductivity decreases in proportion to the ratio of the pore size to the mean free path. In vacuum type insulations, the mean free path of the gas is increased by reducing its pressure.

Many investigators have studied the effect of pressure and pore size on gas thermal conductivity. Although different derivations have been employed, all the theoretical treatments generally reduce to the following form:*

$$k_g = \frac{k_{go}}{1 + C_1} \qquad (2.1)$$

$$\text{where: } C_1 = \left(\frac{P_o}{P}\right) \left(\frac{L_o}{d}\right) \left(\frac{T}{T_o}\right)^n \left(\frac{2-\alpha}{\alpha}\right) \qquad (2.2)$$

k_{go} = Thermal conductivity of the gas at standard conditions

k_g = Thermal conductivity of the gas for a given pore size and gas pressure

P_o = Standard Atmospheric pressure

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* Arthur D. Little, Inc., "Advanced Studies on Multi-layer Insulation Systems", NASA Report CR-54929, 1966, p 11-114.

L_o = Mean-free path of gas molecules at P_o and T_o

T_o = Gas temperature at standard conditions

P = Pressure of gas within the insulation; same units as P_o

T = Mean temperature of the insulation; same units as T_o

d = Average pore size; same units as L_o

α = Accommodation coefficient (dimensionless)

n = A constant often taken as unity.

As a simplification, α is assumed to have a value of 1.0. For refrigerators and freezers (T/T_o) has a value of 1.0 for practical purposes. L_o is calculated from the equation.*

$$L_o = 15.5 \times 10^4 \frac{\nu}{P_o} \sqrt{\frac{T_o}{M}} \quad \text{(microns)} \qquad (2.3)$$

where: ν = the viscosity of the gas in poise

M = Molecular weight.
P_o is in mm Hg and T_o in ^oR

Using these equations, the pore size and gas pressure required to reduce the conductivity of air to about one-tenth of its value at standard conditions (an approximation) were calculated (see Figure 2.1).

The experimentally determined values of thermal conductivity of evacuated insulations usually exhibit a characteristic "S" shaped relationship with gas pressure. As the pressure of the interstitial gas is reduced, its mean free path is lengthened. Because the mean pore size of a particular insulation is fixed, the gas component of the insulation's thermal conductivity is directly related to gas pressure. From Equation 2.1 and 2.3 the value of the ratio of K_g/K_{go} for air has been computed over a 5 decade range of pressures for an insulation with a mean pore size of 25 microns. The values obtained are tabulated below and are also shown in Figure 2.1. These reveal the typical "S" curve exhibited by the experimental data for evacuated insulation systems.

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* Scott, R.B., "Cryogenic Engineering," D. Van Nostrand Co., Inc., New York, March 1959, p 146.

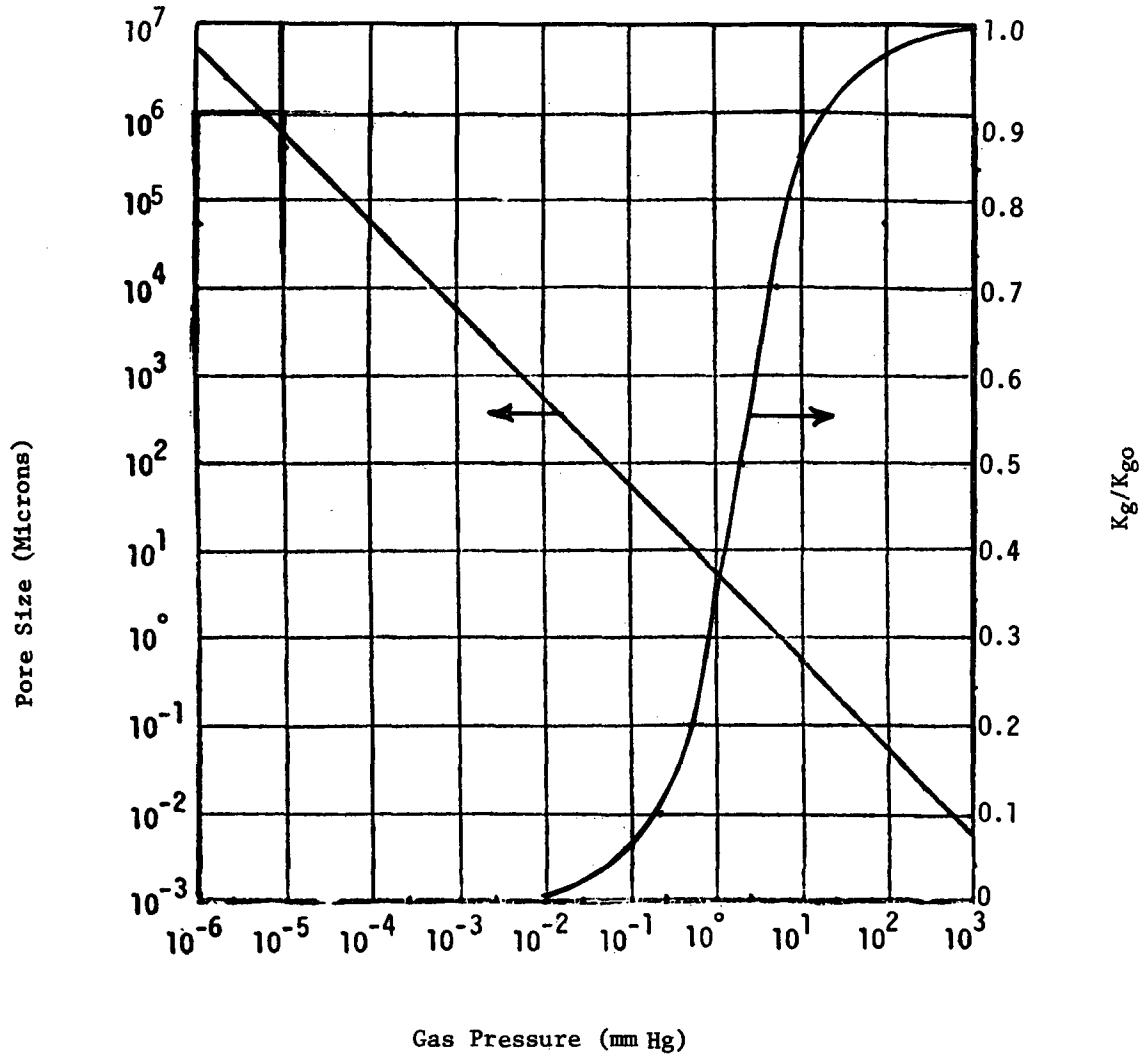

FIGURE 2.1 RELATIONSHIP BETWEEN GAS PRESSURE AND PORE SIZE NECESSARY TO REDUCE THE THERMAL CONDUCTIVITY OF AIR TO ONE TENTH ITS VALUE AT STANDARD CONDITIONS AND THE RATIO OF K_g/K_{go} FOR AIR AS A FUNCTION OF GAS PRESSURE AT A MEAN PORE SIZE OF 25 MICRONS

Air Pressure p(mm Hg)	$\frac{1}{(1+C_1)} = \frac{K_g}{K_{go}}$
2.0×10^2	0.99
2.0×10	0.91
2.0×10^0	0.50
2.0×10^{-1}	0.09
2.0×10^{-2}	0.01
2.0×10^{-3}	0.001

The computed ratios are an approximation of the decrease in k_g for several reasons. Pore size in an insulation is not a fixed quantity, but one varying with the local particle size, shape, orientations (fibers only) and insulation bulk density. Further, the free paths of the gas molecules have a wide statistical distribution around the mean value of the free path. Another effect on the ratio is the accomodation coefficients which were assumed to have a value of 1.0 for the atmospheric gases, when in fact, the actual values, which cannot be accurately evaluated, are always less than 1.0. Finally, the ratio is modified by the interaction of the gas heat transfer mode with the solid and radiation heat transfer modes.

As is evident from Equations 2.1 and 2.2, for any gas pressure as pore size is reduced the thermal conductivity of the gas is reduced. Thus, even at atmospheric pressure the conductivity of air can be reduced by an order of magnitude in a material with pore size of 10^{-2} microns. (This compares with the wave length of the visible spectrum of light which is 0.4 to 0.7 microns). Pore sizes of 10^{-2} microns can be produced only with compacted particles whose principal dimensions are of the order of 10^{-2} microns. Chrysolite, the serpentine form of asbestos, and very fine carbon black powders have dimensions in this range. Type AAA and AAAA Fiberglass fine bulk fibers have diameters of 0.63 and 0.35 microns respectively and are about an order of magnitude larger than that required to reduce the thermal conductivity of air at atmospheric pressure. To be effective, the fibers and/or powders would have to be compacted so that the mean dimension of spaces between them was comparable to their physical dimensions. This would would yield an insulation with a high bulk density (greater than 10 lb/ft^3). Further, the decrease in gas conductivity will most likely be offset by solid

conduction heat transfer that is caused by the large amount of compacted solid present.

The reduction in the thermal conductivity of the gas from reducing the gas pressure can be accomplished only if the insulation is placed inside of an enclosure. (Closed cell foams are an exception to this and will be discussed later.) The boundaries of the enclosure must be leak tight to maintain the subatmospheric pressure and to prevent an exchange of gases with the atmosphere. The differential pressure across the boundaries can be maintained by structurally rigid walls or by allowing the walls to be supported by the insulating material. However, the compression of the insulations by the enclosure causes the solid conduction heat transfer to rise significantly and sometimes may completely negate the reduction in heat transfer produced by gas evacuation.

2.3.3 Solid Conduction

Heat is conducted in nonhomogenious materials through the solid particles and between the solid particles where they are in contact. The heat transmitted through the fibers and/or powder-granular particles follows Fourier's law of heat conduction and is dependent upon their shape, size and thermal conductivity. The heat transmitted across the contacts is dependent upon a number of factors, which include but is not limited to particle-to-particle forces, Young's modulus of the material and the roughness of the contacting surfaces. The particle-to-particle forces are variable and strongly dependent on the density of the material and any compression within the material caused by mechanical pressure at the boundaries. Because the heat is conducted through many tortuous paths and across many contacts, the solid conduction component of heat flow is difficult to evaluate accurately without experimental information.

The solid conduction response among foams, powders and fibers is different because of their different mechanical structures. The solid conduction component of rigid foams is generally insensitive to compression induced in the materials by mechanical loads at the boundaries because the material making up the cell walls is continuous in all directions and compression does not greatly affect cross sectional area and length of the heat paths. In powders, the solid conduction component is sensitive to compression because the contact area between particles is increased even though the number of contacts between powder particles in a settled material is reasonably independent of compression load. In fibrous materials, the solid conduction component is more sensitive to compression because both the contact area of contacting fibers and the number of contacting fibers increases with compression. The latter is due to the beam stiffness of the fibers which maintains separation distances.

In powders, the contact area between particles is dependent upon the forces they exert on one another and upon the Young Modulus of the material. Timoskenko* treats the idealized cases of pairs of smooth surface spheres and cylinders forced together by a mechanical load. The contact area, A_c, produced between two elastic spheres pressed together with a force F is a circular area that is proportional to the 2/3 power of the ratio of the force to Young's Modulus, E.

$$A_c \sim \left(\frac{F}{E}\right)^{2/3} \tag{2.4}$$

The same result is obtained when a sphere is pressed against a plane surface and when two cylinders, one at 90 degrees from the other, are forced together. Contacting cylinders with parallel axis produce a contact area:

$$A_c \sim \left(\frac{F}{E}\right)^{1/2} \tag{2.5}$$

The effect of compressive loads on the solid conduction heat transfer in a variety of multilayer insulations was reported by Arthur D. Little, Inc.** The results show that for a particular insulation the heat flow was proportional to a fractional power of compressive load. This power varied among the different insulations from 0.44 to 0.66 showing good agreement with the expected values. From this it was concluded that the thermal conductance of the contacts dominates the solid mode conduction of heat transfer.

In low density fibrous insulations, both the number and size of contacts between fibers increase with increasing load. It can be assumed that the heat flow for the existing contacts will be proportional to a power of the applied load between 1/2 and 2/3. There is experimental

*Timoshenko, S., "Theory of Elasticity", McGraw-Hill Book Company, Inc., New York, 1934, p. 339.

**Arthur D. Little, Inc., "Advanced Studies on Multi-Layer Insulation Systems", NASA Report CR-54929, 1966, p. 11-27.

evidence which indicates that for a specific material this power will increase with increasing load showing the additional effect of the increasing number of contacts. It is obvious, however, that the number of contacts does not increase indefinitely. It is expected that for a particular material there is a compressive load value beyond which there is a negligible number of new fiber contacts made as the compressive load continues to increase.

The thermal conductivity of an evacuated and encapsulated thermal insulation depends primarily on two quasi independent effects: 1, evacuation increases the solid conduction component and 2, evacuation decreases the gas conduction component. The solid conduction component increases because sub-atmospheric pressure in the insulation produced by evacuation results in a compressive load at the encapulation boundaries.

Usually the gas conduction effects become fully developed at a pressure that is far below the pressure at which the solid conduction effects are fully developed. For example, for the case illustrated in Figure 2.1, the component of thermal conductivity related to the gas is reduced by a factor of 10 at about 0.1 mm Hg. On the other hand, the compressive load component is fully developed at 10 mm Hg. Thus, there is no optimum value of pressure for the two effects cited because the effective pressure range of the two phenomena do not coincide.

2.3.4 Radiation

Radiation generally accounts for a small fraction of the heat transfer in insulations currently used in appliances being considered in this study. In some cases, the steps that must be taken to raise the total thermal resistivity of thermal insulations by a factor of three or more will also raise the radiation component. In others, resistivity due to thermal radiation will remain unchanged, and thus become a larger fraction of the total. For example, a perlite powder used in cryogenic insulation has a thermal resistivity of 4 hr ft^2°F/Btu-in, air filled and 138 hr ft^2°F/Btu-in, evacuated. The former resistivity is due mainly to the low air resistivity while the latter is due mainly to the solid and radiation resistivities.

Few solid materials are transparent and then only at specific wave lengths. Glass, a form of silicon dioxide, is transparent in the visual range corresponding to objects radiating at high temperatures, such as an electric light bulb filament. This factor accounts in part for the low thermal resistivity of glass fiber insulations in high temperature applications. Glass is opaque to ambient temperature radiation.

Radiation is transmitted in thermal insulations between two boundaries at different temperatures along two paths. Portions of the two paths often coincide making it difficult to distinguish between them. Thus, one is generally identified as the transmitted radiation and the other as the absorbed radiation, although the latter also results in transmitted thermal energy.

The transmitted radiation can travel directly between the boundaries through the pores in the material or it can interact with the material. When it interacts, it becomes scattered due to the reflection and refraction at the inner and outer boundaries as it passes through the particles. This scattering causes some of the radiation to return to its source, some to go forward to the opposite boundary and some to be directed to the sides. The wave length of the radiation interacting with the solid material remains unchanged.

The absorbed radiation is stopped by the intervening materials and reemitted at wave lengths depending upon the temperature of the material and its thermo-optical properties. The solids are continually emitting in all directions in a wide range of wave lengths which also causes scattering of the absorbed energy. In an insulation with a temperature difference across the boundaries, the net movement of the absorbed and emitted energy is from the warmer to the colder boundaries. Nontransmitting materials such as carbon black absorbers and aluminum flake reflectors are purposely added to high temperature insulation to interrupt and thus to minimize the transmitted components of radiation.

Because of the great complexity of radiation phenomena in thermal insulations, emperical relations have evolved for dealing with the estimation and prediction of this heat transfer mode in specific insulations such as fiber and multi-layer types and in limited temperature ranges. For example, Bankvall* has derived the following relationship for the thermal radiation resistivity R_r of fibrous insulations;

$$R_r = \frac{1}{4\sigma \ell_o \beta T_m^3}$$

$4\sigma T_m^3$ is the radiation coefficient Btu/hr ft^2°F and is derived from the basic radiation heat flow where σ is Stefan-Boltzmann constant and T_m is the mean temperature of the insulation material. The Factor β (dimensionless) is the radiation coefficient which must be emperically determined and is a direct function of the fiber temperatures, diameter and void fraction.

The factor ℓ_o has a meaning similar to pore size and represents the average distance that radiation will travel before being intercepted by the fibrous material. This is evaluated from the expression $\ell_o = \pi d/(1-f_v)$ where d is the average fiber diameter in inches and f_v is the void fraction. Other investigators have derived similar expressions for describing the same phenomena. This expression is valid for materials and temperature conditions that produce a large amount of absorbed radiation and a small amount of transmitted radiation transfer, i.e., glass fibers at ambient temperatures.

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*Bankvall, C.G., "Heat Transfer in Fibrous Materials," Journal of Testing and Evaluation, JTEVA, Vol 1, No. 3, May, 1973, pp 235-243.

The above relation indicates that the thermal radiation resistivity of a fibrous material can be increased by reducing the operating temperatures, pore size and radiation coefficient; the latter two are accomplished by using fibrous insulations with large bulk densities and small fiber diameters. It also illustrates one reason for the low insulation thermal resistivities encountered at high temperatures, i.e., dependence upon T^3.

A similar relation applies to multi-layer thermal insulations. For these, the thermal resistance is given approximately by:

$$R_r = \frac{1}{2\sigma \ell_s \varepsilon T_m^3}$$

Where:

ℓ_s is spacing between shields in inches and ε is the emittance of the shield surfaces and $2\ \sigma T_m^3$ is the radiation coefficient Btu/hr ft^2°F.

2.4 Design Considerations for Air-filled Insulations

Some thermal insulating powders and fibers have thermal resistivities comparable to or greater than still air. This indicates that the resistivity of the solid material is quite large and is confirmed by data obtained with the same materials under vacuum conditions. Examples of materials illustrating this effect are show in Table 2.2. Through appropriate modifications in the size of the particulates it is theoretically possible to increase the resistivity of some of these materials. This is accomplished by reducing the pore size of the material to a fraction of the mean free path of the air molecules at the prevailing temperature while retaining the low solid content of the original material.

The pore size of particulate materials can be made very small in comparison to the mean free path of air by filling the spaces between the larger particles with appropriately sized smaller particles. Applied in successive steps, this process leads ultimately to low porosity and small pore sizes. For example, Furnas* demonstrated that it is possible to produce a bed of particulate materials having a porosity of 2.5 percent using only four different sized particles in proper proportions such that the diameter of the largest particle is 10^5 times greater than the diameter of the smallest particle. The pore size is directly related to this porosity and to the diameter of the smallest particles in the blended materials. This beneficial effect is offset by the increase in the bulk density of the material which approaches the true density of the solid particles as the porosity is made

*Furnas, C.C., "Grading Aggregates", Journal of Industrial and Engineering Chemistry, Vol. 23, No. 9, 1931.

Material	Thermal Resistivity* (hr ft^2°F/Btu-in) Filled with Air at One Atmosphere	Evacuated to 10^{-2} mm Hg	Bulk Density (lb/ft^3)
Air	5.6	N/A	0.072
Perlite, Granular	4.0	138	8.7
Silica Aerogel Powder	8.0	49	6.0
Silicon Dioxide Powder	7.0	69	3.3
Glass Fiber, Type AA	8.0	417	15.0

*Data taken from Table 2.1 except for air.
Refer to Table 2.1 for temperature conditions.

TABLE 2.2 THERMAL RESISTIVITY OF POWDERS AND FIBERS

successively smaller. However, the large amount of solid lowers the solid resistivity of the material counter to the increase in the air resistivity produced by the small pores.

The high bulk density and solid resistivity effects can be counteracted by using hollow particles such as perlite instead of solid particles. Perlite is a granular silicate obtained from lava ore expanded under heat. The particles range in size from about 50 to 1000 microns, are irregular in shape, and have interior voids which are open to the outside. Thermal conduction in the particle is greatly reduced because of its thin wall structure. As an evacuated insulation in cryogenic equipment, Perlite is a very effective thermal insulator having thermal resistivities of about 140 hr ft^2°F/Btu-in at vacuum pressures of about 10 microns. This is evidence of the low level of thermal conduction taking place within the solid material. However, filled with air at atmospheric pressure, the thermal resistivity rises to 4 hr ft^2°F/Btu-in. The lowering of the thermal resistance is due to the presence of air inside as well as outside of the particles. Assuming that the thermal conduction of the gas outside of the particle can be reduced by filling the spaces with smaller particles to reduce the pore size, nothing can be done to reduce the thermal conduction of the gas within the particles because the interior dimensions are at least 1000 times larger than the mean-free path of air molecules. Thus, hollow particles larger than the mean free path of air do not provide a solution for achieving high bulk thermal resistivities.

Particles with internal cellular structures, small enough to reduce the gas spaces below the mean free path of air, would appear to provide a solution to the large voids found in perlite and other hollow spheres. Analysis indicates that a bulk thermal resistivity of 20 hr ft^2°F/Btu-in would require a cellular material having cells of about 6×10^{-3} microns in diameter and cell walls of about 5×10^{-5} microns thick. The latter is clearly unattainable as the molecular diameters of some common gases are in the 2×10^{-4} to 6×10^{-4} micron range. Further, though there are many cellular insulating materials, such as foams, cork and seaweed, none have the necessary cell sizes indicated above. Vermiculite, a micaceous material that exfoliates when heated, is made up of parallel spaced platelets. This material has the appropriate wall distances in one dimension only; the other dimensions are quite large, typically 100 to 10,000 microns. Thus, cellular materials whether in plank form or in particle form are not appropriate for producing high thermal resistivities in air at atmospheric pressure.

Each of the foregoing concepts for producing high thermal resistivity materials has at least one theoretical limitation. One concept, to be considered now, appears to meet all the theoretical requirements and has been reported in the patent literature as technically demonstrated. This concept involves the use of fine powders consisting of solid particles that are modified to produce pore sizes smaller than the mean free path of air.

The production of small pores requires small particles; the approximate pore size for particulate beds of fixed sized spheres and cylinders is:

$L_s = .66\ d/f_s$ for spherical particles

$L_c = 1.18\ d/f_s$ for cylinderical particles

where:

L = pore size

d = particle diameter

f_s = solid fraction.

The pore size for spherical and cylindrical particles having diameters of 0.003 microns are shown in Figure 2.2 as a function of bulk density for silicon dioxide (specific gravity of 2.2). The pore sizes become smaller than the mean free path of air at 70°F at bulk densities larger than 4-6 lb/ft^3. At bulk densities of 16 lb/ft^3 the pore sizes are 1/2 to 1/4 the mean free path of air. Beyond this bulk density, small reductions in pore size are obtained only with significant increases in bulk density of the material.

Data are presented in U.S. Patent 4,159,359 showing the increased resistivity obtained with compressed fumed silica particles having mean diameters in the range of 0.0025 to 0.005 microns. At a density of 14 lb/ft^3, a 0.005 micron particle size material produced a thermal resistivity of about 20 hr ft^2°F/Btu-in, filled with air at one atmosphere and at a mean temperature of -120°F. Smaller sized particles in the 0.0025 to 0.0030 micron range produced a thermal resistivity of about 28 hr ft^2°F/Btu-in under the same conditions. These data demonstrate the technical possibility for producing higher than normal resistivities using compacted beds of small solid particles. The effect of temperature on these materials is not available in the published data.

2.5 Design Considerations

2.5.1 Leak Detection

Construction of an evacuated panel which will last for a period of years requires the ability to detect very small leaks at the manufacturing stage. All panels will leak to some degree, and this will cause the internal gas pressure to rise. This pressure rise must not cause an unacceptable deterioration in resistivity at the end of appliance life. It is assumed that the panel will be evacuated initially to at least a tenth of the final acceptable pressure.

For a given panel, the following approximate relationship can be used to establish the leak detection capability required at the manufacturing stage:

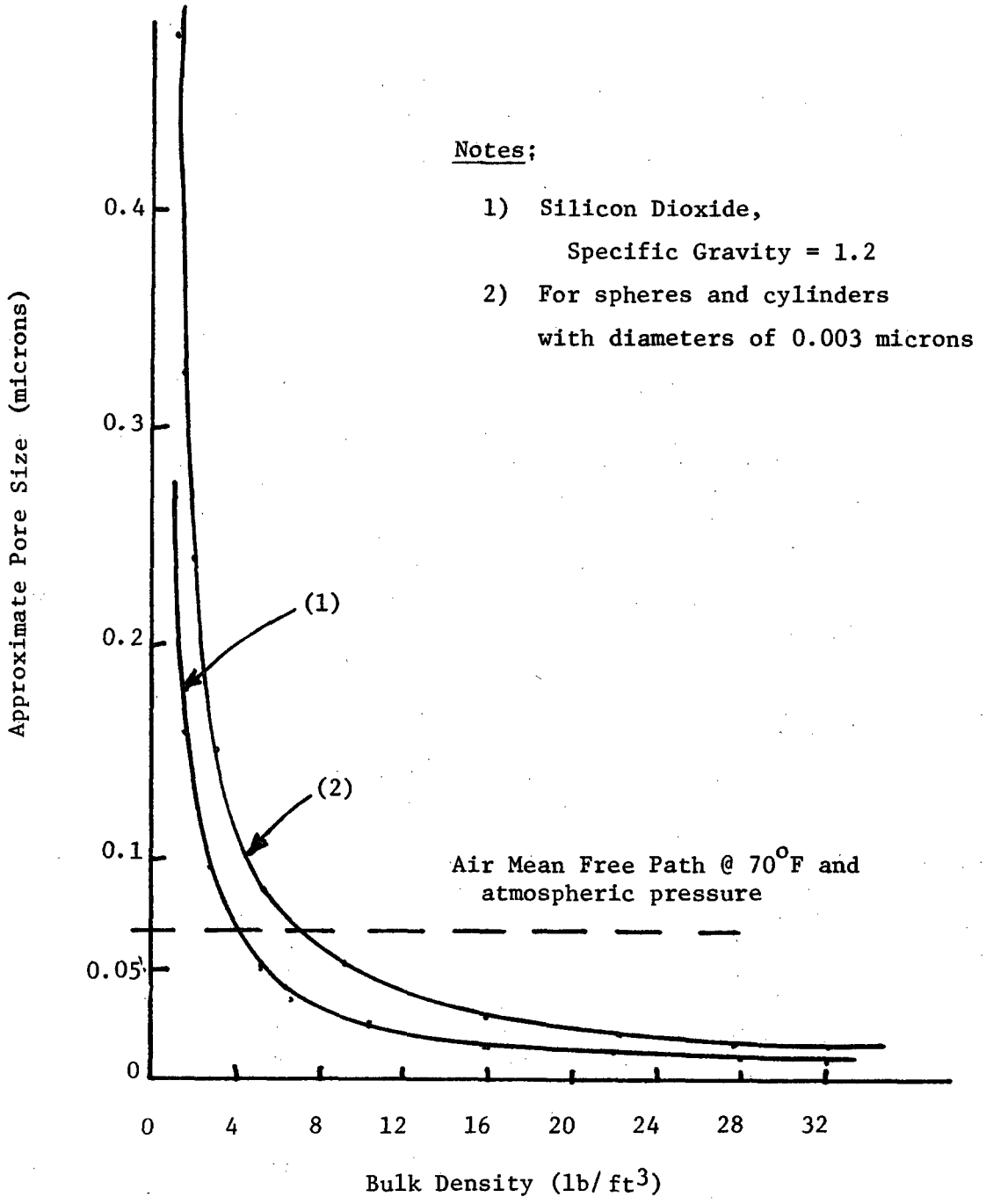


FIGURE 2.2 PORE SIZE AS A FUNCTION OF BULK DENSITY.

$$Q = 1.141 \times 10^{-8} \frac{PVfv}{TY}$$

where:

Q = leak detection capability required; Standard cubic centimeters per second

P = Maximum allowable internal pressure of the panel; mm Hg

V = Panel Volume, cubic meters

fv = Void fraction of the encapsulated material; dimensionless

T = Mean temperature of the panel; K

Y = Expected panel life; Years.

Figure 2.3, based on this equation, can be used to estimate the leak detection capability in standard cubic centimeters per second for a panel 2.5 inches thick, 10 feet square and having a 10 year life time*. For example, if the maximum allowable internal pressure of the panel is 10^{-1} mm Hg then the leak detection capability must be at least 2.3 X 10^{-8} standard c^3/s. This type of measurement requires specialized equipment and personnel and could consume significant time during the manufacturing process. The level of leak detection required has been used to establish encapsulation classes as described in Section 2.5.3.

2.5.2 Insulation Encapsulation

Encapsulation isolates the insulation materials from the gases in the environment and limits the rate at which they are transmitted into the enclosure. The gas pressure rise within an evacuated insulation is due mainly to leakage and/or permeation through the encapsulating material. The gas pressure rise may also result from the desorption of gas from all the surfaces within the enclosure including insulation and encapsulation materials. Gas leakage and permeation are controlled by the encapsulating materials and the joining and sealing methods used. It results from the use of materials that contain pinholes or are gas permeable. The absorbed gases are controlled by special treatment of the materials prior to encapsulation such as the use of solvent washing and vacuum bake-out techniques.

Three classes of encapsulating materials have been identified. These are summarized in Table 2.3. Each class is distinguished by the maximum leakage rates that would cause serious degradation of the insulation at the end of a 10 year period.

Class 1 encapsulation covers applications in which the pressure on both sides of the material is atmospheric. The major requirement for the Class 1 encapsulation material is low permeability.

Class 2 encapsulation is required to be impermeable and leak free to a very high degree for the life of the appliance. Establishing 10 mm Hg as the highest internal panel pressure permitted at the end of life, a

*The use of a gettering material extends the life.

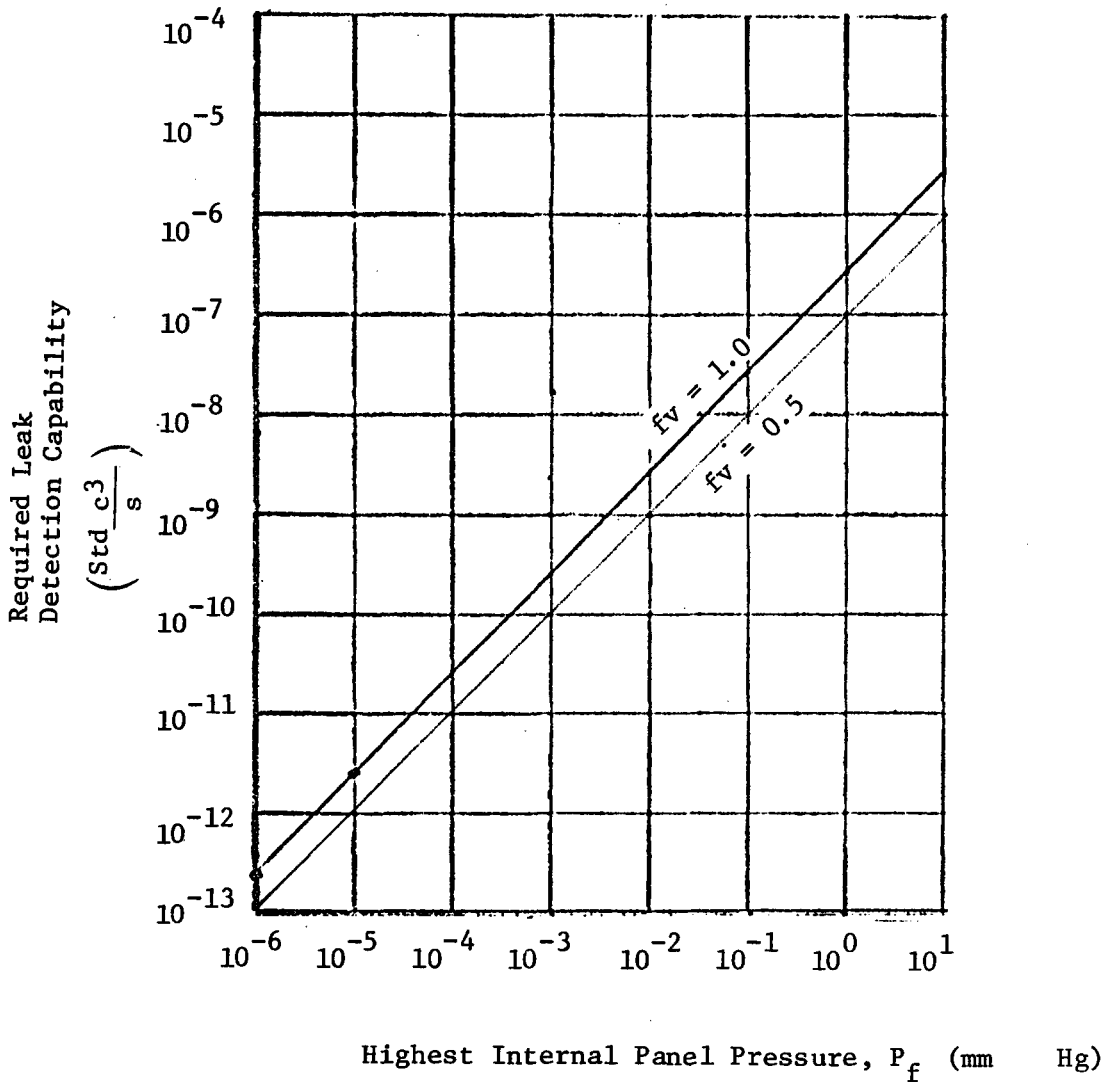


FIGURE 2.3: LEAK DETECTION CAPABILITY NEEDED FOR EVACUATED PANELS 2.5 INCH THICK, 10 SQUARE FEET IN AREA AND A 10 YEAR LIFE VERSUS END OF LIFE PRESSURE

TABLE 2.3 ENCAPSULATION CLASSES AND SUGGESTED MATERIALS

Encapsulation Class 1

1. Function:

   To seal the surfaces of insulating materials against the diffusion of atmospheric gases. Atmospheric pressure on both sides.

2. Description:

   A membrane continuously bonded to and covering all the surfaces of rigid closed cell thermal insulation sections, having a low permeability to the atmospheric gases. Localized leakage at pin holes, joints, and seams are permitted.

3. Typical Materials:

   3.1 Aluminum foil, 0.001 - 0.002 inches thick.

   3.2 Aluminum foil, 0.001 - 0.002 inches thick bonded to Kraft paper for strength and durability.

Encapsulation Class 2

1. Function:

   To seal open type insulating materials from atmospheric gases and maintain them in an evacuated condition. Maximum allowable pressure required after 10 years is 10 mm Hg.

2. Description:

   An evacuated enclosure that is capable of maintaining an internal pressure in the range of 0.001 to 1.0 mm Hg for a period of 10 years or more. The enclosure is supported by the insulating material and is to have low gas leakage and diffusion rates and all joints, seams and connections are to have low gas leakage rates.

3. Typical Materials:

   3.1 Pinhole-free sheet steel and welded joints, seams and connections.

   3.2 Aluminized Mylar laminates with adhesive sealed joints, and connections.

TABLE 2.3 (Continued)

Encapsulation Class 3

1. Function:

   To seal open type insulating materials from atmospheric gases and maintain them in an evacuated condition. Maximum allowable pressure required after 10 years is 10^{-3} mm Hg.

2. Description:

   An evacuated enclosure that is capable of maintaining an internal pressure of less than 10^{-3} mm Hg for 10 years or more. The enclosure is supported by the insulating material and is to have essentially zero leakage and diffusion through the surface and essentially zero leakage at the joints, seams, and connections.

3. Typical Materials:

   3.1 Pinhole-free sheet steel and welded joints, seams and connections.

leak detection capability of about 10^{-6} standard c^3/s of air is required. This can be achieved on a routine basis during manufacture, although actual panel test and repair times may be lengthy. This leakage rate makes no allowance for gas diffusion through the membrane or for outgassing of the evacuated materials.

Class 3 encapsulation is required to be impermeable and leak free to a higher degree than Class 2. This class would be required for high performance compressed multi-layer thermal insulations. Although such an insulation may be evacuated to 10^{-5} mm Hg, its performance would be necessarily compromised if leakage and outgassing caused the internal gas pressure to rise to about 10^{-3} mm Hg. The leak detection capability required for a 10 square foot, 2.5 inch thick, 10 year life panel is about 10^{-10} standard c^3/s. To test and repair evacuated panels to this level is beyond routine mass manufacturing practices and leakage detection systems since leakage rates can be measured on a routine basis only at about 10^{-7} standard c^3/s. These leakage rates make no allowance for gas diffusion through the membrane or for outgassing of the evacuated material. If these effects are significant, the leakage rates must be made smaller than indicated above.

2.5.3 Integration of Evacuated or Sealed Insulation Systems and Appliances

The insulation may be encapsulated using the inner and outer walls of the appliance or it may be encapsulated separately. In the former case, the material used is generally sheet steel, which serves many purposes including structural and aesthetic as well as encapsulation for the insulation. When separate encapsulation is used, the insulation will be encapsulated before it is integrated with the walls of the appliance using one of the methods discussed above.

The insulation, with or without encapsulation, can be integrated with the appliance in a number of ways. Four integration methods considered are shown in Figure 2.4 and are described below:

- Concept 1

This concept applies to appliances with rectangular shapes. The appliance is made up from flat encapsulated insulation panels which are joined at the corners to form the appliance walls. Doors, for those appliances requiring them, are made up from single flat encapsulated panels.

- Concept 2

This applies to appliances with rectangular shapes. This is the box within a box concept. A five sided box is placed within a larger five sided box and the two are joined together at the free edges. The two boxes encapsulate the insulation. Doors, for those appliances requiring them, are made up from single flat encapsulated panels.

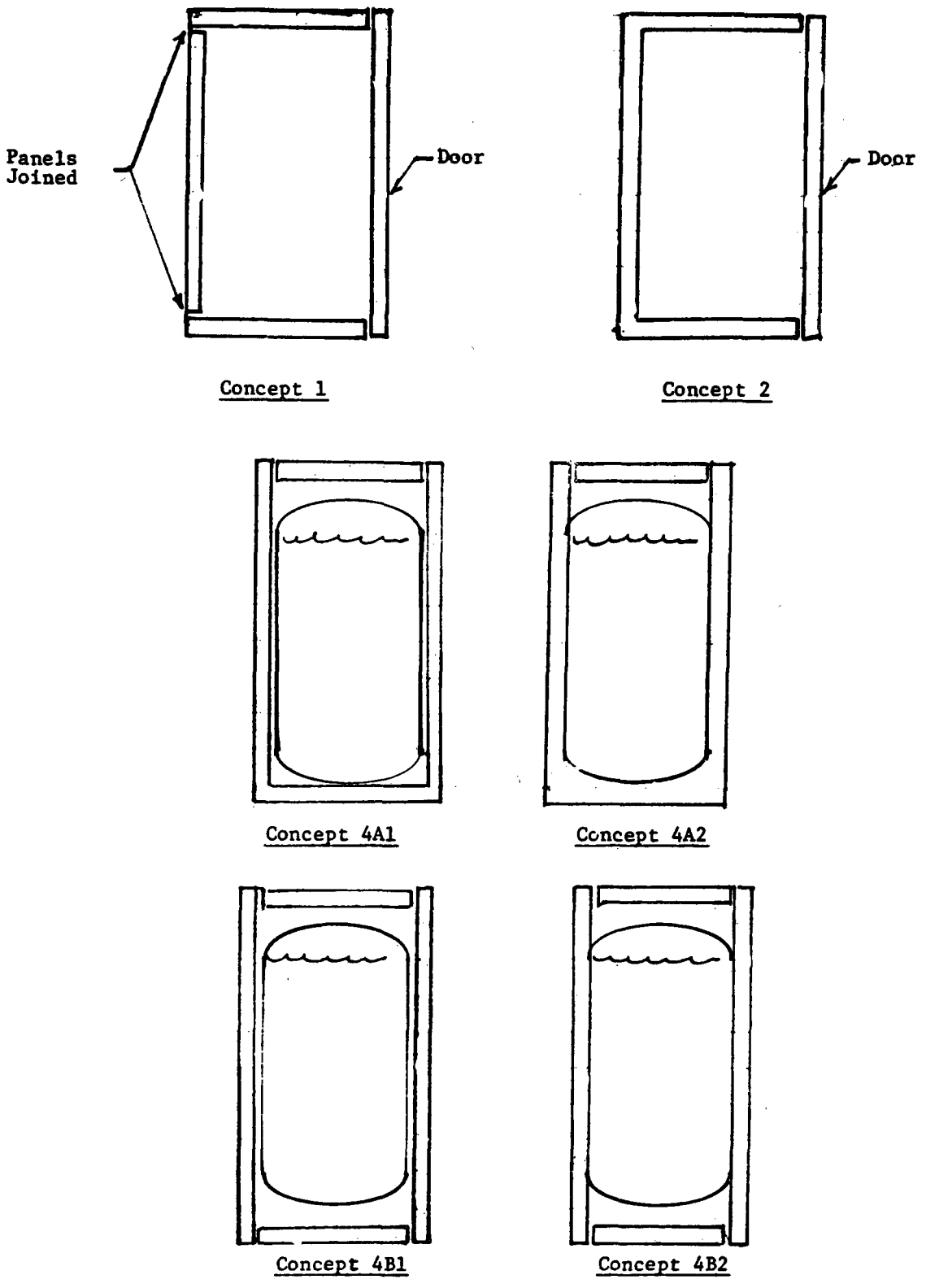


FIGURE 2.4 INTEGRATION CONCEPTS

- Concept 3 (Not Illustrated)

This is identical to Concept 2 except that the encapsulated panels are placed within the spaces formed by the inner and the outer box. The inner and outer walls of the appliance can be used for sealing if a rigid cell foam is formed in place (Encapsulation Class 1) (around the panels). Doors, for those appliances requiring them, can be made directly from flat encapsulated insulation panels or these panels can be inserted into door panels.

- Concept 4

This concept applies to appliances which are cylindrical in shape such as water heaters. It is similar to concept 2 and has two main configurations. Configuration "A" consists of two cylindrical cans, each with one end removed and the smaller one inside the larger, both being joined together and sealed at the free edge. The space between the cans contains the insulation material. The inner container could serve as the water reservoir if the open end were closed. The insulation of the open end can be done using one of the concepts identified in this program or using current insulation technology.

Configuration "B" consists of two cylindrical shells, a smaller shell inside a larger shell and both joined together at the free ends of each cylinder. The space between the shells can contain or encapsulate the insulation material. Further, the inner container can serve as the water reservoir when both ends are capped several inches away from the ends of the shell. The insulation of the open ends can be done using one of the concepts identified in this program or using current insulation approaches.

2.5.4 Edge Losses

In evacuated systems using steel or other materials with relatively high thermal conductivities, edge effects can significantly reduce the overall R value of the insulation system. The edge acts like a thermal short which conducts heat around the insulation. As an example, a 2 in thick by 2 ft by 6 ft panel with a thermal resistivity of 20 hr ft^2 °F/Btu·in encapsulated in 1/32 in thick steel would give an overall R of 1.8 $\frac{hr\ ft^2 °F}{Btu \cdot in}$ Using 1/64 in thick stainless steel would increase this value to 7.7. Thus, to minimize edge effects, the encapsulation method and its integration with the appliance are of primary importance. To reduce these effects, conduction paths should be long, thin metal possibly laminated with plastic for strength or other low conductivity material should be used, and the number and length of edges should be minimized.

2.6 Insulation Concepts

The following pages describe insulation concepts developed or identified during this program. Table 2.4 summarizes these results. A description of each concept is given on the pages following the summary. Where specific materials for implementing the concept could be identified they are shown.

Properties	Concept Number											
	I	II	IV	V-1	V-2	VI	VII	VIII	IX	X	XI	XII
1. R-Value Potential (R/in)	8	8	15	14	∿20	20	20	556	15	>20	>20	>20
2. Temperature Limitations (°F)	150	250	250	>600	>600	>600	>600	200	>600	>150	150	>400
3. Bulk Density (lb/ft^3)	2-3	2-3	2.4-3	20	15	9	12	16	6.5	7	2.5	10
4. Cost Relative to I*	1.0	2.0	2.5	487	22	1.0	51	36	4	1.5	3.5	3
5. Interstitial Gas Pressure (mm Hg)	ATMOS	ATMOS	10^{-2}	ATMOS	ATMOS	10^{-1}	10^{-4}	10^{-5}	10^{-6}	10^{-1}	10^{-5}	10^{-1}
6. Compressibility (%)	>10	>10	>10	∿10	N/A	11	∿10	>>10	>10	>10	>>10	>>10
7. Leak Detectors Level (Std c^3/s)	N/A	10^{-5}	10^{-9}	10^{-5}	N/A	10^{-8}	10^{-11}	10^{-12}	10^{-13}	10^{-8}	10^{-12}	10^{-8}
8. Encapsulation Close	1	2	2	2	1	2	3	3	3	2	3	3
9. Insulation Material	PU Closed Cell Foam	Phenolic Open Cell Foam	Phenolic Open Cell Foam	Min K-1400	Compressed Fumed Silica Powder	Perlite Powder	Ceramic Fiber Paper	Multi-layer Shields and Spacers	Hollow Glass Micro-spheres	Perlite Powder and PU Foam	Multi-layer Insulation and Foam	Perlite and NEMA-GIO Honeycomb Composite

*Relative costs are for materials only and are based on an insulation volume of 13 Cubic Feet and an insulation area of 115 Square Feet.

TABLE 2.4 SUMMARY THERMAL INSULATION CONCEPTS

CONCEPT I

This is a concept that is in wide use in many industries and products. It consists of a closed cell rigid plastic foam filled with a low conductance gas at about atmospheric pressure. The material is available with bonded covers of aluminum foil or it can be foamed in place between the inner and outer walls of systems requiring thermal insulations. Either method seals over 90 percent of the foam's surface area which greatly reduces air permeation into the foam and a subsequent loss in thermal performance. The best example of this insulating material is polyurethane foam blown with fluorocarbon 11. Because of its wide use and application, this concept serves as a basis for comparison with other concepts considered and evaluated in this study.

The limitations of this concept are specific to the foam material. Polyurethane and similar polymeric foams have a limiting value of R/in of about 8 at room temperature. Further, many structural properties of polymeric foams are seriously degraded at temperatures above 150°F. These considerations limit the application of polymeric foams, e.g., freezer and refrigerator appliances. Materials such as fiberglass and mineral wool, whose insulation properties are inferior to the foam mentioned, must be used in applications such as ovens because they are compatible with the peak operating temperatures.

A number of modifications to the foam material to improve its resistivity have been considered. For example, a reduction in the cell size from about 250 microns to .01 microns could theoretically reduce the component of heat flow in the insulation due to gas conduction. However, the cell wall's thickness would be of molecular dimensions. This is clearly impractical. A second consideration is the lowering of the bulk density of the material to a fraction of present value (2 lb/ft^3) in order to reduce the component of heat flow in the insulation due to solid conduction. This too is impractical because the open cell content increases dramatically and the structural properties are seriously degraded. The use of exotic gases with lower thermal conductivities than R-11, such as Krypton, Xenon, and Radon is not practical because of cost, availability and other problems. One last example is the use of low density foam materials having lower thermal conductivities than urethane. A preliminary evaluation indicates small improvements are possible in this area.

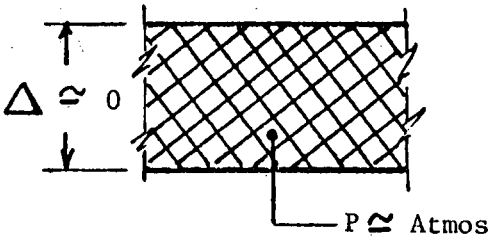
Specification No. I

1. Insulation Concept

1.1 Concept: I

1.2 Description: Closed cell rigid plastic foam filled with low conductance gas at about 15 psia and sealed at the surfaces with a bonded, gas-impermeable, encapsulation membrane. The best example of this system is R-11 blown polyurethane foam.

1.3 Figure:



2. Insulation

2.1	Material	Polyurethane
2.2	Structure	Foam
2.3	Closed/Open Cell	Closed cell
2.4	Bulk density (in service)	2 (lb/ft^3)
2.5	Temperature limit (°F)	∿ 150
2.6	Deflection at 15 psi comp.	<< 10%
2.7	Cell or interstitial gas	R-11
2.8	Gas pressure	∿ atmosphere
2.9	Dominant Failure mode	Air diffusion into cells
2.10	Material ($/lb)	0.70
2.11	Estimated R-Value (R/in)	∿ 8

3. Encapsulation

3.1	Class:	1 - (Aluminum foil)
3.2	Materials cost ($/$ft^2$):	.06
3.3	Temperature Limitations (°F):	150

CONCEPT II

This concept consists of an open cell rigid plastic foam filled with low conductance gas at atmospheric pressure. Gas retention must be maintained by encapsulation with a leak-free and impermeable material.

We would estimate the R-values of Concepts I and II to be comparable. However, Concept II is inferior to Concept I in a number of areas. The encapsulation of the foam must be sealed to prevent dilution of the low conductivity gas with air which will degrade the thermal performance of the the insulation. This requires leak tight construction using a low permeable material. Such construction will introduce requirements for increased levels of technology such as sealing and testing for gas tightness. Further, cycling barometric pressure and/or changes in mean insulation temperature will cause the encapsulation to bulge and distort the appliance because of the pressue imbalance between the sealed insulation spaces and the external environment. Thus, we believe this system shows no promise for improvement over the current standard practice.

Specification No. II

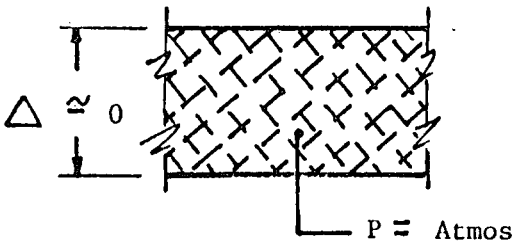
1. Insulation Concept

1.1 Concept: II

1.2 Description: Open cell rigid plastic foam filled with low conductance gas at about 15 psia and encapsulated in a gas-impermeable and leak-free encapsulation membrane. R-value ≤ 8.

Concept not pursued

1.3 Figure:



2. Insulation

2.1	Material	Phenolic
2.2	Structure	Foam
2.3	Closed/Open Cell	Open Cell
2.4	Bulk density (in service)	2.5 - 3.0 lb/ft^3
2.5	Temperature limit (°F)	250°F
2.6	Deflection at 15 psi comp.	~ 3%
2.7	Cell or interstitial gas	R-11
2.8	Gas pressure	Atmospheric
2.9	Dominant Failure mode	Pin hole in encapsulation
2.10	Material ($/lb)	~ 1.0
2.11	Estimated R-Value (R/in)	~ 8

3. Encapsulation

3.1	Class:	2
3.2	Materials cost ($/ft^2):	0.07 - 0.15
3.3	Temperature Limitations (°F):	250

CONCEPT III

The surfaces of the foam are sealed with a bonded gas-impermeable encapsulation to prevent air diffusion into the cells. In the discussion of Concept I, the reduction of the cell size to below the mean free path of R-11 gas was considered impractical. Concept III considers increasing the mean free path of the gas to 10 times the cell size by lowering the gas pressure within the cell through condensation of the foaming vapor. During manufacture, the foam is formed at a temperature substantially above the ambient temperature using a fluid that is a gas at the blowing temperature and which condenses when the foam is cooled. The vapor pressure of the condensed phase is subatmospheric. Referring to Figure 2.1, 0.1 mm Hg will reduce the thermal conductivity of the gas to one tenth its value at standard conditions if the cell size is 100 microns. For example, a 250 pores per inch (ppi) foam (100 micron diameter cells) would require a gas with a mean free path (mfp) greater than 1000 microns. Data for two gases, representative of the ones considered, are tabulated below. The estimated mfp of each of these gases at 80°F is 100 or more times smaller than the required value. Because foams with smaller cells are not practical, this approach is not technically feasible. In addition, the foaming temperatures, corresponding to the fluid boiling points are significantly above the forming temperature of normal foam plastics. Because of its low potential, this approach was not considered further.

Fluorocarbon Materials

Company	3M Co	DuPont
Material	FC40	F112
Molecular Weight	650	204
Boiling Point (°F)	311	199
Vapor Pressure (mm Hg)	3 (at 77°F)	46 (at 70°F)

CONCEPT IV

The elements of this concept are a rigid open cell foam and an evacuated gas-impermeable, leak-free encapsulation. This concept is a further attempt to utilize the low thermal conductivity of polymeric foam materials to achieve high R-values by eliminating gas conduction by evacuation. The open structure of the foam permits the pore area to be evacuated and the rigidity of the foam, derived from its strength properties, allows it to carry the atmospheric pressure forces acting on the encapsulation.

The reviewed literature contained no reference or property information for rigid open foams. We estimate, the potential maximum thermal resistivity attainable in this concept at about 15 hr ft^2°F/Btu·in by assuming a polyurethane based material or a material with a similar low solid thermal conductivity.

A practical difficulty in implementing this concept is the availability of the foam material because rigid foams tend to be closed rather than opened celled. For example, polyurethane foams at densities less than 1.8 lb/ft^3 can have a large percentage of open cells but they do not possess the compressive strength required to support the atmospheric pressure loads. The closed cell content of polyurethane foams increases with density. Foams having the appropriate compressive strength also have a large fraction of closed cells.

Phenolic foams with 90 percent and greater open cell content are becoming available. Laukes of England and Oasis in the U.S. manufacture phenolic foams with a density of about 1 lb/ft^3 and compressive strengths up to 20 psi. The thermal, strength and other relevant properties of these foams are not currently available and should be investigated further to establish their potential in this insulation concept.

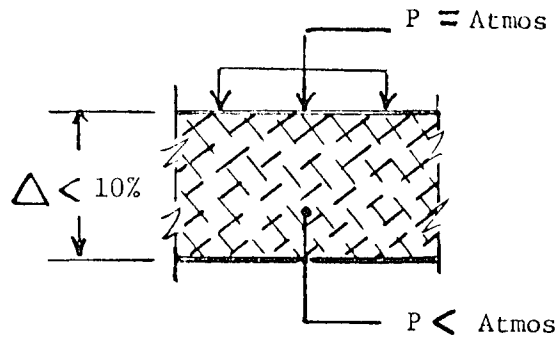
Specification No. IV

1. Insulation Concept

1.1 Concept IV

1.2 Description: Evacuated open cell, rigid plastic foam encapsulated in a gas-impermeable and leak-free membrane.

1.3 Figure:



2. Insulation

2.1	Material	Phenolic
2.2	Structure	foam, board stock
2.3	Closed/Open Cell	open
2.4	Bulk density (in service)	2.5 - 3.0 lb/ft^3
2.5	Temperature limit (°F)	250°F
2.6	Deflection at 15 psi comp.	∿3%
2.7	Cell or interstitial gas	air
2.8	Gas pressure	10^{-2} mm Hg
2.9	Dominant Failure mode	leak, outgassing, diffusion
2.10	Material ($/lb)	1.0
2.11	Estimated R-Value (R/in)	15

3. Encapsulation

3.1	Class:	2 (steel)
3.2	Materials cost ($/$ft^2$):	0.21
3.3	Temperature Limitations (°F):	>600

CONCEPT V

This concept uses fiber, powder or granular materials filled with a low conductance gas at atmospheric pressure in a containment which is gas-impermeable and leak free. This concept is identical to Concept II in all respects except that a particulate material is used here instead of the rigid, open-cell foam of Concept II. The combination gas and particulate materials produce a high system thermal resistivity by virtue of the high thermal resistivity of the components.

The considerations discussed in Concept II are applicable here regarding the use of low thermal conductivity gases. The potential R-value of this concept is dependent upon many factors including the selection of the insulating material and the gas filler. This concept is not unique, for the general literature on thermal insulations contains many references on the use of low thermal conductivity gases, but there are limited thermal performance measurements reported for the concept in the literature. Specification V-1 is a commercially available representation of this concept.

This concept has an additional degree of freedom compared to Concept II. Concept II is limited by the pore size that can be produced in a foam material in the range of 25 to 250 microns. However, with Concept V, it is possible to use a material consisting of extremely small particles producing a range of pore size less than the mean free path of gases at atmospheric conditions, i.e., about 0.07 microns. Theoretical considerations indicate that the apparent thermal resistivity of the gas fillers would be greatly increased from their normal values and result in a high thermal resistivity insulation.

The use of air with such a material would have the additional advantage over other gases of not requiring an impermeable and leak tight encapsulation.

An attempt was made in this program to achieve a fine pore material producing a high thermal resistivity in air. Air has a mean free path of about 0.07 microns at room temperature. The desired pore size was about one tenth this value. A fumed silica was selected that consists of spherical particles, 0.007 microns in diameter, joined together to form low density aggregates This material has a nominal manufactured density of 4-5 lb/ft^3 and an estimated pore size of about twice the mean free path of air. We ball milled this material to higher densities (up to 33 lb/ft^3) in order to reduce its pore size. The intent of this operation was to reduce the size of the aggregates so that they would nest closely together yielding a material with a higher density and smaller pore size than that of the original material.

Five thermal conductivity samples were prepared, one with the as-received material and four ball milled to densities ranging from 10.9 to 33 lbs/ft^3. The measured thermal conductivities are presented in Table 2.5. The expected increase in the thermal resistivity was not achieved. Photomicrographs of the tested materials revealed that the ball milling agglomerated and densified the primary particles; the longer the

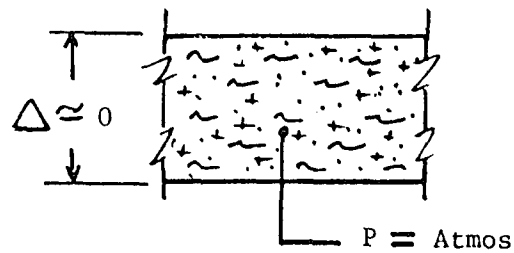
Specification No. V

1. Insulation Concept

1.1 Concept V-1

1.2 Description: Fiber, powder or granular materials filled with low conductance gas at about 15 psi and encapsulated in a gas-impermeable and leak-free membrane.

1.3 Figure:



2. Insulation

2.1	Material	Min K-1400
2.2	Structure	Silica fiber and powder
2.3	Closed/Open Cell	Open
2.4	Bulk density (in service)	20 lb/ft^3
2.5	Temperature limit (°F)	> 600
2.6	Deflection at 15 psi comp.	N/A
2.7	Cell or interstitial gas	Xenon
2.8	Gas pressure	Atmospheric
2.9	Dominant Failure mode	Gas diffusion leaks
2.10	Material ($/lb)	46.75
2.11	Estimated R-Value (R/in)	14

3. Encapsulation

3.1	Class:	2 (steel)
3.2	Materials cost ($/$ft^2$):	0.21
3.3	Temperature Limitations (°F):	> 600

DENSITY (lb/ft^3)	APPARENT THERMAL CONDUCTIVITY ($Btu \cdot in/hr\ ft^2 °F$)	APPARENT THERMAL RESISTIVITY ($hr\ ft^2 °F/Btu \cdot in$
5.6	0.18	5.6
10.9	0.21	4.8
13.7	0.23	4.4
18.8	0.24	4.1
33.0	0.35	2.9

Notes:

- Cab-O-Sil S-17 from Cabot Corporation, Billerica, Ma.
- Tested at a mean temperature of 75°F and atmospheric pressure.
- 5.6 $lb.ft^3$ sample was "as received" from Cabot.

TABLE 2.5 THERMAL PERFORMANCE OF BALL MILLED FUMED SILICA

period of ball milling, the larger and more dense the agglomerates. The photographic evidence indicates further that ball milling reduced the material pore size, but not uniformly. The space between the particles generated pores, larger than those in the untreated material, counteracting the small pores within the particles. This twofold effect on apparent thermal conductivity of the gas resulted in a small net decrease in thermal resistivity with density. This is evident from data in Table 2.5 where a density increase of 490 percent produced a thermal resistivity decrease of 49 percent. We would expect a rise in the resistivity of the material if the pore distribution were uniformly small.

Support for the concept has been found in U.S. Patent 4,159,359, awarded June 26, 1979. The patent illustrates that thermal resistivities of 21 and 28 hr ft^2°F/Btu·in have been obtained in air with compressed powdered silicates that have particle sizes of .005 microns and .0025 to .0035 microns respectively at bulk densities of about 14 lb/ft^3. The compression method, described in the patent, produces a uniformly small pore in the material. This in turn results in an increase in thermal resistivity to a level sought in this program. Specification V-2 is a possible representation of this concept.

The data referred to from Patent 4,159,359 is evidence for the theoretical feasibility of obtaining thermal resistivities greater than 20 hr ft^2°F/Btu·in with air filled micropore thermal insulations. It must be noted, however, that these data were obtained at a mean temperature of -121°F for use in cryogenic applications. It is well known that the thermal resistivity of insulating materials generally decreases with temperature. Thus, the resistivity of the same materials at the operating temperatures of appliances considered in this study will be less than those indicated.

U.S. Patent 4,159,359 is assigned to L'Air Liquide of France. Correspondence with them indicates that current production capacity accommodates the requirements for test samples only. By November, 1980, they expect that their thermal performance tests, application and cost studies will be completed and available for distribution to potential licensees.

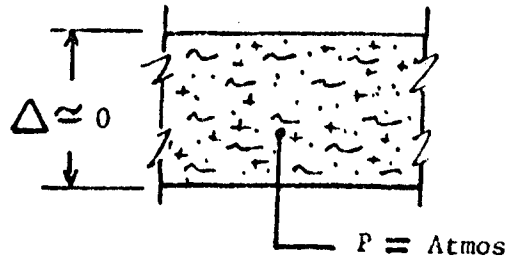
Specification No. _V_

1. **Insulation Concept**

1.1 **Concept** V-2

1.2 **Description:** Fiber, powder or granular materials filled with air at about 15 psi and encapsulated in a membrane to support powder. Good at elevated temperatures.

1.3 **Figure:**



2. **Insulation**

2.1 Material	Fumed Silica
2.2 Structure	Compressed Powder
2.3 Closed/Open Cell	Open
2.4 Bulk density (in service)	15 lbs/ft^3
2.5 Temperature limit (°F)	>600
2.6 Deflection at 15 psi comp	N/A
2.7 Cell or interstitial gas	Air
2.8 Gas pressure	Atmos
2.9 Dominant Failure mode	Gas Diffusion Leaks
2.10 Material ($/lb)	2.00
2.11 Estimated R-Value (R/in)	20

3. **Encapsulation**

3.1 Class:	1 (Steel)
3.2 Materials cost ($/ft^2):	0.21
3.3 Temperature Limitations (°F):	>600

CONCEPT VI

In this concept, an evacuated powder or granular material is encapsulated in a gas impermeable and leak free membrane. Evacuation increases the mean free path of the interstitial gas. As indicated previously, when this path is made a tenth or less of the material pore size the apparent thermal resistivity of the material is increased by up to two orders of magnitude. However, the unbalanced gaseous pressure forces introduced at the encapsulation boundaries compress the powder causing its thermal resistance at the particles' boundaries to decrease. The net result depends upon which of the two changes dominates.

No data were found in the open literature relating to the thermal resistivity of compressed particulate materials. However, data were obtained from private sources and are presented in Table 2.6. This includes thermal resistivity values for seven materials in the range from 20.4 to 76.9 hr ft^2°F Btu·in and for density in the range of 2 to 20 lbs/ft^3. Percent deflection at a compressive load of 15 psi ranges from 1.5 to 11 percent. These data attest to the techncial feasibility of this concept.

DATA SOURCE	DESIGNATION	DENSITY (lb/ft^3)	TEMPERATURE MEAN (°F)	TEMPERATURE BOUNDARIES (°F)	THERMAL CONDUCTIVITY ($Btu \cdot in/hr\ ft^2 °F$)	RESISTIVITY ($hr\ ft^2 °F/Btu \cdot in$)	GAS PRESSURE (mm Hg)	PERCENT COMPRESSION at 15 psi LOADING	MAXIMUM APPLICATION TEMPERATURE
Owens-Corning	N/A	N/A	N/A	N/A	0.030	33.3	N/A	N/A	N/A
UCC/ Linde	Linde-A	2	51	32-10	0.030	33.3	$10^{-3} - 10^{-1}$	1.5	150
UCC/ Linde	Linde-B	12	110	10-150	0.021	47.6	$10^{-3} - 10^{-1}$	10	450
UCC/ Linde	Linde-C	20	210	70-350	0.013	76.9	$10^{-3} - 10^{-1}$	10	900
ADL	Perlite	9	44	N/A	0.049	20.4	2×10^{-1}	11	N/A
ADL	Glass Fiber	16	43	N/A	0.014	71.4	1.51×10^{-1}	N/A	N/A
ADL	Santocel	10	47	N/A	0.039	25.6	6×10^{-3}	N/A	N/A

N/A = not available

TABLE 2.6 SUMMARY DATA PROPERTIES OF EVACUATED THERMAL INSULATION WITH A COMPRESSIVE LOAD OF 15 PSI

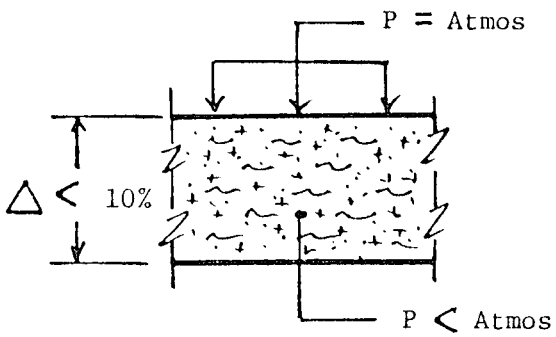
Specification No. VI

1. Insulation Concept

1.1 Concept VI

1.2 Description: Evacuated powder or granular materials encapsulated in a gas-impermeable and leak-free membrane.

1.3 Figure:



2. Insulation

2.1	Material	Perlite
2.2	Structure	powder
2.3	Closed/Open Cell	Open
2.4	Bulk density (in service)	9 lb/ft^3 (Table 2.6)
2.5	Temperature limit (°F)	>600
2.6	Deflection at 15 psi comp.	11 percent
2.7	Cell or interstitial gas	Air
2.8	Gas pressure	10^{-1} mm Hg
2.9	Dominant Failure mode	leaks, outgassing, diffusion
2.10	Material ($/lb)	0.053
2.11	Estimated R-Value (R/in)	20

3. Encapsulation

3.1	Class:	2 (steel)
3.2	Materials cost ($/$ft^2$):	0.21
3.3	Temperature Limitations (°F):	>600

CONCEPT VII

This concept is for evacuated fibrous materials that are contained in a gas impermeable and leak free membrane. Evacuation reduces gas conduction in the matrix. However, this is counteracted by the increase in conduction through the fibrous material that results from the compression induced at the boundaries due to the exterior atmospheric pressure and the interior subatmospheric pressure. As considered in Concept VI, the net effect of these two heat transfer modes is difficult to predict and must be determined from measurements. These measurements have been made for only a few materials.

Fibrous materials generally undergo large deflections when compressed with one atmosphere distributed load. For example, a fiberglass thermal insulation with an initial bulk density of 0.7 lbs/ft^3 was deflected to almost 90 percent of its normal thickness under a compressive load of about 3 psi. This deflection is not the same for all fibrous materials, but is determined by the fiber orientation and starting bulk density. With a low starting density and randomly orientated fibers (two directions), the deflection is large. The deflection decreases as starting density increases and as fibers are oriented in one direction only (load) applied perpendicular to the normal fiber axis).

These large deflections affect the use of the product. For example, a failure in the panel vacuum will cause the boundaries to bulge in response to the action of the spring force in the insulation material present on the internal encapsulated surfaces when the gas pressure is equalized. This bulging can induce additional leaks, permanent distortions in the appliance boundaries and affect the removal of the appliance from its installed location. It is desirable, therefore, that the fibrous materials used in implementing this concept have small deflections under atmospheric load pressures.

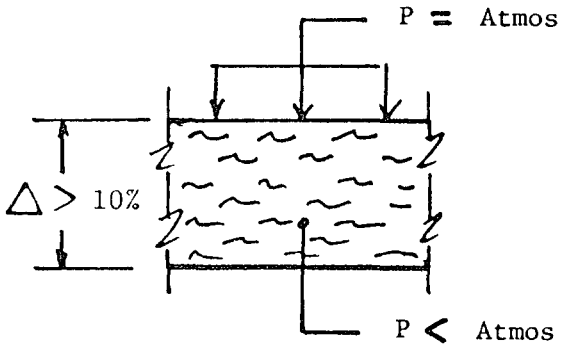
Specification No. VII

1. Insulation Concept

1.1 Concept VII

1.2 Description: Evacuated fibrous materials encapsulated in a gas-impermeable and leak-free membrane.

1.3 Figure:



2. Insulation

2.1	Material	Fiberfrax 660 paper
2.2	Structure	Fiber
2.3	Closed/Open Cell	Open
2.4	Bulk density (in service)	12
2.5	Temperature limit (°F)	>600
2.6	Deflection	30% @ 40 psi
2.7	Cell or interstitial gas	air
2.8	Gas pressure	10^{-4} mm Hg
2.9	Dominant Failure mode	Gas leakage and diffusion
2.10	Material ($/lb)	8
2.11	Estimated R-Value (R/in)	20

3. Encapsulation

3.1	Class:	3 (steel)
3.2	Materials cost ($/ft^2):	0.21
3.3	Temperature Limitations (°F):	>600

CONCEPT VIII

This concept makes use of evacuated multilayer thermal insulations developed and widely used for terrestial cryogenic and spacecraft applications. This class of insulations attains its best performance at evacuation pressure below 10^{-5} mm Hg. In the applications mentioned, resistivities of 1000-5000 hr ft^2°F/Btu-in have been achieved in practice with uncompressed material. However, the applications under consideration require that this insulation be compressed because of geometry and other considerations. This compression (one atmosphere) causes the insulation performance to decrease. Measurements made at these conditions indicate that certain multilayer insulations perform significantly better than all other types of insulations even though their performance is degraded by compression. Thus, they offer large improvements in thermal performance for the present applications. On the other hand, some of their properties will limit their use in appliances. The most important of these are discussed below.

The deflection of multilayer insulations under a compressive load is large and could lead to problems in replacing failed panels. The alternate layers of shields and spacers constituting a multi-layer insulation results in the formation of a loose, low density blanket of 2 to 6 lbs/ft^3. When placed into a flat panel and evacuated, the atmospheric pressure at the encapsulation surfaces compresses the insulation by a factor of 2 to 10 times. This deflection could cause serious deformation of an appliance made from evacuated multi-layer insulation panels if the vacuum fails. A single puncture of the panel would cause the internal and external pressure forces to equalize. The unbalanced spring force of the compressed insulation would expand the panels from 2 inches to 10 or more inches, distorting the appliance and possibly wedging it in place, causing damage to the surroundings and making its removal difficult and costly.

The shields of the multi-layer insulation in a mechanically compressed panel will impede the evacuation process. Unless the panels are manufactured in a vacuum chamber, their assembly will require mechanical presses to hold the encapsulation in place around the multi-layer insulation while it is being closed at the edges, evacuated and sealed. Under such high compression, the spacer materials, generally netting fabrics and open frames placed between the shields, will produce a large number of closed cells which would require days or possibly months to evacuate to the required pressure level of less than 10^{-5} mm Hg. This lengthy evacuation process will seriously impact the rates and costs of production.

Assuming that it is practical to fabricate evacuated multi-layer insulation panels, their performance would be significantly reduced by outgassing from the system components and leaks in the encapsulation. As previously noted, multi-layer insulations require a pressure less than 10^{-5} mm Hg. The maximum tolerable leakage rate into the panel for a 10 year life would be about 10^{-12} standard c^3/s compared to practical leak detection capabilities of 10^{-7} to 10^{-8} standard c^3/s. This gap between requirement and practical capabilities can be overcome by using a gettering material which is added to the panel during manufacture. Enough gettering material must be provided to absorb the outgassing products and to "pump" the pressure to a low level.

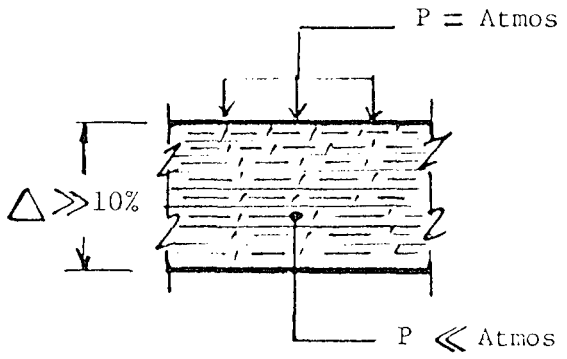
Specification No. VIII

1. Insulation Concept

1.1 Concept VIII

1.2 Description: Evacuated multi-layer radiation shields and spacers encapsulated in a gas-impermeable and leak-free membrane.

1.3 Figure:



2. Insulation

2.1	Material	Aluminized mylar and silk net
2.2	Structure	Plastic film and net spacer layers
2.3	Closed/Open Cell	Open
2.4	Bulk density (in service)	16 lb/ft^3
2.5	Temperature limit (°F)	250 (mylar), 200 (silk)
2.6	Deflection at 15 psi comp.	>>10%
2.7	Cell or interstitial gas	Air
2.8	Gas pressure	10^{-5} mm Hg
2.9	Dominant Failure mode	Leak or gas diffusion in encapsulation
2.10	Material ($/lb)	4.25
2.11	Estimated R-Value (R/in)	556 (Table 2.1)

3. Encapsulation

3.1	Class:	3 (Steel)
3.2	Materials cost ($/$ft^2$):	0.21
3.3	Temperature limitations (°F):	>600

NOTE: Concept Evaluated as Composite See Concept XI

CONCEPT IX Hollow Glass Spheres

This concept employs hollow spheres contained within an evacuated gas-impermeable and leak free encapsulation material. Hollow spheres are produced in a variety of materials and sizes. They occur naturally in steam boiler fly ash and are produced artificially from glasses. They are also manufactured commercially from ceramics, carbon, epoxy, phenolic polystyrene and other materials. Their sizes range, generally, from 10 to 150 microns in diameter. These materials are used primarily as fillers for reinforced plastics.

Artificially produced hollow glass spheres have been found to be useful insulation materials from cryogenic temperatures to 1000°F when placed in an evacuated space. Typically, they range in size from 45 to 125 microns, have walls about 1-2 microns in thickness and are filled with a partial atmosphere of SO_2 and O_2. Vapor deposited aluminum coatings can be applied to the sphere exterior to reduce radiative heat transfer. The low pressure of the gas within the spheres combines with the low pressure within the evacuated interstitial spaces to significantly reduce gas conduction effects. The spheres will tolerate large compressive loads without crushing while deflecting less than 10 percent. The resulting insulation has a relatively low density and heat capacity.

The limited values available in the literature indicate the technical feasibility of the concept at a thermal resistivity of 20 hr ft^2°F/Btu·in. Using hollow glass spheres at a compressive load of one atmosphere in an evacuated space at a pressure of 0.1 microns Parmley* has measured thermal resistivities from 15 to 28 hr ft^2°F/Btu·in at a mean temperature of -117°F. At similar conditons, Cunnington** reports a value of 31 hr ft^2/Btu·in for a 50 percent mixture of aluminized spheres and uncoated spheres. Both reported measurements were made at conditions simulating the requirements for cryogenic vessels. Thus, the cold and warm boundaries of the insulation were respectively maintained at -321°F and 80°F. The thermal resistivity values of the tested materials can be expected to be smaller at the higher mean temperatures experienced in appliance applications.

The ability of the hollow glass spheres to withstand the compressive load induced by the atmospheric pressure is critical to this concept. The spheres have a crush resistance of 150 to 1500 psi under hydrostatic pressure depending on the specific product. Parmley* has measured the net increase

* Parmley, R.T., and J.R. Barber, Evacuated Load-Bearing High Performance Insulation Study, NASA Contract NAS3-17817, Aug. 1977.

** Cunnington, G.R., C.L. Lien Heat Transfer in Microsphere Insulation Advances in Cryogenic Engineering, Vol. 18, Plenum Press, New York 1973.

in cracked or broken spheres as a function of compressive load for a specific material. The results show that at a load of one atmosphere, no breakage could be measured while at a load of 1.5 atmospheres, 5 percent of the spheres cracked or broke. The required crush resistance can be obtained by varying the sphere size and wall thickness in selecting the appropriate grade of glass sphere materials.

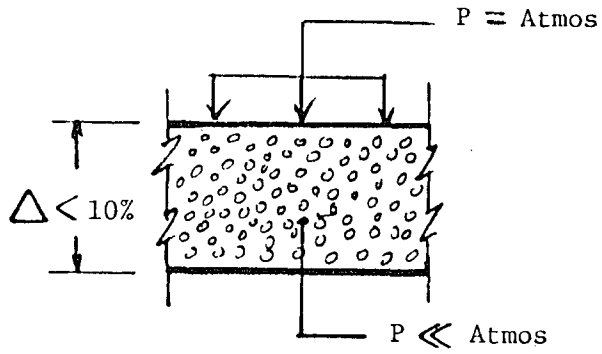
Specification No. IX

1. Insulation Concept

1.1 Concept IX

1.2 Description: Hollow glass spheres with evacuated interstitial areas encapsulated in a gas-impermeable and leak-free membrane.

1.3 Figure:



2. Insulation

2.1	Material	Hollow glass microspheres (uncoated)
2.2	Structure	Compacted glass spheres
2.3	Closed/Open Cell	Open
2.4	Bulk density (in service)	6.5 lb/ft^3
2.5	Temperature limit (°F)	> 600
2.6	Deflection at 15 psi comp.	< 10%
2.7	Cell or interstitial gas	Air
2.8	Gas pressure	10^{-6} mm Hg*
2.9	Dominant Failure mode	Gas leakage or diffusion
2.10	Material ($/lb)	0.85
2.11	Estimated R-Value (R/in)	15

3. Encapsulation

3.1	Class:	3 (steel)
3.2	Materials cost ($/$ft^2$):	0.21
3.3	Temperature Limitations (°F):	> 600

* Data for this material is available at this pressure only. The R-value of this material should remain unchanged to a pressure of 10^{-3} mm Hg.

CONCEPT X

This is a concept of an evacuated composite thermal insulating panel. This concept employs an evacuated composite of rigid foam and powder as the insulation materials. The rigid foam supports atmospheric pressure forces at the panel boundaries freeing the powder from compressive forces. The powder is evacuated to reduce its thermal conductance to very low values, compensating for the higher thermal conductance of the foam, so that in combination the two materials produce a resistivity greater than 20 hr $ft^2 °F/Btu \cdot in$.

Either closed cell or open celled foam can be used for this application. Rigid closed cell insulating foams are presently available in a number of varieties of which polyurethane, polystyrene and glass are typical. Rigid open cell foams are not commercially available but may become available in the near future.* Of the two varieties, an open cell foam would be the most desirable for the present concept.

The closed cell foams will contribute to the outgassing load within the panel. The foaming gases are sealed in the closed cells of the foam and are prevented from direct communication with the vacuum spaces (except for cell ruptures that can occur through use and movement of the appliance). However, diffusion of the gases through the walls of the cells and into the space occupied by the powder will cause the pressure of the evacuated space to slowly rise with time. Thus, the outgassing of foam materials must be evaluated and factored into the panel design.

The foam can be configurated in two ways to support the panel boundaries. First, the foam can be molded into a perforated board with the powder located within the perforations. Second, the foam can be formed into cylinders that are positioned by attaching them to a net type material with the powder filling the spaces between the foam supports.

The panel with cylinder supports is easier to evacuate during fabrication than the panel with perforated boards. In the former, the space containing the evacuated powder is continuous. The spaces in the perforated board are not naturally continuous and must be artifically connected to allow the powder filled spaces in the panel to be evacuated during fabrication. The interconnecting passages can be molded, pressed, cut, etc. in the perforated board during its manufacture.

The fractional area of the panel which must be occupied by the foam is dependent on the foams compressive strength, its spring rate and its thermal conductivity. Generally, for a specific foam, these properties are related to foam density, i.e., compressive strength and thermal conductivity increase and load deflection decreases as density increases. The foam area must be large enough to prevent compressive failure of the foam when the panel is evacuated and to minimize the thickness change. On the other hand, the foam area must be as small as possible because its high conductance contributes to the overall panel conductance. These factors must be optimized in the design.

* See Concept IV

The panel surfaces may have a noticeable dimpled appearance due to the localized support provided by the rigid foam. The size and depth of the dimples will depend upon the thicknesses and materials of the encapsulation, as well as the size and arrangement of foam plugs and perforations. These factors must be considered in the design to assure that structurally stable panels of suitable appearance can be made.

An example of this system used for cost estimates was based on polyurethane foam making up about 30 percent of the insulation volume with the remainder filled with perlite powder. The composite thermal resistivity would be about 20 hr ft^2°F/Btu·in. An increased thermal resisitivity can be achieved by reducing the foam fraction.

Important factors in the application of this concept are the operating temperature levels of the appliance and the temperature limitations of foam materials. The temperature dependent compressive strength and deflection under load of polymeric materials limits their use to refrigeration and freezer applications. Available glass, ceramic and phenolic foams currently under development would be suitable for water heater applications. The higher operating temperatures required in ovens further limits the selection to glass and ceramic foams.

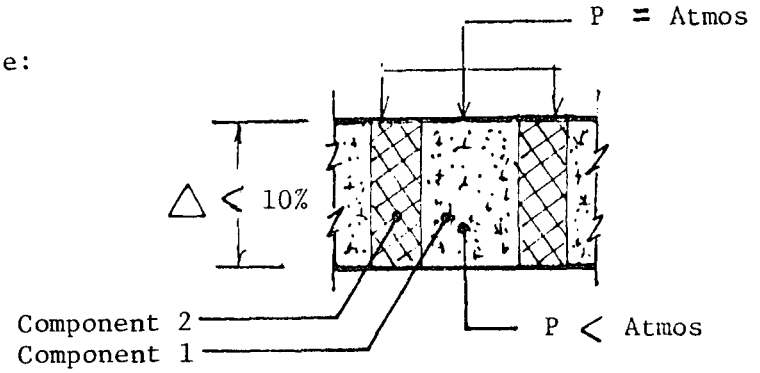
Specification No. X

1. Insulation Concept

1.1 Concept X

1.2 Description: Evacuated composite of rigid plastic foam and powders encapsulated in a gas-impermeable and leak-free membrane. The area of the foam in the direction of heat flow is one third of the total. Composite R/in = 20

1.3 Figure:



2. Insulation

	Component 1	Component 2
2.1 Material	Perlite	Polyurethane
2.2 Structure	Powder	Foam
2.3 Closed/Open Cell	Open	Closed cell
2.4 Bulk density (in service)	9 lb/ft^3	2 lb/ft^3
2.5 Temperature limit (°F)	>600	150-190
2.6 Deflection at 15 psi comp.	Not compressed	< 10% *
2.7 Cell or interstitial gas	Air	R-11
2.8 Gas pressure	10^{-1} mm Hg	∿ 1 atmos
2.9 Dominant Failure mode	Gas leakage, diff.	Structural, compressive
2.10 Material ($/lb)	.053	0.70
2.11 Estimated R-Value (R/in)	69	8.
	>20 Composite	

3. Encapsulation

3.1 Class:	
3.2 Materials cost ($/$ft^2$):	MAM,** 0.15; Steel, 0.21
3.3 Temperature Limitations (°F):	MAM, 300; Steel >600

* Foam must have a minimum compressive strength between 50-75 psi.
** MAM is a Mylar/aluminum/Mylar composite.

CONCEPT XI Sandwich of Multilayer Insulation and Plastic Foam

This concept uses a composite of multilayer insulation and plastic foam. The multilayer is encapsulated in a gas-impermeable and leak-free closed cell foam. The latter material can be either board form or foamed in place around the encapsulated multilayer. The thickness of the multilayer is determined by the desired panel thermal restivity.

The multilayer insulation has the same properties as the material described for Concept VIII. Smaller amounts of this material are used in this concept because of its high thermal resistivity. The rigid closed cell plastic foam placed around the multilayer insulation protects the encapsulation against mechanical damage. Alternatively, thinner walled appliances are possible through elimination of the foam material.

Refer to the discussion of Concept VIII for the limitations in the use of multilayer insulations.

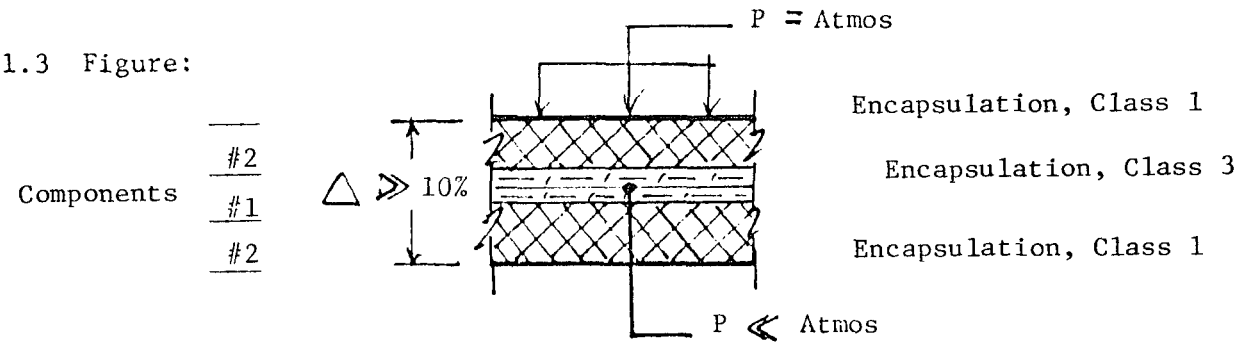
Specification No. XI

1. Insulation Concept

1.1 Concept XI

1.2 Description: Evacuated multi-layer radiation shields and spacers encapsulated in a gas-impermeable and leak-free membrane sandwiched between rigid closed cell plastic foam panels. Component 1 occupies 5% of the space and component 2 the remainder. Composite R/in = 20.

1.3 Figure:



2. Insulation

	Component 1	Component 2
2.1 Material	Al/Mylar & Silk	Polyurethane
2.2 Structure	Layered	Foam
2.3 Closed/Open Cell	Open	Closed cell
2.4 Bulk density (in service)	16 lb/ft^3	2 lb/ft^3
2.5 Temperature limit (°F)	200	150-190
2.6 Deflection at 15 psi comp.	>> 10%	< 10%
2.7 Cell or interstitial gas	Air	R-11
2.8 Gas pressure	10-5 mm Hg	∿ Atmospheric
2.9 Dominant Failure mode	Leaks & diff.	Air diffusion
2.10 Material ($/lb)	4.25	0.70
2.11 Estimated R-Value (R/in)	556	8
	> 20 Composite	

3. Encapsulation

3.1 Class:	3 (Steel)
3.2 Materials cost ($/ft^2):	0.21
3.3 Temperature Limitations (°F):	>600

CONCEPT XII

This concept is similar to Concept X, except for the materials used to support the encapsulation against atmospheric pressure. A low conductance honeycomb, instead of foam, provides the spaces into which the lower conductance evacuated powder can function free of the thermally degrading compressive forces. The appropriate selection of the honeycomb in this concept can result in thermal resistivities greater than 20.

A good candidate honeycomb is one produced from a composite of fiberglass and epoxy. The strength and thermal properties of this material are suited to the present application. Light weight structural panels used for high performance aircraft and for space craft are examples of the honeycomb technology. A further advantage of this material is its low thermal conductivity which derives from the low conductivity of both the glass and epoxy components. Low outgassing is another important useful characteristic of this class of materials which makes them suitable for evacuated panels.

The honeycomb must be perforated to permit evacuation of the panel. Generally, the honeycomb is bonded to the encapsulating face sheets sealing the cells thus formed. It is common practice to perforate the honeycomb to permit pressure equalization between the interior and exterior spaces through the panel edges. The evacuated panel, being sealed on the edges as well as the surfaces, would make use of the perforations to connect all the cells which can then be evacuated through a common pumpout connection which would be subsequently sealed.

The honeycomb must be kept to a small fraction of the composite. The effective thermal conductivity of the composite is very sensitive to the conductivity of the honeycomb because it acts in parallel with the powder and has a much larger thermal conductivity than the powder. If Nema G10 Fiber glass Epoxy honeycomb (resistivity of 3.3) and perlite are used, the honeycomb cross-sectional area must be only about 6 percent of the total in order to achieve a composite thermal resistivity of 20.

A honeycomb forming 6 percent of the area must have a minimum compressive strength of 250 psi to balance the external pressure forces. This is a fraction of the compressive strength of an appropriately designed fiberglass/epoxy composite which can achieve compressive strength at least 100 times this value. Thus, the large strength of this class of materials makes possible panels of larger R/in. values because the quantity of fiberglass/epoxy can be reduced.

The amount of honeycomb material, material strength and cell size must also be optimized with the constraints imposed by the encapsulation. For example, thin cover sheets having a low Young's modulus (plastics and lamination of aluminum and Mylar) will produce dimples that are more pronounced than those formed in thick cover sheets having a high Young's modulus (steel) for a given cell size. The dimpled surface will affect the appearance of the panels, as well as the reliability of the

encapsulation and the amount of compression introduced into the powder. The latter must be minimal because the thermal conductance of the powder is degraded directly by this compression. The honeycomb will introduce smaller compressive loads in the powder than the polymeric foams mentioned in Concept X, because of its higher Young's modulus.

The application of this concept is limited to refrigerators, freezers and hot water heaters. This is due mainly to the available reinforced plastic honeycomb materials which generally have a maximum continuous heat resistance temperature of 400°F. A further limitation of these materials is the temperature related outgassing rate which is an important consideration in the present application. The plastic and plastic-metal laminates that may be used as encapsulation materials have similar limitations.

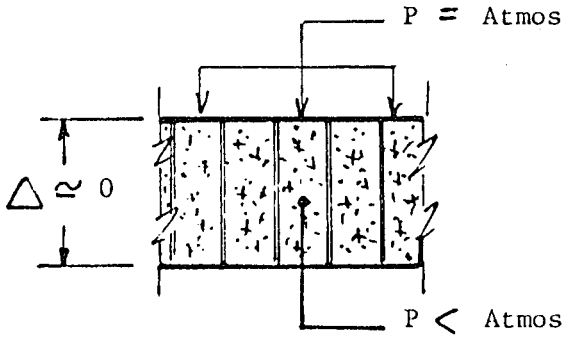
Specification No. XII

1. Insulation Concept

1.1 **Concept:** XII

1.2 **Description:** Evacuated honeycomb filled with powder or granular materials and a low conductance gas and encapsulated in a gas-impermeable and leak-free membrane. (1% of the space is filled with component 2 (NEMA-G10 Fiberglass Epoxy.) R/in.-value ∿20.

1.3 Figure:



2. Insulation

	Component 1	Component 2
2.1 Material	Perlite	NEMA-G10 Fiberglas/Epoxy
2.2 Structure	Powder	Solid
2.3 Closed/Open Cell	Open	N/A
2.4 Bulk density (in service)	9 lb/ft^3	116
2.5 Temperature limit (°F)	>600	400-500*
2.6 Deflection at 15 psi comp.	Not Compressed	<< 10%
2.7 Cell or interstitial gas	Air	N/A
2.8 Gas pressure	10^{-1} mm Hg	N/A
2.9 Dominant Failure mode	Leak & Diffu.	Structural, Compressive
2.10 Material ($/lb)	.053	3.00
2.11 Estimated R-Value (R/in)	69	0.3
	> 20 Composite	

3. Encapsulation

3.1 Class:	3
3.2 Materials cost ($/ft^2):	0.21 (steel)
3.3 Temperature Limitations (°F):	>600

*This is the zero load temperature limits. Load temperature limits are below this value.

CONCEPTS LINDE A, B, C, AND D

Linde Corporation submitted data (shown in Table 2.7) on four insulations they have developed. Details beyond those shown in the table were not submitted because they were considered proprietary by Linde. In this project, the Linde thermal resistivities and costs were used to estimate the added cost, energy savings and years-to-payback from substituting them for the existing insulation for the appliances indicated. In all cases, we combined them in a composite similar to concept XI using foam as the second material for lower temperature applications (refrigerators, freezers, and water heaters) and fiberglass for higher temperature applications (ovens).

2.7 Cost Estimates

As part of this project estimates were made of the added costs to manufacture appliances using advanced insulation concepts. This was done by estimating the cost differential of the materials required and the cost to fabricate appliances utilizing the new insulation systems.

Estimates of the materials costs were made by calculating the volume and cost of the insulation in the initial appliance and the cost of the new insulation system having the same volume. The difference of these two values gives the added materials cost. The cost used for materials are summarized in Table 2.8.

In order to estimate the added manufacturing cost, representative manufacturing lines for different configurations were designed conceptually. In each case the following basic steps were carried out:

- cut raw material to size;
- assemble elements;
- evacuate;
- store for a period of time depending on the level of vacuum required;
- inspect for leaks by sampling gas content;
- rework rejected units (10% rework requirement was assumed);
- paint final product.

The evacuation was done through a valve welded into the assembly during the initial assembly process. It was assumed that the tube to which the valves were mounted would be crimped and welded following the final inspection, and the valves recycled for use on other units. The following assumptions were made in the analysis:

INSULATION	POSSIBLE APPLICATIONS	R/in $\left(\frac{hr\ ft^2 °F}{Btu \cdot in}\right)$	PRESSURE RANGE (mm Hg)	DENSITY (LB/FT^3)	TEMPERATURE APPLICATION RANGE (°F)	PERCENT COMPRESSION FROM 0 to 15 PSI LOAD	MATERIAL COST ($/BOARDFOOT)
LINDE-A	Refrigerators Freezers Water Heaters	33	10^{-3} - 10^{-1}	2	Cryogenic to 150	1.5	.50
LINDE-B	Water Heaters	47.6	10^{-3} - 10^{-1}	12	Cryogenic to 450	10	2.00
LINDE-C	Ovens	72.5	10^{-3} - 10^{-1}	20	Cryogenic to 900	10	1.00 - 2.50
LINDE-D	Cylindrical Applications Non-Load Bearing Water Heaters	167	10^{-3} - 5×10^{-3}	1.5	Cyrogenic to 900	--	.40

TABLE 2.7 LINDE VACUUM INSULATION PROPERTIES

	Material	Material Density (lb/ft^3)	Cost ($/lb)	Cost Used in Calculation
1.	Cab-O-Sil/Monarch Carbon, 50-50 Mixture	9.5	1.80	17 ($/ft^3)*
2.	Polyurethane Foam	2.0	0.70	1.40
3.	MIN K-1400	20.0	46	920
4.	Fiberfrax Paper	10-12	8	96
5.	Hollow Glass Microspheres	6.5	0.85	5.50
6.	NEMA G-10 (For Honeycomb)	116	2-4	348
7.	Fluorocarbon 11			0.15
8.	Xenon Gas			0.75
9.	Phenolic Foam	2.75	1.0	2.75
10.	Perlite	9.0	0.053	0.48
11.	Aluminum/Mylar & Silk (Multilayer)	16	4.25	68
12.	Gettering Material			1
13.	Fiberflass Insulation			
	Water Heater	1		0.75
		2		1.04
	Ovens	1.09		0.55
		1.45		0.73
14.	Aluminum Foil			0.06 ($/ft^2)**
15.	Aluminum/Mlyar Laminate			0.15
16.	Sheet Steel (25 Gage)			0.21

* per cubic foot of insulation
** per square foot of panel surface

TABLE 2.8 SUMMARY OF MATERIAL COSTS

- 1,000 appliances/day were produced. This would involve producing either six flat panels/appliance, or one assembly depending on the appliance type;
- the facility operated 250 days/year @ 1 shift/day;
- the process operated at 85% productivity;
- a standard panel size of 2 ft x 6 ft x 2 in thick was used.

Figures 2.5 through 2.8 show schematics of the types of manufacturing processes evaluated. The first figure shows the steps utilized in manufacturing a typical flat panel in which several elements, including fillers and supports against atmospheric pressure were utilized. The second figure shows the operation for a unit assembled as a single package, such as a water heater in which a round container is used. The third figure shows the procedure for fabricating a multi-layer type insulation which would be encapsulated in a film and then foamed in place in a panel. The final figure shows the details of the evacuation and finishing operations, which would be similar for the various concepts.

The analysis of these different fabrication processes were carried out in detail for a number of different configurations. When the final results were compared, the manufacturing costs for the different configurations were close enough so that the generalization shown in Table 2.9 could be made. This table shows the assumptions related to the manufacturing facility and the estimated cost/unit required to manufacture the various configurations. For the cost analysis, a value of $12.00/appliance was used for the manufacturing costs. Approximately 50% of this total is direct and indirect labor, 10% is depreciation on plant, equipment and tooling, and the balance is costs such as fringe benefits, administration, maintenance, insurance and taxes.

The cost of the materials for each insulation type was added to this assembly cost and the total marked up by a factor of 3 to estimate the added costs to the end user of the appliance. This mark-up takes into account typical manufacturer, distributor, and dealer margins.

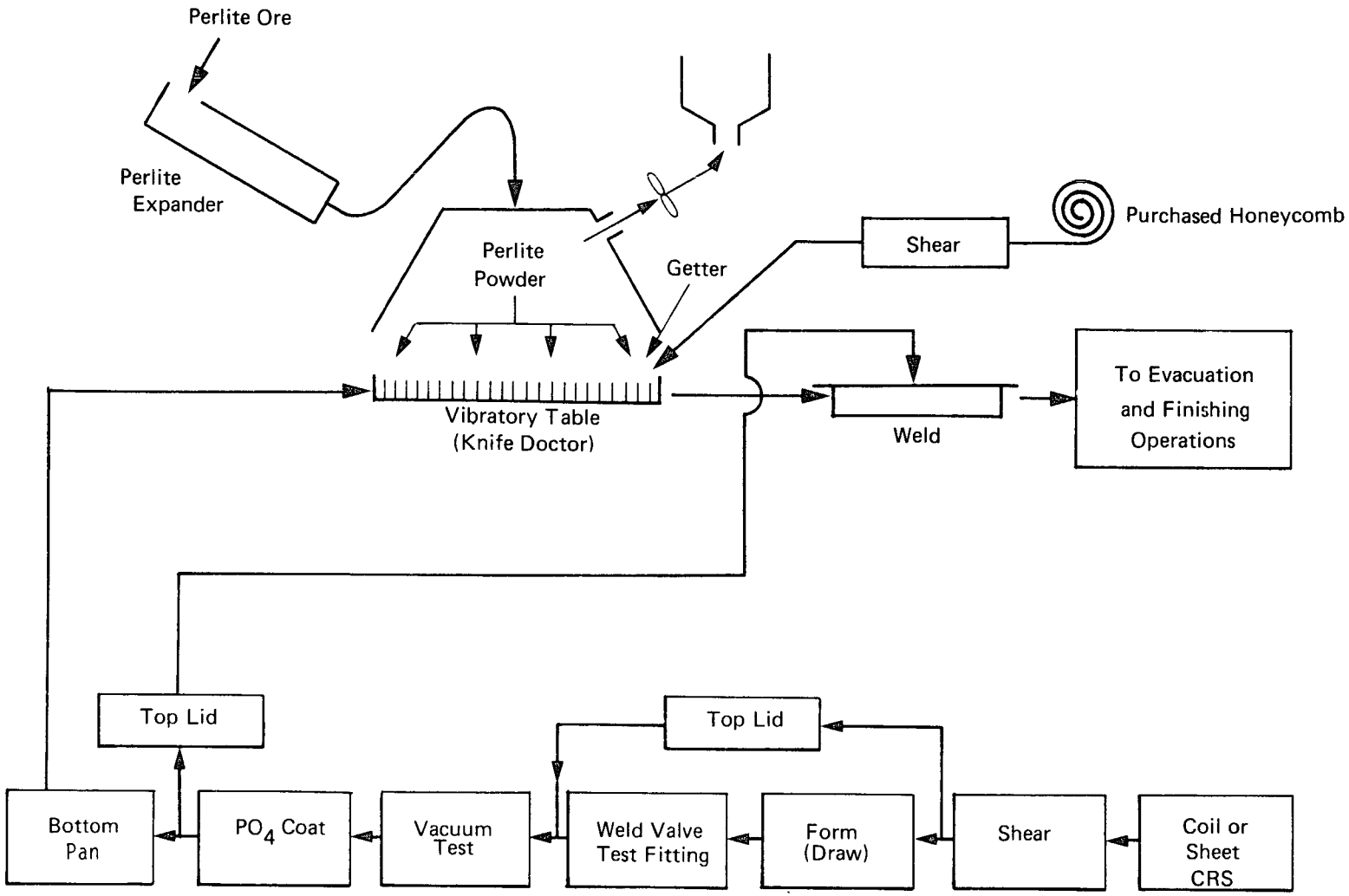


Figure 2.5 MANUFACTURING SCHEMATIC FOR CONCEPTS IV, VI, X, and XII (FLAT PANEL)

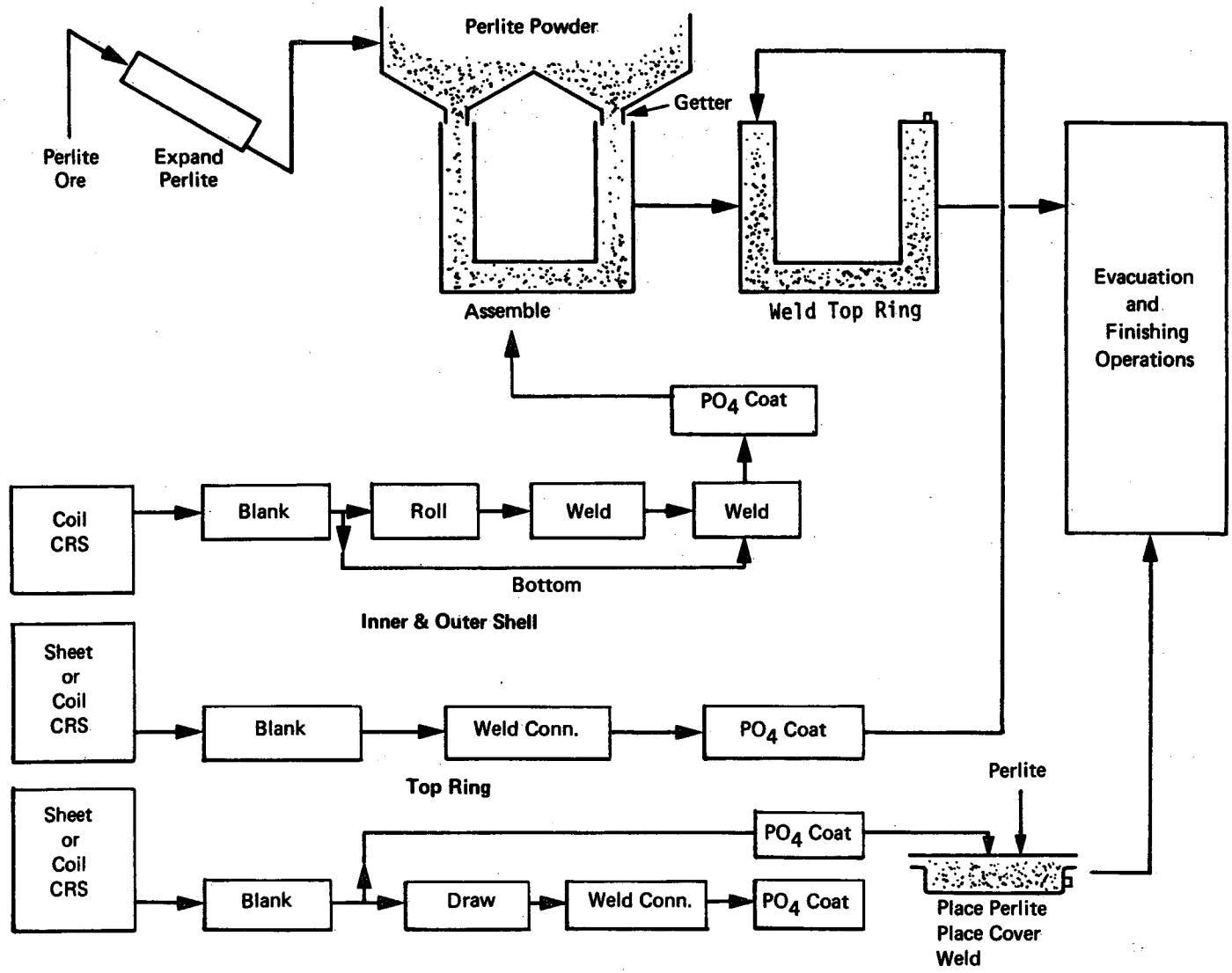


Figure 2.6 MANUFACTURING SCHEMATIC FOR CONCEPT VI (ROUND)

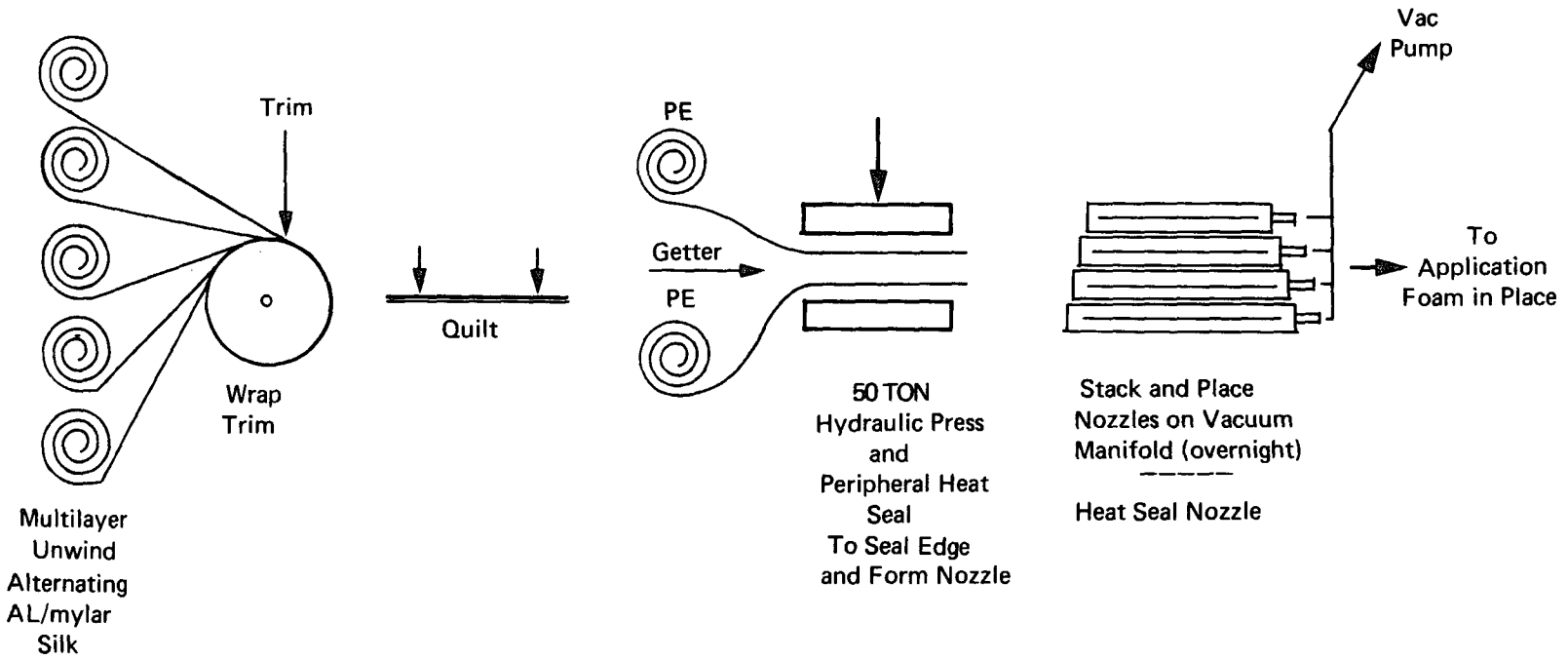


Figure 2.7 MANUFACTURING SCHEMATIC FOR CONCEPT XI (MULTILAYER)

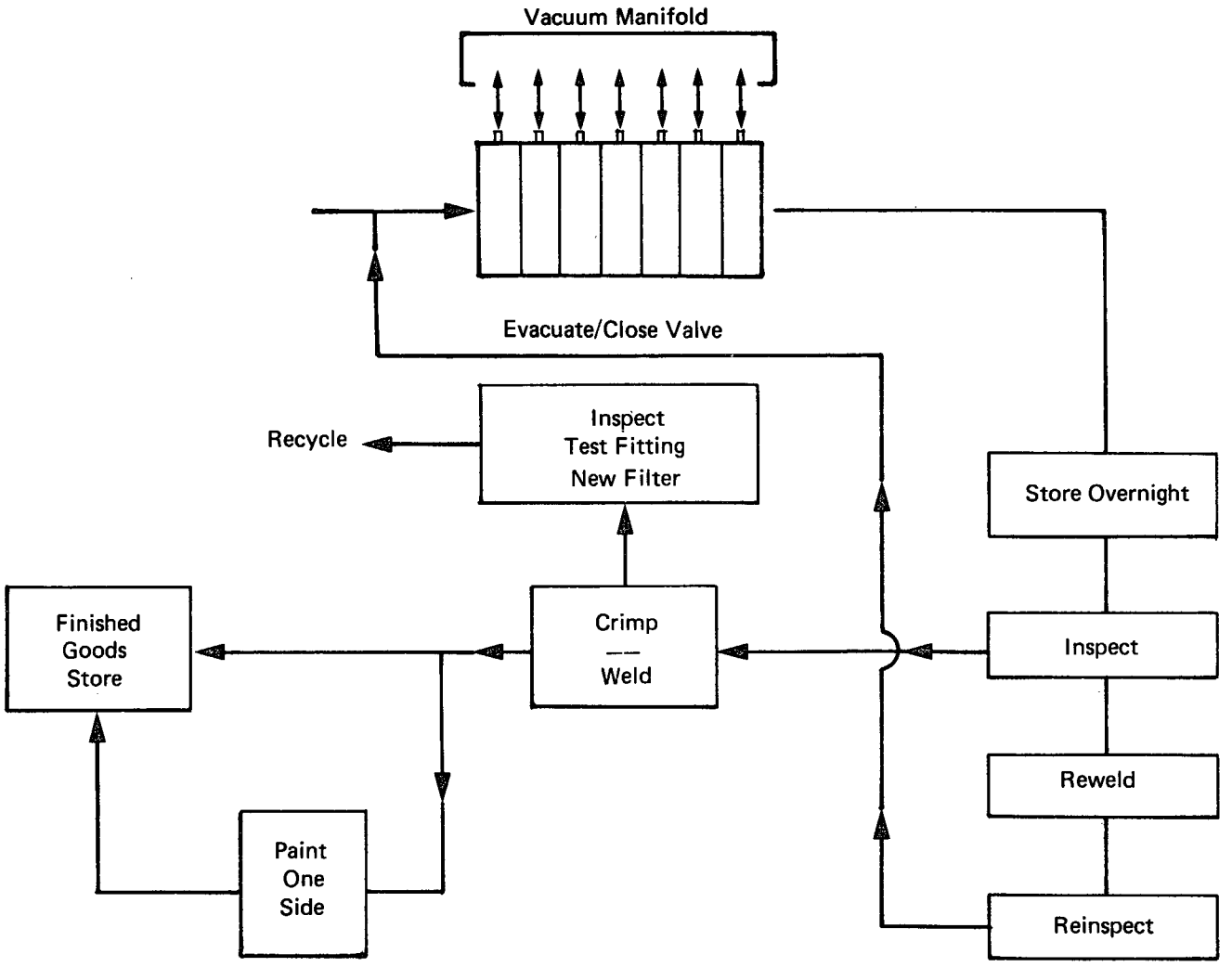


Figure 2.8 EVACUATION AND FINISHING OPERATIONS

	FLAT PANELS		UNIT	
	Range	Typical Value	Range	Typical Value
Quantity/day	-	6,000	-	1,000
Quantity/year	-	1,500,000	-	250,000
Plant Size (ft^2)[1]	70,000 - 92,000	80,000	70,000 - 92,000	80,000
Number of Hourly People[2]	77 - 101	85	60 - 84	70
Equipment Costs[3] (1000 $)	1920 - 3085	2300	1900 - 2550	2200
Cost/Unit ($)	1.73 - 2.28	2.00	8.65 - 11.45	10.00

Notes: 1. Costed at $35/ft^2$; amortized over 30 years.

2. Wage of $8/hr assumed.

3. Amortized over 10 years.

TABLE 2.9 BASIS FOR ESTIMATED MANUFACTURING COSTS

CHAPTER 3

ANALYSIS OF ENERGY SAVINGS FROM USE OF IMPROVED INSULATION

3.1 General Approach

In order to estimate the energy savings potential of using improved insulation, an engineering analysis was performed for each appliance type to partition the flow of energy through the appliance. First, a baseline unit representative of the most widely used model of each appliance type analyzed was selected. Next, the typical useful output, and total losses were predicted using laboratory test data, data from the literature, and computer models. The losses were further partioned to estimate the portion which passes directly through the insulation. Finally, the reduction in the losses due to the insulation were estimated, assuming arbitrary reductions in the insulation thermal conductivity while maintaining the same thickness as originally used. In some cases, the reduction in losses depended directly on insulation improvement. In other cases, changes in the performance of other components (e.g., compressor in refrigerators) had to be considered as well.

The energy savings for various levels of insulation improvement are summarized in Tables 3.1 and 3.2 for residential and commercial appliances respectively. For residential applications, the largest savings potential comes from improving water heaters. Both gas-fired and electric units offer significant dollar savings potential. Refrigerators and freezers also appear to be reasonable candidates while the savings for ovens is quite small.

For commercial applications, by contrast, the potential savings for ovens are substantial. This occurs because competitive pressures have forced manufacturers to make larger ovens within fixed outside dimensions. This results in thin insulation. Commercial ovens also are typically on for long periods each day. Commercial water heaters and freezers also have savings potentials large enough so that they appear to be good candidates for improved insulation systems.

The analyses used to derive these results are present in the remainder of Chapter 3.

Appliance	Baseline Energy Consumption (10^6 Btu/year)	Savings with Insulation Resistivity: R/in = 10 (10^6 Btu/year)	R/in = 10 $/year	R/in = 15 (10^6 Btu/year)	R/in = 15 $/year	R/in = 20 (10^6 Btu/year)	R/in = 20 $/year
Refrigerator	3.7	0.77	11.28	1.53	22.41	1.88	27.54
Freezer (Chest)	3.9	0.78	11.43	1.54	22.56	1.86	27.25
Water Heater							
Gas	30.7	2.43	9.23	2.86	10.87	3.07	11.67
Electric	20.1	1.52	22.27	1.66	24.32	1.75	25.64
Oven							
Gas	2.82	0.17	0.64	0.20	0.77	0.22	0.84
Electric	1.23	0.17	2.46	0.20	2.97	0.22	3.24

Note:
- All savings at point at use
- Electricity @ $0.05/Kwh
- Gas @ $3.80/$10^6$ Btu

TABLE 3.1 SAVINGS FROM USE OF IMPROVED INSULATION IN RESIDENTIAL APPLIANCES

Appliance	Baseline Losses through Insulation (10^6 Btu/year)	Savings with Insulation Resistivity: R/in = 10 (10^6 Btu/year)	R/in = 10 ($/year)	R/in = 15 (10^6 Btu/year)	R/in = 15 ($/year)	R/in = 20 (10^6 Btu/year)	R/in = 20 ($/year)
Freezer*	2.30	0.73	10.69	1.59	23.29	1.96	28.71
Water Heater - Gas	3.61	2.35	8.93	2.76	10.49	2.97	11.29
Electric	3.89	2.54	37.21	2.98	43.66	3.21	47.03
Oven** - Gas	4.2	2.6	9.87	3.04	11.56	3.30	12.54
Electric	4.2	2.6	38.07	3.04	44.56	3.30	48.35

* Per 3 feet of door section for display freezer

** Single free standing oven

Note:
- All savings at point of use
- Electricity @ $0.05/Kwh
- Gas @ $3.80/$10^6$ Btu

TABLE 3.2 SAVINGS FROM USE OF IMPROVED INSULATION IN COMMERCIAL APPLIANCES

3.2 Effect of Improved Insulation in Residential Ovens

The baseline residential oven selection was derived from SAI* data. This report shows the following energy partitioning for electric and gas ovens during the Department of Energy standard test.

	Electric		GAS	
	Btu/Cycle	%	Btu/Cycle	%
to load	457	13.6	457	5.9
to oven mass	1535	45.7	1535	19.9
thru walls	754	22.4	754	9.8
thru door glass	102	3.0	102	1.3
thru vents	512	15.2	4875	63.1
Total..	3360	100	7723	100

In both the gas and electric oven the insulation used was 3 inches of 1.45 lb/ft^3 fiberglass with a thermal conductivity of .26 Btu·in/hr ft^2°F at a mean temperature of 130°F.

$$\left[R = 11.5 \ \frac{hr\ ft^2 °F}{Btu} \quad \text{or } R/in = 3.8 \ \frac{hr\ ft^2 °F}{Btu\ in} \right]$$

This is equivalent to the best 1980 practice and is the normal insulation for self cleaning ovens.

SAI estimates annual energy use at 2.7 X 10^6 Btu for gas ovens without pilots and 344 Kwh (1.17 X 10^6Btu) for electric ovens. An additional 28 Kwh/year is used for self cleaning electric ovens. Assuming a 50% split between surface heating units and the ovens, this is in good agreement with MRI data** on electric ranges which measured total annual usage

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* Energy Efficiency Program for Kitchen Ranges and Ovens, DOE Contract #CR-04-60727-00, Science Application, Inc., November, 1977.

** Patterns of Energy Use by Electrical Appliances, Prepared by Midwest Research Institute, for the Department of Energy, Project #EC-75-C-01-8397.

of 782 Kwh. The value for gas fired ovens agrees with that of Hirst* (4.7×10^6 assuming a 50-50 split) if a pilot light is used and is consistent with values in the Department of Energy standard test procedure. The electric value is significantly lower than the Hirst value (680 Kwh/yr) but is consistent with Department of Energy standard test values.

Using the SAI estimates for energy savings from the use of improved insulation gives the results shown in Figure 3.1 and Table 3.3. The energy saving per unit with significant changes in insulation can be seen to be relatively small. This occurs because residential oven use is typically short term so that the heat transfer is primarily transient. The standard test requires 30-45 minutes during which time the oven mass acts to dampen losses through the insulation. In addition, some losses occur due to heat transfer through edges and, therefore, they are not limited by insulation. Finally, in considering the net energy savings from improved insulation, allowance should be made for the fact that some fraction of these losses shown are useful in displacing winter space heating requirements. This effect has not been included in this analysis. An earlier Arthur D. Little, Inc.,** study estimated that, on a national average, approximately 50% of these losses were actually useful.

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* Eric Hirst, Oak Ridge National Laboratory, Private Communications, 1979.

** Design, Development, and Demonstration of a Promising Integrated Appliance. Final Report, Phase I - Design, Arthur D. Little, Inc., C-79627, Sept. 1977

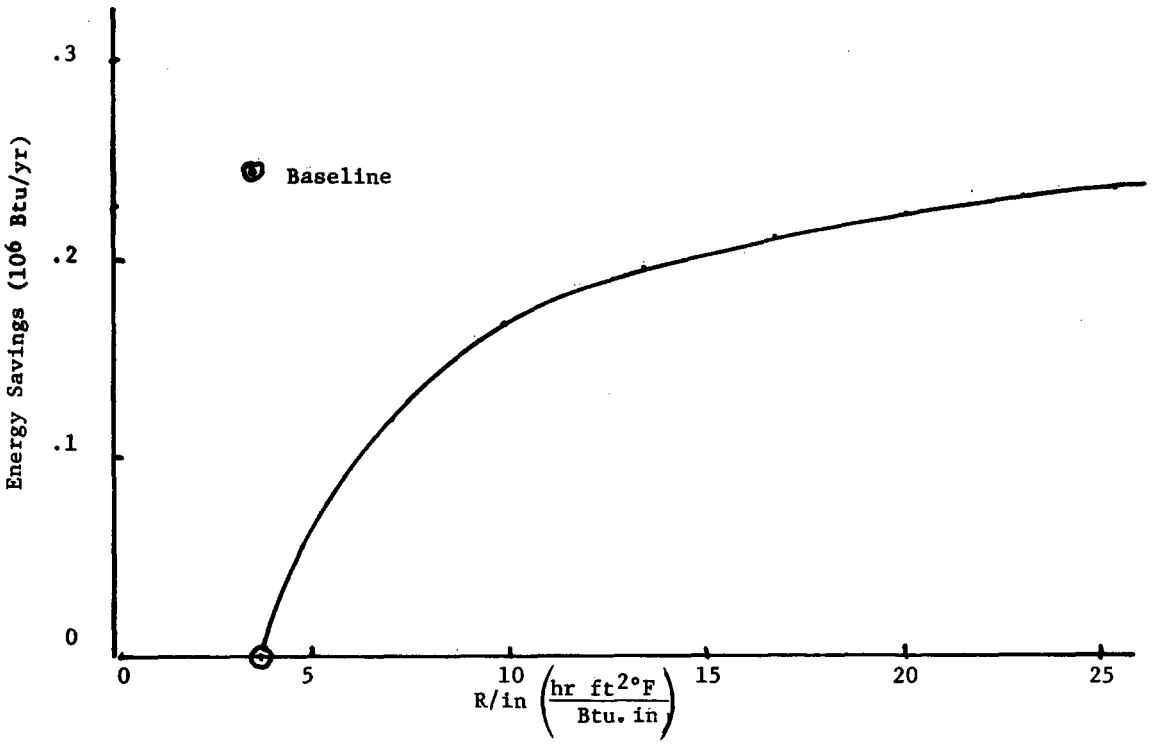


FIGURE 3.1 ENERGY SAVINGS FROM USE OF IMPROVED INSULATION IN RESIDENTIAL GAS AND ELECTRIC OVENS

R/in $\left(\frac{hr\ ft^2 °F}{Btu\ in}\right)$	Losses Through Insulation (Btu/day)	Savings		
		(10^6Btu/year)	Gas Oven* ($/Year)	Electric Oven** ($/Year)
3.8	754	-	-	-
5	580	.064	0.24	0.94
10	295	.168	0.64	2.46
15	198	.203	0.77	2.97
20	149	.221	0.84	3.24
25	120	.231	0.88	3.38

* Gas at $3.80/$10^6$ Btu
** Electricity at $14.65/$10^6$ Btu ($.05/Kwh)

TABLE 3.3 SAVINGS FROM USE OF IMPROVED INSULATION IN RESIDENTIAL GAS AND ELECTRIC OVENS

3.3 Effect of Improved Insulation in Residential Refrigerator-Freezers and Chest Freezers

Figure 3.2 and 3.3 taken from a recent Arthur D. Little study for the Department of Energy* shows the shipments of refrigerators and freezers by size and type. A 16 cubic foot refrigerator-freezer and a 23 cubic foot chest freezer were selected for this project to represent the majority of the marketplace, as these units fall in the middle of the sales statistics of the most popular models.

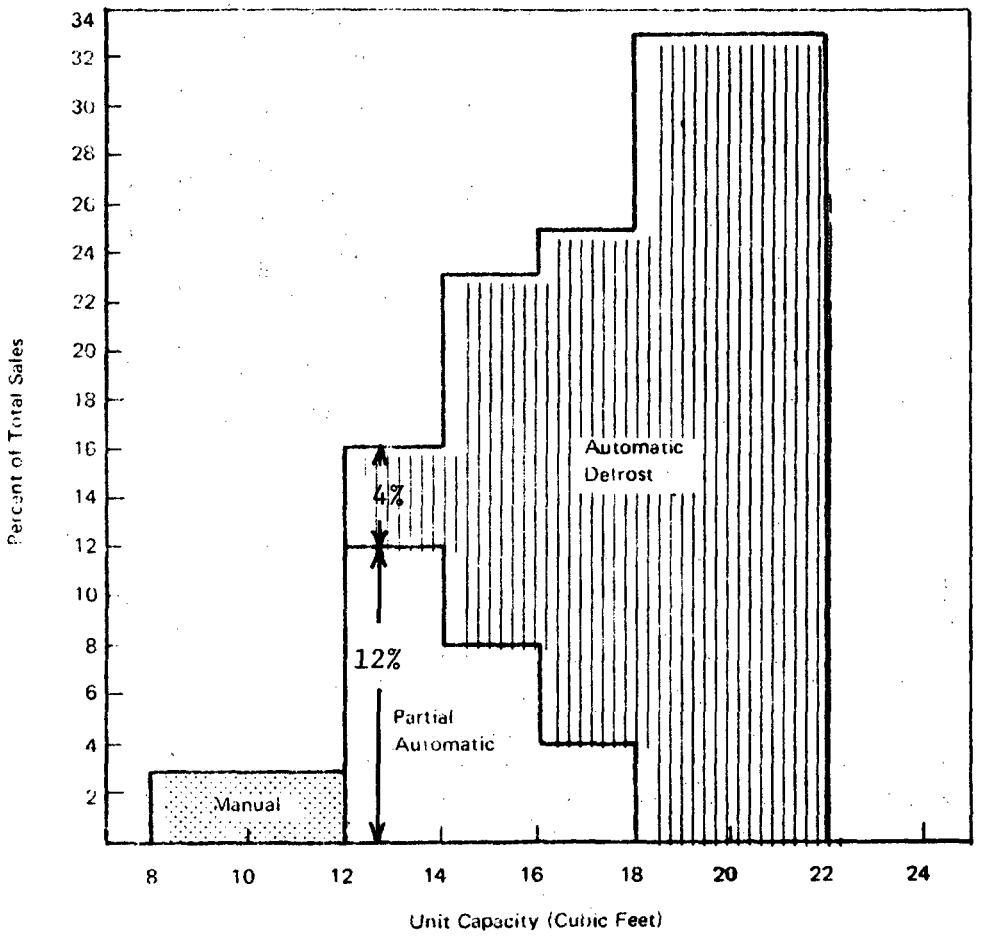
Through a series of public hearings and evaluation of the present product efficiencies , the Department of Energy has published (Federal Register Vol. 44, No. 1, Tuesday, January 2, 1979) maximum technologically feasible energy factors (energy factor is the efficiency and is defined as the ratio of the unit storage volume to the unit daily energy consumption in kilowatt-hours) for refrigerator-freezers for the 1980 market. These levels of efficiency represent the best design practice expected of manufacturers. The baseline units we have selected for this project have energy factors very close to the maximum technologically feasible and represent what we believe to be the best insulation practice likely to be available in 1980. Table 3.4 summarizes the unit design features.

As part of the same program*, Arthur D. Little developed a computer program for predicting the performance of refrigerators and freezers. The model was calibrated against laboratory data based on the standard DOE test procedure. The model is a steady state approximation to the cyclic performance of the refrigerator-freezer and is governed by the general equation given below:

$$\frac{\text{kilowatt hours}}{\text{day}} = 0.024 \left(\frac{\text{heat load thru insulation plus other steady heat loads}}{\text{refrigeration capacity} - \text{internal heat generation}} \right) \left(\text{compressor watts} + \text{fan \& heater watts} \right)$$

This relation can be used to estimate the effect of insulation improvements on the daily unit energy consumption for the Federal Test Procedure (a 90°F closed door test) which is thought to approximate actual in-home energy consumption. The 90°F closed door test is roughly equivalent to a 70 to 80°F kitchen environment with door openings and food loads. The 90°F closed door test, however, tends to overestimate the energy savings from insulation improvements compared to what is likely to occur in the field when the user load (food, door openings, etc.) bypass the insulation. The daily energy consumption equation above can be used to predict both the unit performance in the 90°F test and in a 70°F

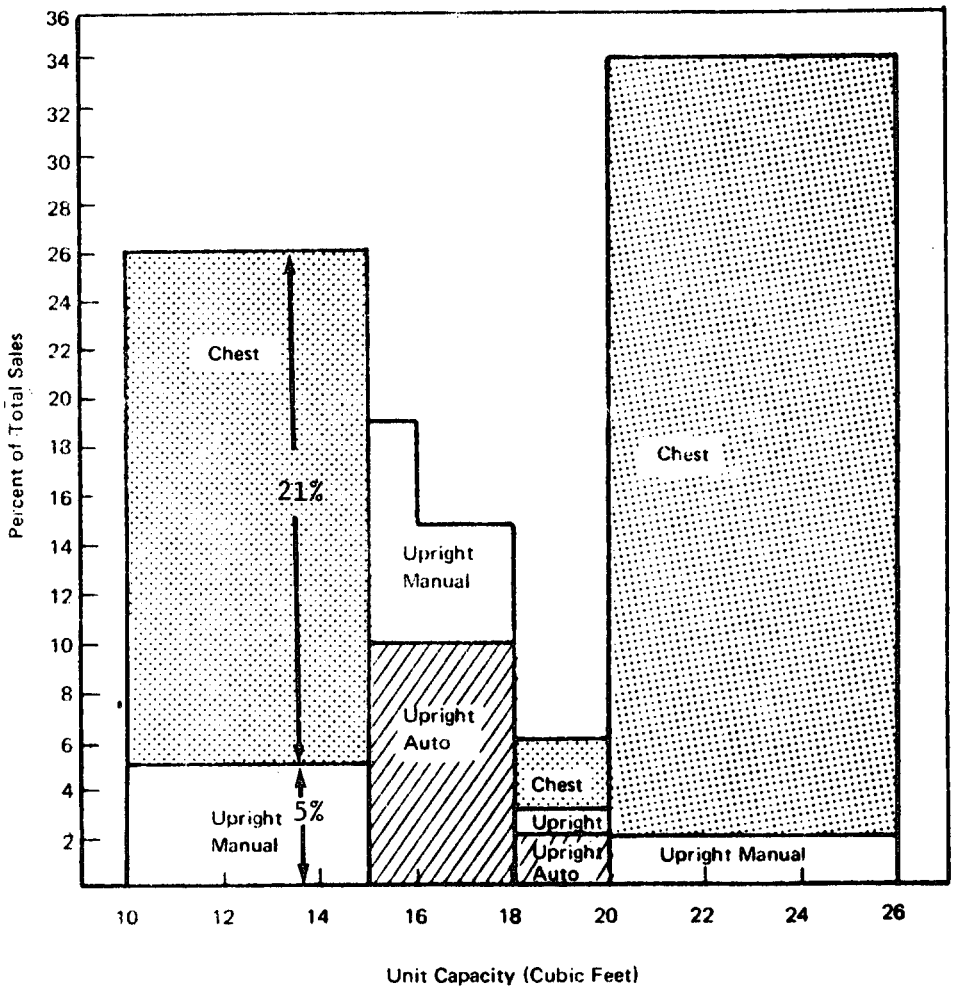
* Arthur D. Little, Inc., "Study of Energy-Savings Options for Refrigerators and Water Heaters, Volume 1: Refrigerators," prepared for the Federal Energy Administration under contract CO-04-50228-00, May, 1977.



Source: % by Size: Arthur D. Little, Inc., estimates and *Merchandising* March 1976,
% by Type: Arthur D. Little, Inc., Estimates.

Example: Approximately 3% of total sales are manual defrost units between 8 and 12 ft^3, 12% of total sales are partial automatic units between 12 and 14 ft^3, and 4% of total sales are automatic defrost units between 12 and 14 ft^3.

FIGURE 3.2 REFRIGERATOR AND REFRIGERATOR-FREEZERS - SALES DISTRIBUTION



Source: Distribution of Chest Freezers: Directory of Refrigerators and Freezers by Type – California Energy Commission.
% by Size: Sept. 1978 Arthur D. Little, Inc., estimates and *Merchandising* March 1976.

Example: Approximately 5% of total sales are Upright Manual Defrost units between 10 and 15 ft^3 and 21% of total sales are chest freezers between 10 and 15 ft^3.

FIGURE 3.3 FREEZER SALES DISTRIBUTION

	Refrigerator-Freezer	Chest Freezer
Size (ft^3)	16	23
Nominal Insulation, Foam (in)	3	2
Defrost	Automatic	Manual
Condenser	Hot wall	Back Mounted Static
Compressor (HP)	1/6	1/4
Baseline (kwh/day)	3.01	3.13
Energy Factor $\left(\frac{ft^3}{Kwh/day}\right)$	6.0	18.1
Internal Heat (Btu/hr)		
Hot wall condenser	66.9	0
Fan	51.5	0
Heater	52.8	0
Heat Gain Through Insulation (Btu/hr)	246.2	440
Refrigeration capacity (Btu/hr)	607	767
Fan and heater electric (watts)	46	0

TABLE 3.4 BASELINE UNIT RESIDENTIAL REFRIGERATOR AND CHEST FREEZER SPECIFICATIONS USING 90°F CLOSED DOOR TEST

room test with additional steady heat loads due to door openings and food loads (entered in the term "Other Steady Heat Loads").

To estimate the magnitude of these loads in a 70°F environment, first, the energy consumed by the baseline units in a closed door 70°F test was estimated. Next, using the data for the 70°F test, the above equation was solved for "Other Steady Heat Loads" using the 90°F closed door test energy use per day for the left hand side.

The energy savings through improved insulation are given in Table 3.5 and 3.6. These simulation results indicate the energy savings through improved insulation under the two test conditions highlighting the sensitivity of design improvements to the test procedure used to evaluate the unit performance. In evaluating the feasibility of improved insulations, we will use the 90°F standard test conditions. However, if a system appears marginal, the 70°F values will be used to bracket the savings. Savings versus insulation resistivity based on the 90° test conditions are shown in Figure 3.4. For this case, the savings for both refrigerator-freezers and freezers are approximately equal.

Insulation Resistivity (Constant Wall Thickness of 3") (hr ft^2°F/Btu·in)	90°F Closed Door Test			Estimated 70°F Operation		
	Kwh/Day	Annual Kwh Saving	Annual $ Savings*	Kwh/Day	Annual Kwh Saving	Annual $ Savings*
7.71	3.01	Baseline		3.01	Baseline	
15.4	1.77	453	$22.60	2.32	251	$12.60
38.6	1.02	726	$36.30	1.93	394	$19.80
77	0.78	814	$40.70	1.79	448	$22.40

* Based on electricity at $0.05/Kwh

TABLE 3.5 EFFECT OF IMPROVED INSULATION OF A RESIDENTIAL REFRIGERATOR-FREEZER

(Starting From the Best 1980 Practice)

Insulation Resistivity (Constant Wall Thickness of 2") (hr ft^2°F/Btu·in)	90°F Closed Door Test			Estimated 70°F Operation		
	Kwh/Day	Annual Kwh Savings	Annual $ Savings*	Kwh/Day	Annual Kwh Savings	Annual $ Savings
7.71	3.13	Baseline		3.13	Baseline	
15.4	1.87	460	$23.00	2.19	342	$17.10
38.6	1.14	726	$35.00	1.65	540	$27.0
77	0.81	847	$42.20	1.41	627	$31.30

* Based on electricity at $0.05/Kwh

TABLE 3.6 EFFECT OF IMPROVED INSULATION ON A RESIDENTIAL CHEST FREEZER

(Starting From the Best 1980 Practice)

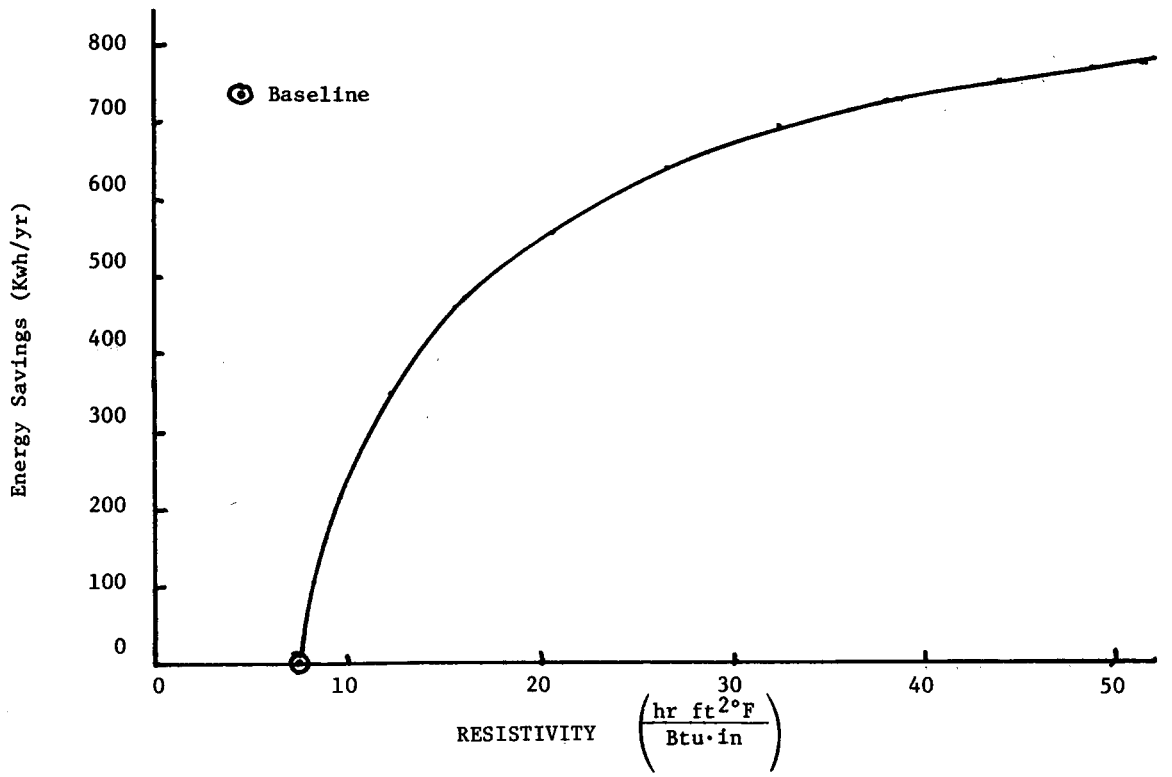


FIGURE 3.4 ENERGY SAVINGS FROM THE USE OF IMPROVED INSULATIONS IN REFRIGERATORS AND FREEZERS BASED ON 90°F CLOSED DOOR TEST

3.4 Effect of Improved Insulation in Residential Electric and Gas Water Heaters

The baseline units which represent the best 1980 insulation practice were selected based on the data shown in Tables 3.7 and 3.8. These tables show the relevant characteristics of both standard and energy conserving water heaters that are presently being sold in the United States. Based on conversations with manufacturers, we anticipate that the energy conserving units shown in these tables will represent the best 1980 insulation practice. Therefore, we have selected the units shown as being typical.* A 52 gallon electric and a 40 gallon gas-fired water heater are used as the baseline sizes, based on data from earlier programs.**

The properties of the insulation shown in these tables are summarized in Table 3.9. The insulation used is typically a glass fiber type with a density ranging from 1/2 lb/ft^3 to 2 lb/ft^3. The resistivities and typical $costs/ft^3$ for these densities are also shown.

For another project,*** in order to estimate the energy savings which result from arbitrary levels of improvement in the insulation systems used in these water heaters, laboratory measurements were made on an electric and gas water heater. Based on data from these tests, estimates of the energy partitioning for the baseline units were made (Table 3.10). Energy savings estimates were made by calculating the reduction in the losses through the insulation as the insulation conductivity was reduced.

The effect of improving the insulation is shown in Table 3.11 and Table 3.12, and plotted in Figure 3.5. Also shown in Table 3.11 and Table 3.12 is the savings from the use of improved insulations on an annual basis in both Btu/yr and in dollars. From this data it can be seen that the incremental savings drops off significantly with improved conductivity and a factor of 10 improvement in the insulation system results in only slightly better than doubling the energy saved by improving the insulation by a factor of 2.

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* Following the completion of this analysis manufacturers began to introduce water heaters with polyurethane foam insulation. They also reduced the thickness used so that the overall R (hr ft^2°F/Btu) value is approximately 20% greater than those used in this study.

** Study of Energy-Savings Options for Refrigerators and Water Heaters, Contract No. CO-04-50228-00 by Arthur D. Little, Inc., May, 1977.

*** Unpublished Arthur D. Little Data.

MANUFACTURER MODEL	INPUTS (Btu/hr)		DIMENSIONS (INCHES)	INSULATION DENSITY (lb/ft^3)	INSULATION THICKNESS (in)
State Industries	Pilot	Burner			
AX-40-NRT8 *	700	46,000	18 X 56	0.5	3/4
PX-40-NRT8 **	400	40,000	20 X 58	1.0	1 1/2
Rheem					
22-40-2*	400	45,000	17 3/4 X 56	0.5	3/4
29J-40-9**	400	30,000	19 3/4 X 56	0.5	2
A. O. Smith					
PGX-40 *	750	40,000	18 1/4 X 54	0.5	3/4
PGC-40 **	400	32,500	20 1/4 X 53 3/8	1.0	1 1/2
Mor-Flo/American					
GB400H *	350	46,000	18 X 58 1/2	1.0	3/4
ESMG41S**	350	33,000	18 X 59 1/8	2.0	1
Bradford-White					
M52S10D*	700	40,000	18 X 58	0.5	1
MPBS10D**	400	35,000	18 X 58	1.0	1
Basis of Analysis	400	35,000	20 X 58	1.0	1 1/2

* Standard

** Energy Conserving

TABLE 3.7 SUMMARY OF CHARACTERISTICS OF TYPICAL 1978 STANDARD AND ENERGY CONSERVING 40 GALLON GAS-FIRED WATER HEATER

MANUFACTURER MODEL	DIMENSIONS (INCHES)	INSULATION DENSITY (lb/ft^3)	INSULATION THICKNESS (in)
State Industries			
AX-52-2R58*	22 X 50 1/4	0.5	2
PX-52-2R58**	22 X 53 1/4	2.0	2
Rheem			
Standard*	19 3/4 X 58 1/2	0.5	2
Energy Cons.**	22 1/4 X 59	0.5	4
A. O. Smith			
PEN-52*	20 1/2 X 54 1/8	0.5	2
PEC-52**	21 3/4 X 55 1/4	1.0	2 1/2
Mor-Flow/American			
MER 52D*	21 X 50	1.0	1 1/2
ESM 52D**	21 X 62 1/2	2.0	2 1/2
Bradford White			
M52S10D*	20 X 47 1/2	0.5	1
MPBS10D**	20 X 58 1/2	2.0	2
Basis of Analysis	21 X 58	2.0	2 1/2

NOTE: Electric water heaters are offered with several element sizes 4.5 Kwh is typical.

* Standard

** Energy Conserving

TABLE 3.8 SUMMARY OF CHARACTERISTICS OF TYPICAL 1978 STANDARD AND ENERGY CONSERVING 52 GALLON ELECTRIC WATER HEATER

DENSITY (lb/ft^3)	CONDUCTIVITY AT MEAN TEMP. OF 115°F $\left(\frac{Btu\ in}{hr\ ft^2\ °F}\right)$	THERMAL RESISTIVITY $\left(\frac{hr\ ft^2\ °F}{Btu\ in}\right)$	COST (1978) ($\$/ft^3$)
0.5	0.37	2.7	0.40
1.0	0.30	3.3	0.70
2.0	0.25	4.0	0.95

TABLE 3.9 PROPERTIES OF GLASS FIBER INSULATION FOR WATER HEATERS

	Gas (40 Gallon) (Btu/day)	Electric (52 Gallon) (Btu/day)
To Water	47743	47743
Through Insulation	10320	5760
Through Feed Throughs*	2400	1440
Stack Standby Losses	8640	-
Recovery Losses (excluding insulation)	15043	-
Total Daily Input	84146	54943

* Feed throughs are conduction paths such as pipes, supports etc. which pass through the insulation.

TABLE 3.10 ENERGY PARTITIONING IN BASELINE RESIDENTIAL WATER HEATERS USING BEST 1980 INSULATION

$\left(\frac{R/in}{\frac{hr\ ft^2 °F}{Btu\ in}}\right)$	Energy Through Insulation (Btu/day)	Savings	
		(10^6Btu/year)	($/year*)
3.3 (Base-line)	10320	-	-
5	7056	1.19	4.52
10	3672	2.43	9.23
15	2472	2.86	10.87
20	1896	3.07	11.56
25	1512	3.21	12.20

* Gas @ $3.80/$10^6$ Btu

TABLE 3.11 SAVINGS FROM THE USE OF IMPROVED INSULATION IN RESIDENTIAL GAS-FIRED WATER HEATERS

R/in $\left(\frac{hr \quad ft^2 \quad °F}{Btu \cdot in}\right)$	Energy Through Insulation (Btu/day)	Savings	
		(10^6 Btu/year)	($/year)*
4 (Baseline)	5760	-	-
10	2376	1.24	18.17
15	1608	1.52	22.27
20	1200	1.66	24.32
25	960	1.75	25.64

* $0.05/Kwh ≡ $14.65/$10^6$ Btu

TABLE 3.12 SAVINGS FROM THE USE OF IMPROVED INSULATION IN RESIDENTIAL ELECTRIC WATER HEATERS

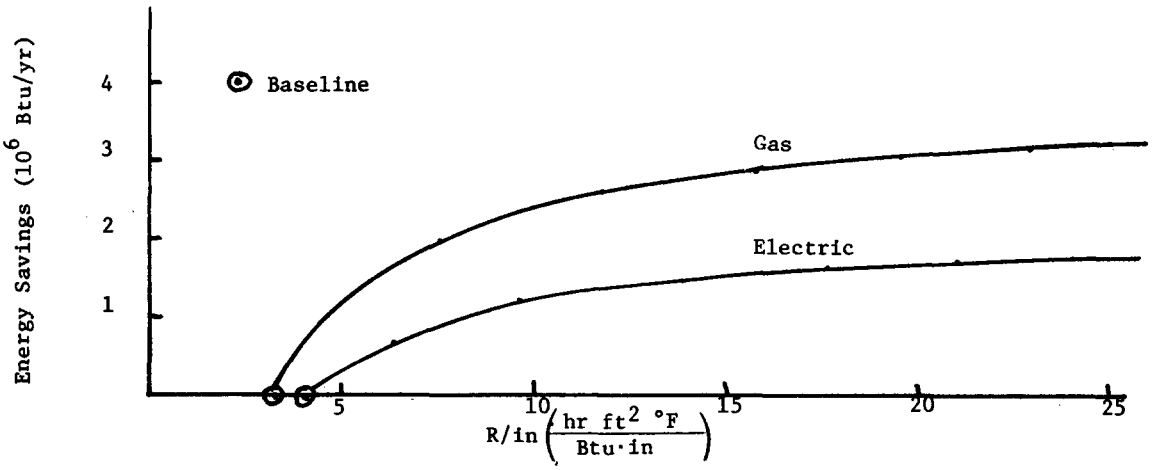


FIGURE 3.5 SAVINGS FROM IMPROVED INSULATIONS IN RESIDENTIAL GAS AND ELECTRIC WATER HEATERS

3.5 Effect of Improved Insulation in Commercial Ovens

Based on conversations with manufacturers, the specifications for baseline units which represent the best 1980 insulation practice are as shown in Table 3.13. Figure 3.6 shows the effect of temperature and density on the conductivity of the insulation used, Owens-Corning SR-26.

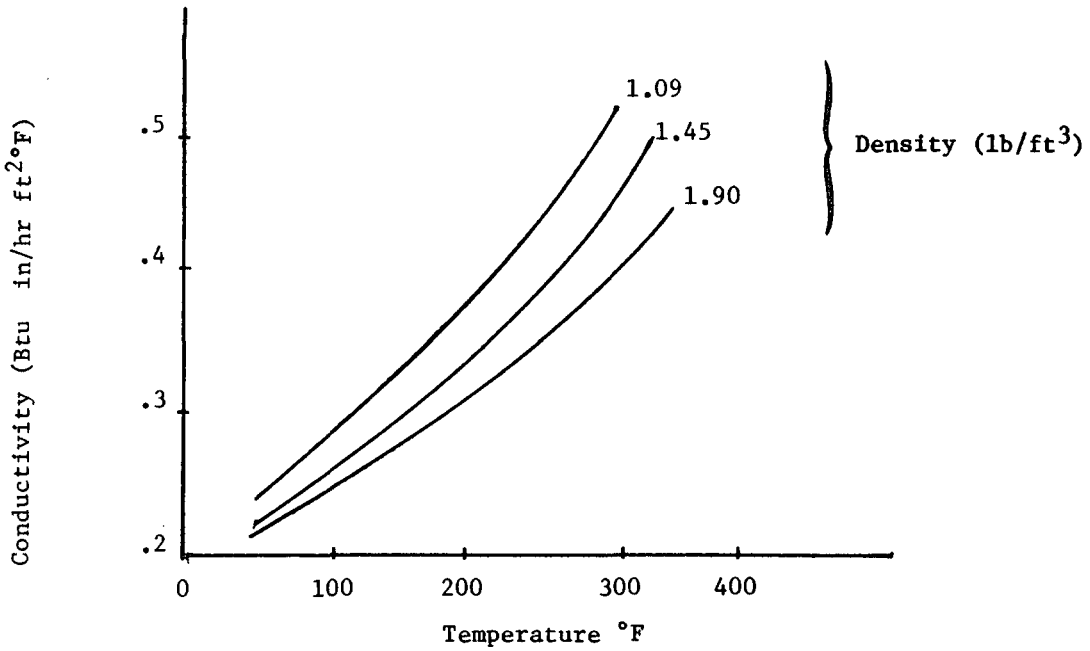
The daily conduction loss for each oven was calculated treating the oven as a cube within a cube with insulation filling the space between the cubes. An oven temperature of 350°F and ambient temperature of 70°F were used. Heat transfer coefficients appropriate for vertical and horizontal, interior and exterior oven surfaces were used (1.0 to 2.0 Btu/hr ft^2°F depending on the temperature and orientation).

Assuming a daily duty cycle of twelve hours, the daily energy savings, annual energy savings, (based on 312 days operation per year) and the dollars value of these savings for electric and gas models were then computed. The results are summarized in Tables 3.14 and 3.15. The results of arbitrarily improving the insulation are shown in Figure 3.7.

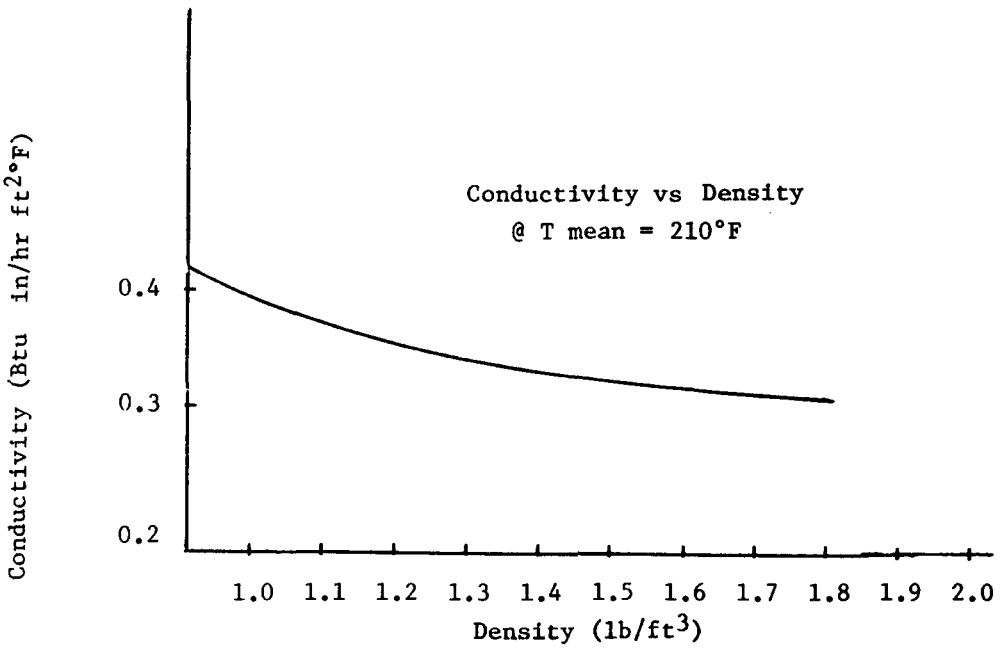
MANUFACTURER MODEL	ENERGY INPUT (BTU/hr)	DIMENSION (INCHES) - LENGTH X BREADTH X THICKNESS (INSULATION DENSITY - LBS/FT^3)				
		LT SIDE	RT SIDE	REAR	TOP	DOOR
Double oven side by side free standing	70,000	20 1/8 X 22 3/4 X 1 (1.09)	20 1/8 X 22 3/4 X 1 (1.09)	2X 28 X 14 3/8 X 1 (1.09)	2X 23 5/8 X 28 1/2 X 1 (1.09)	2X 29 1/2 X 16 X 2 (1.09)
Single oven - free standing	45,000	21 1/2 X 26 X 9/16 (1.94)	21 1/2 X 26 X 1 9/16 (1.7)	19 1/2 X 18 1/2 X 1 (1.09)	28 7/8 X 30 1/4 X 1 (1.09)	27 1/2 X 16 1/4 X 2 (1.09)

NOTE: Bottom is uninsulated.

TABLE 3.13 CHARACTERISTICS OF REPRESENTATIVE COMMERCIAL OVENS



(a) Conductivity as a Function of Temperature for 3 Densities of Fiberglass Oven Insulation



(b) Conductivity as a Function of Density at a Mean Temperature of 210°F for Compressed Fiberglass Insulation

SOURCE: Owen Corning Sales Literature

FIGURE 3.6 RELATIONSHIP BETWEEN TEMPERATURE, DENSITY AND THERMAL CONDUCTIVITY OF FIBERGLASS OVEN INSULATION

	(BASELINE)						
R/in $\left(\frac{hr\ ft^2 °F}{Btu\ \ in}\right)$	2.54	5.08	7.62	10.16	12.7	15.24	25.4
Savings/day (Btu)	*	6937	10113	11929	13057	13867	15659
Annual Savings (10^6 Btu/Yr)	-	2.16	3.16	3.72	4.07	4.33	4.89
Value of Savings ($/yr)							
Electricity @ $.05/Kwh	-	31.64	46.29	54.50	59.62	63.43	71.64
Gas @ $3.80/$10^6$Btu	-	8.21	12.01	14.14	15.47	16.46	18.59

* NOTE: Estimated baseline energy loss through insulation is 5.8 X 10^6 Btu/yr assuming 12 hours/day operation for 312 days/yr.

TABLE 3.14 SAVINGS FROM THE USE OF IMPROVED INSULATION IN DOUBLE SIDE-BY-SIDE COMMERCIAL OVENS

R/in $\left(\frac{hr\ ft^2 °F}{Btu\ in}\right)$	2.54 (BASELINE)	5.08	7.62	10.16	12.7	15.24	25.4
Savings/day (Btu)	*	4835	7125	8461	9302	9910	11272
Annual Savings (10^6 Btu/yr)	-	1.51	2.22	2.64	2.90	3.09	3.52
Value of Saving ($/yr)							
Electricity @ $0.05/Kwh	-	22.12	32.52	38.68	42.48	45.27	51.57
Gas @ $3.80/$10^6$Btu	-	5.74	8.44	10.03	11.02	11.75	13.38

*NOTE: Estimated baseline energy loss through insulation is 4.2 X 10^6 Btu/yr assuming 12 hours/day operation for 312 days/yr.

TABLE 3.15 SAVINGS FROM THE USE OF IMPROVED INSULATION IN SINGLE FREE-STANDING COMMERCIAL OVENS

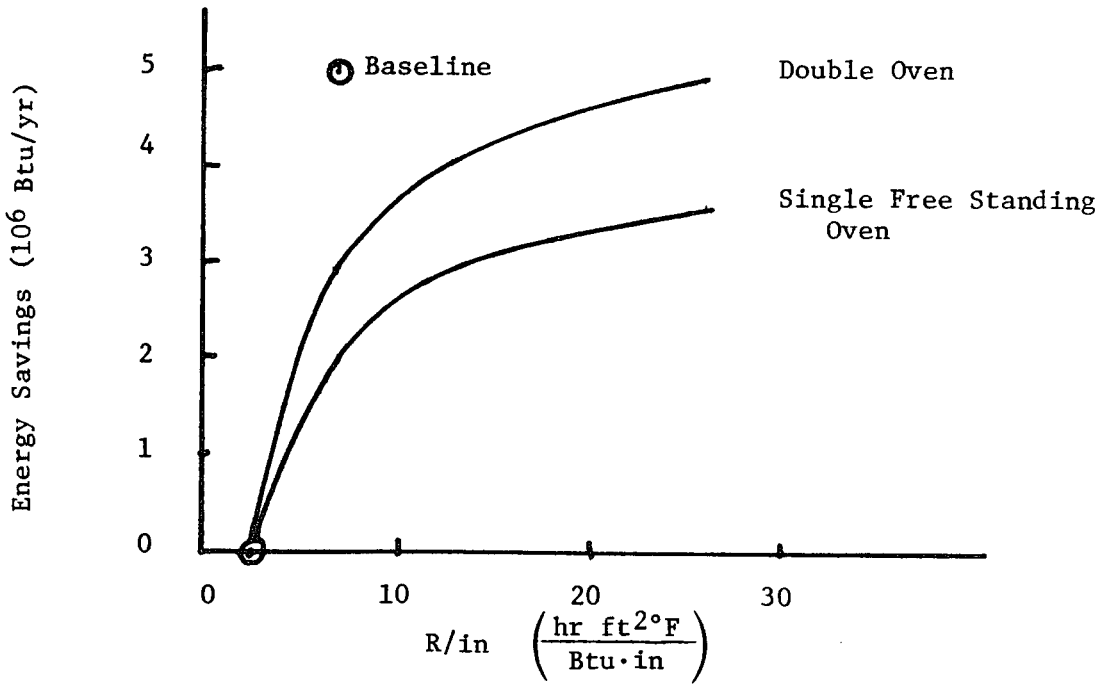


FIGURE 3.7 ENERGY SAVING FROM USE OF IMPROVED INSULATION IN COMMERCIAL OVENS

3.6 Effect of Insulation Improvements in Commercial Display Refrigeration

Table 3.16 summarizes the estimated annual nationwide energy consumption of the several commercial refrigeration types. The largest energy users can be classified as either display refrigeration units or central system refrigerators or freezers (warehouses).

Refrigerated Warehouse

Table 3.17 shows that about 98 tons of cooling is required to meet the heat flow load through the insulation. Assuming a refrigeration unit coefficient of performance*(COP) of .6 and an energy cost of $0.05/Kwh, the following savings can be expected from an improved insulation system.

Refrigerated Warehouse
100,000 Sq. Ft. Freezer
($0.05 Kwh)

Insulation	Annual Energy Savings Cost
2 X Baseline	$125,000
5 X Baseline	200,000
10 X Baseline	225,000

Although these savings are substantial, insulation thickness is not a significant constraint in this application and thicker conventional insulations could be used.

Commercial Display Units

Commercial Display Units are typically modular in design, and are manufactured in 8-foot lengths for field assembly to the desired overall size (typically 40 to 60 feet). The condensing units are remote and the evaporators and defrost functions are contained in each of the modules. The display units may be either open display shelves or reach-in units (units with doors) with full view glass doors. Merchandizing of the products plays a large role in the selection of open or reach-in type units. In the past retailers have preferred the open type shelving as it increases sales. Over the past five years the popularity of the reach-in units has grown considerably until they now hold more than 50% of the market.**

* COP is the ratio of cooling capacity to electric power requirements.

** Estimate from Hussman Refrigeration.

	Nationwide Energy Consumption (10^{12} Btu/yr)		
	ADL Motor* Study		Thermo** Electron
Central System (Warehouse)	109		114.8
Unit Cooler/Display (Supermarket)	92		91.6
Beverage Refrigerator	17		
Water Cooler	12	37.5	44.5
Ice Maker	8.5		
	238.5		250.9

* Arthur D. Little, Inc.,"Energy Efficiency and Electric Motors," NTIS, No. PB-259-129, August 1976.

** J. R. Hurley, "Energy Conservation Opportunities in Commercial Appliances," Prepared by Thermo. Electron Corporation for Oak Ridge National Laboratory, ORNL/Sub 726 1/1, December, 1978.

TABLE 3.16 SUMMARY OF COMMERCIAL REFRIGERATION ENERGY CONSUMPTION

Items of Refrigeration Load	Long-Term Storage		Short-Term Storage		Distribution Operation	
	Tons	Percent	Tons	Percent	Tons	Percent
Leakage	98	49	98	43	98	36
Infiltration	10	5	20	9	40	15
Internal Operation Load	50	25	56	24	62	22
Cooling of Goods Received	7	3	15	6	30	11
Other Factors	35	18	41	18	45	16
Total Design Capacity	200	100	230	100	275	100

TABLE 3.17 REFRIGERATION DESIGN LOAD FACTORS FOR A TYPICAL 100,000 SQ. FT., SINGLE FLOOR, FREEZER

The high energy cost of the open display refrigeration and its sizeable impact on store profits has accounted for the transition to reach-in units. In 1980 the best insulation practice is represented by the reach-in type display units and we have selected this unit for the analysis in this project.

The reach-in display is shown in cross section in Figure 3.8. The glass doors are a significant heat load and must be included in any design considerations of the insulation system.

Depending on the ambient store humidity, fogging or frosting of the outer panes of glass may significantly reduce the visual transmission. Therefore manufacturers prevent fogging or frosting by using electrical resistance heating of the outer pane to maintain the outer surface above the ambient dew point. This is accomplished with a thin film of conductive material (metal film) that is transparent but which will conduct sufficient current with the proper ohmic resistance to create the required surface heating. Additional electric heat is applied to the door flanges to prevent sweating. These anti-sweat heaters add a substantial heat load on the refrigeration unit.

The display units offer a net cooling to the building area and can be considered to provide useful air conditioning during the cooling season. During the heating season, heat rejected by the condensing units can be returned to the building space offsetting the cooling effect and providing a net heating gain as a result of operating the display units. It can be argued that improved insulation on the refrigerated display is not required as the unit presently provides useful cooling to the air conditioned space. This, however, does not prove to be the case when one considers the efficiency of providing space cooling at the evaporator temperatures of the display. The display units' coefficient of performance (COP) is approximately .8 while that of a conventional air-cooled rooftop air conditioner is approximately 2.0. Insulating the display units and shifting the space cooling requirement from the display units to larger rooftop air conditioning units will reduce energy consumption as the rooftop units operate at a higher COP. The added cost of increased air conditioner capacity should be included as part of any future comprehensive economic evaluation.

Allowing for the useful space cooling of the reach-in unit and assuming that improved insulation will require additional rooftop air conditioning capacity, the energy savings potential of improved insulation is summarized in Table 3.18 and shown graphically in Figure 3.9.

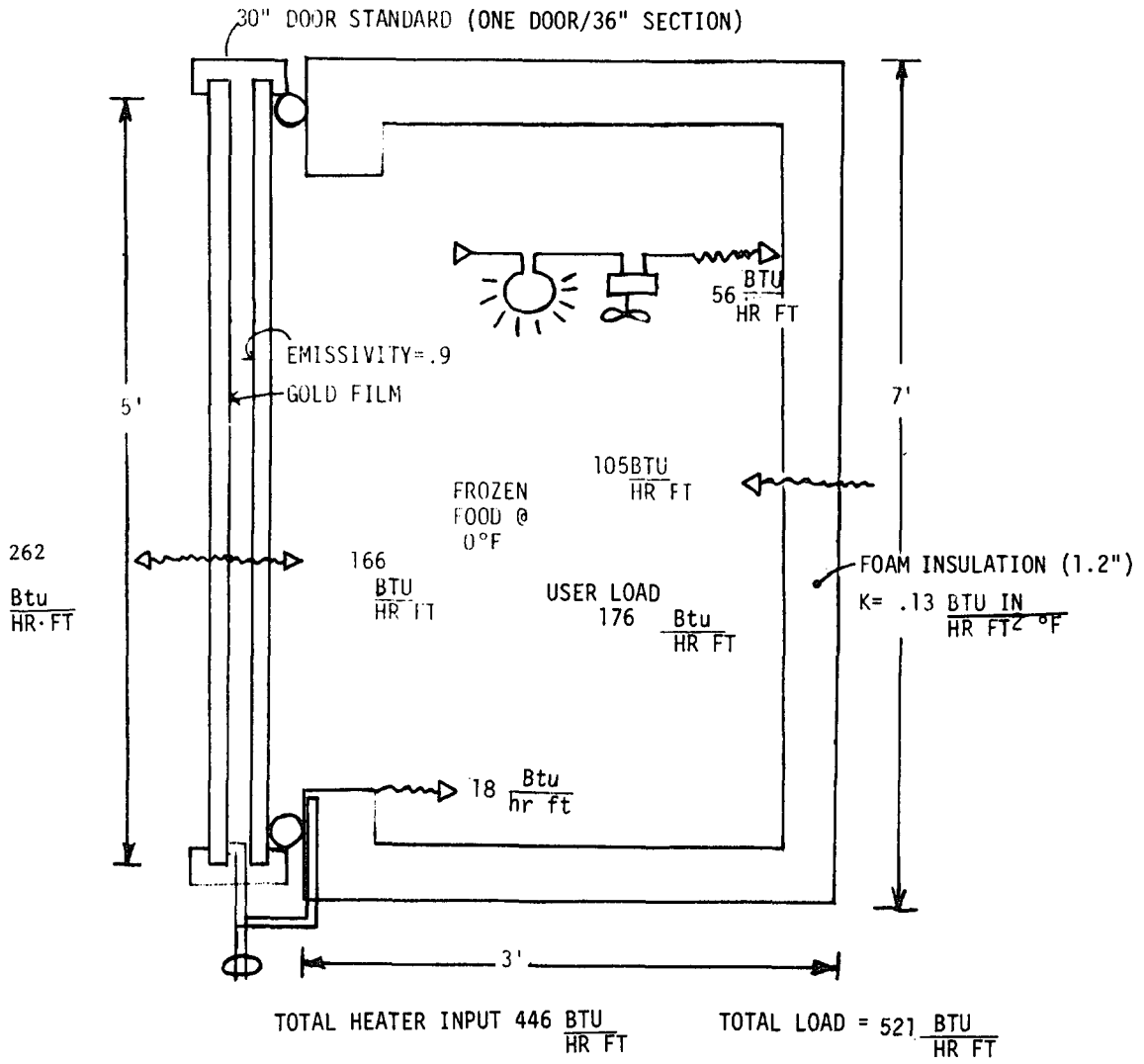


FIGURE 3.8

ENERGY PARTITIONING OF A REACH-IN DISPLAY FREEZER
(Ambient 75°F, 50%RH, 63°F Dew Point)

Insulation Resistivity* (hr ft^2 °F/Btu in)	Annual Energy and Cost Savings Per 3 Foot Door Section**	
WALL	$	Kwh
15.4	24	475
38.5	29	775
77	44	875
DOOR		
15.4	109	2175
38.5	156	3125
77	175	3500

*Baseline Unit value 7.7 hr ft^2°F/Btu in
**Based on electricity @ $0.05/Kwh

TABLE 3.18 ESTIMATED EFFECT OF INSULATION IMPROVEMENT ON COMMERCIAL DISPLAY UNIT

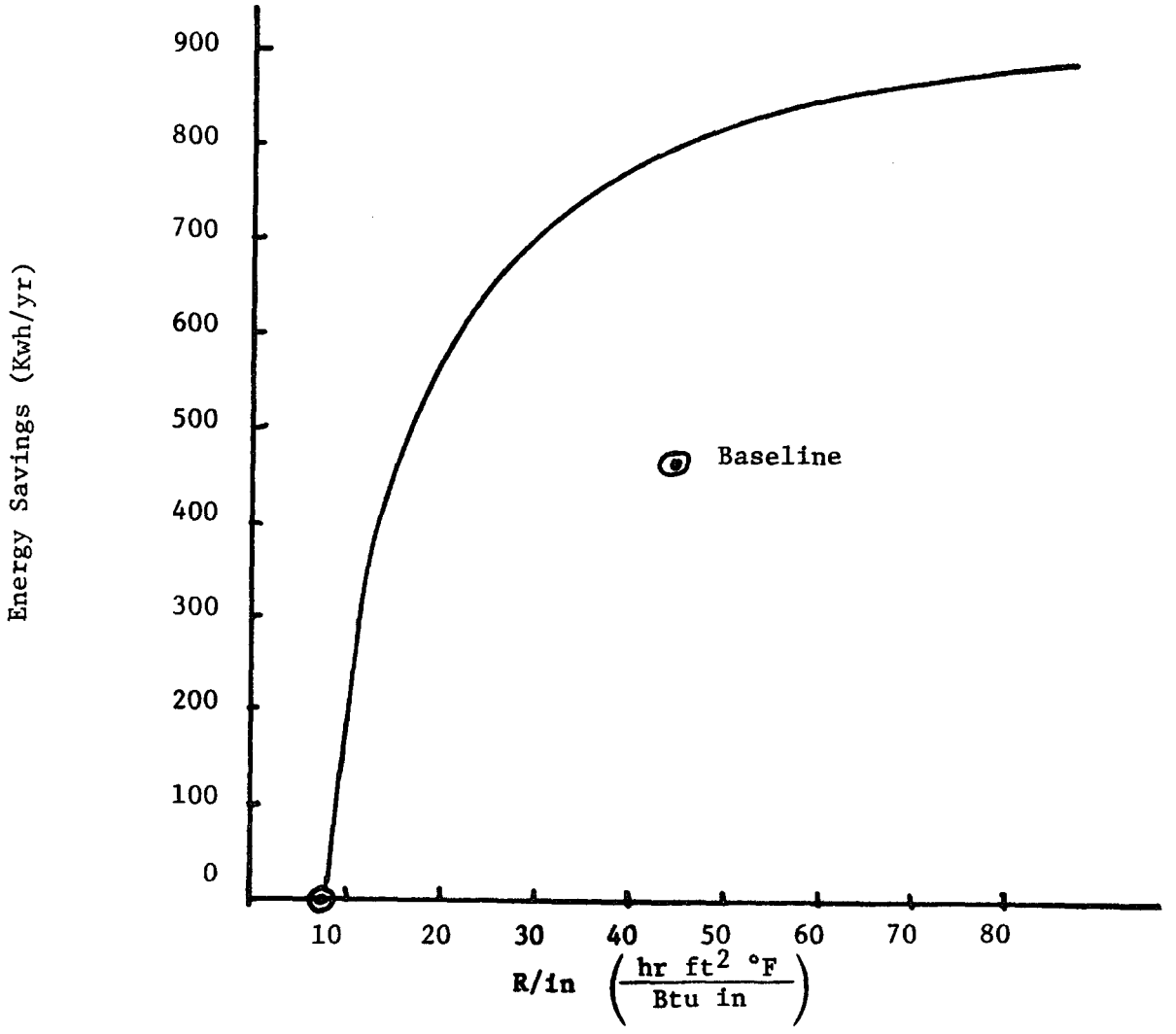


FIGURE 3.9 ENERGY SAVINGS FROM USE OF IMPROVED INSULATION IN DISPLAY FREEZERS

3.7 Effect of Improved Insulation in Commercial Water Heaters

INTRODUCTION

Limited data is available on commercial water heaters. A report by Thermo Electron Corporation (TECO)* gives some estimates of the total energy used by water heating in different sectors. However, it does not define a typical water heater in terms of storage capacity and input rate, nor does it give any indication of how much of the energy used is for useful water heating and how much of it is for losses. The Gas Appliance Manufacturers Association (GAMA) has data** on the sales of different types of water heaters. We have used the TECO inventory data, the GAMA data, information on various water heaters configurations from manufacturers' literature, and inputs from discussions with manufacturers to make an estimate of the energy partitioning and inventory as shown below.

BASELINE UNIT SELECTION

The data sources for the selection of baseline units are the inventory data from TECO*, manufacturers' specifications and GAMA data. TECO reports numbers of units as a function of input rate (see Table 3.19). In order to make estimates of losses and to identify baseline units, it is necessary to determine the capacity of these units. Figure 3.10 shows a plot of input rate as a function of capacity for various gas fired units. Manufacturers have indicated that the 80 gallon gas units are the most popular size. This is consistent with the data shown and the TECO report which indicated that 200,000 Btu/hr is the baseline input rate.

For electric water heaters, the storage capacity to input capacity ratio is larger than for gas-fired water heaters because higher input rates are available with gas-fired water heaters. Thus, a gas-fired unit can deliver more hot water than an electric unit with the same volumetric capacity because of the cost of supplying large current capacity to electric units. A plot of capacity versus input for electric units is shown in Figure 3.11. We have selected 120 gallons as the baseline unit size. This is the largest tank which can be sold without ASME certification. The one hour delivery capacity at 120,000 Btu/hr is consistent with that of an 80 gallon gas-fired unit with an input capacity of 200,000 Btu/hr.

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* J.R. Hurley, "Energy Conservation Opportunities in Commercial Appliances." Prepared by Thermo Electron Corporation (TECO) for Oak Ridge National Laboratory, ORNL/Sub-7261/1, Dist UC-95d, December, 1978.

** "Ten Year Statistical Abstract," published annually by GAMA, Arlington, Virginia.

	Input Rate (10^3 Btu/hr)		1975 Inventory* of Coil and Storage (10^3 Units)	1975 Inventory*** of Storage Type (10^3 Units)
	Range*	Assumed** Ave.		
Gas	75-100	88	155.7	89.4
	100-200	150	286.0	164.2
	200-300	250	173.4	99.5
	400-500	400	107.5	61.7
	500-1000	750	83.1	47.7
Electric	30-185	120	171	171

* Op. Cit. TECO, page 3-35
** ADL Estimate
*** Op. Cit. GAMA, page 3-35

TABLE 3.19 ESTIMATES OF COMMERCIAL WATER HEATER INVENTORY

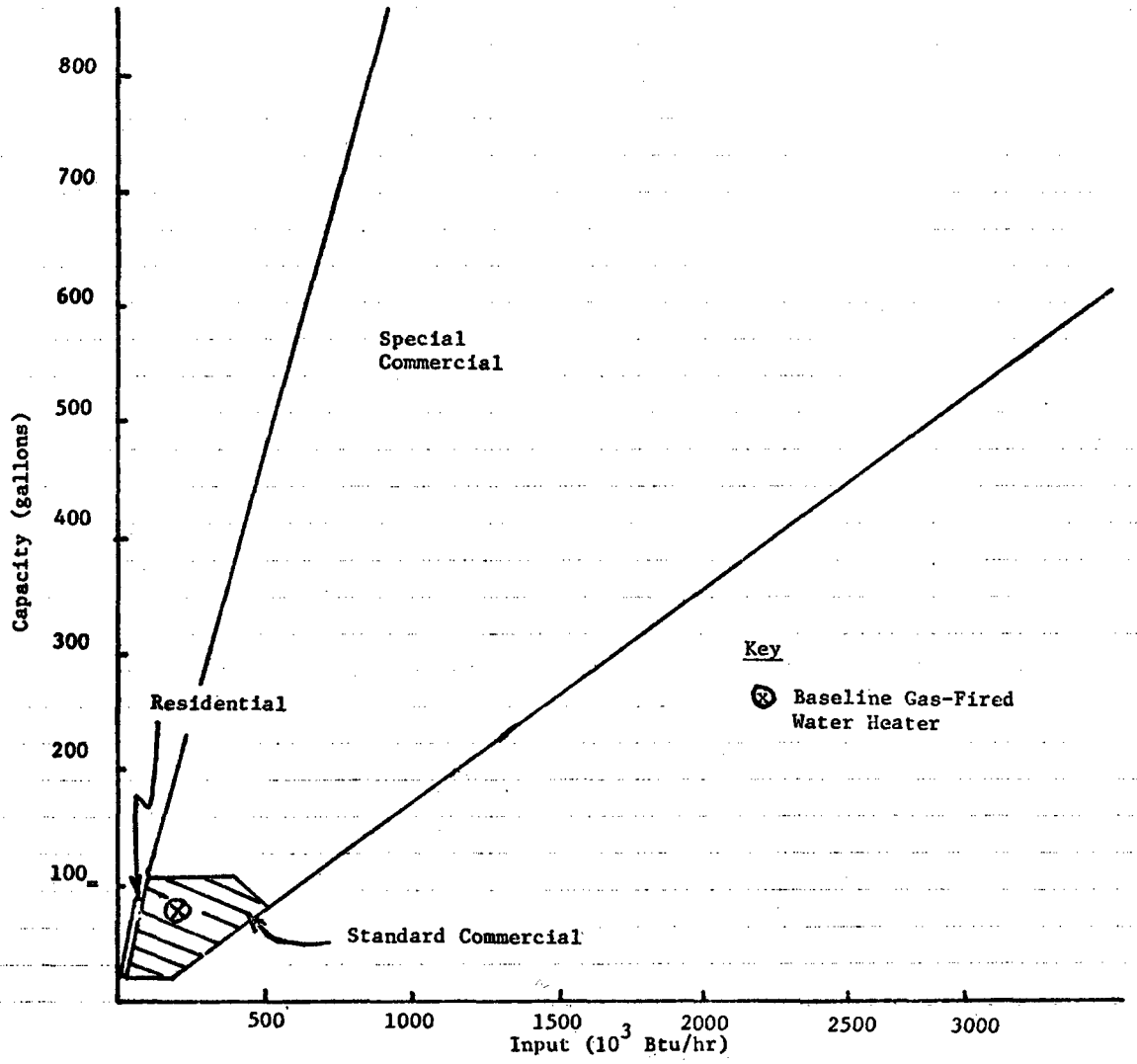

FIGURE 3.10 RELATIONSHIP BETWEEN STORAGE CAPACITY AND INPUT FOR GAS-FIRED WATER HEATERS

The inventory of units in 1975 was calculated from GAMA data which show a split of 57.4% storage units and 42.6% coil type. We consider only the storage type as the coil type are generally incorporated in boilers as part of space heating systems and are not free-standing units used only for hot water.

CALCULATIONS OF ENERGY SAVINGS FROM IMPROVED INSULATION

From manufacturers' specifications, the typical height to diameter ratio is 2.3 and the normal range of diameter is 26 to 44 inches. For these values, the overall conductance is approximately (see Figure 3.12 for schematic and nomenclature):

$$UA = \frac{A_{total}}{\frac{1}{h} + \frac{t}{k}}$$

where

h = 1.5 Btu/hr ft^2 °F natural convection heat transfer coefficient

t = insulation thickness (inches)

A_{total} = area of side and top of jacket (ft^2)

k = insulation conductivity $\left(\frac{\text{Btu in}}{\text{hr ft}^2\ °\text{F}}\right)$

If $T_{water} = 160°F$ and $T_a = 70°F$ then

$$Q_I = \frac{90\ A_{total}}{.67 + t/k}$$

The bottom is neglected for simplicity since bottoms of gas-fired units are uninsulated and losses through the bottoms of electric units are a small percentage of the total. During draws, cold water drawn into tank settles on the bottom and commercial units tend to have frequent water draws.

For gas-fired and electric water heaters best 1980 insulation practice will use 2 inches and 2.5 inches, respectively, of 1 lb/ft^3 glass fiber insulation with a mean conductivity of 0.30 Btu in/hr ft^2°F.

By plotting data on jacket surface area versus capacity obtained from manufacturers' specifications, an approximate relationship between capacity and area can be obtained. This is shown in Figure 3.13. The solid line shows the equation used in this analysis.

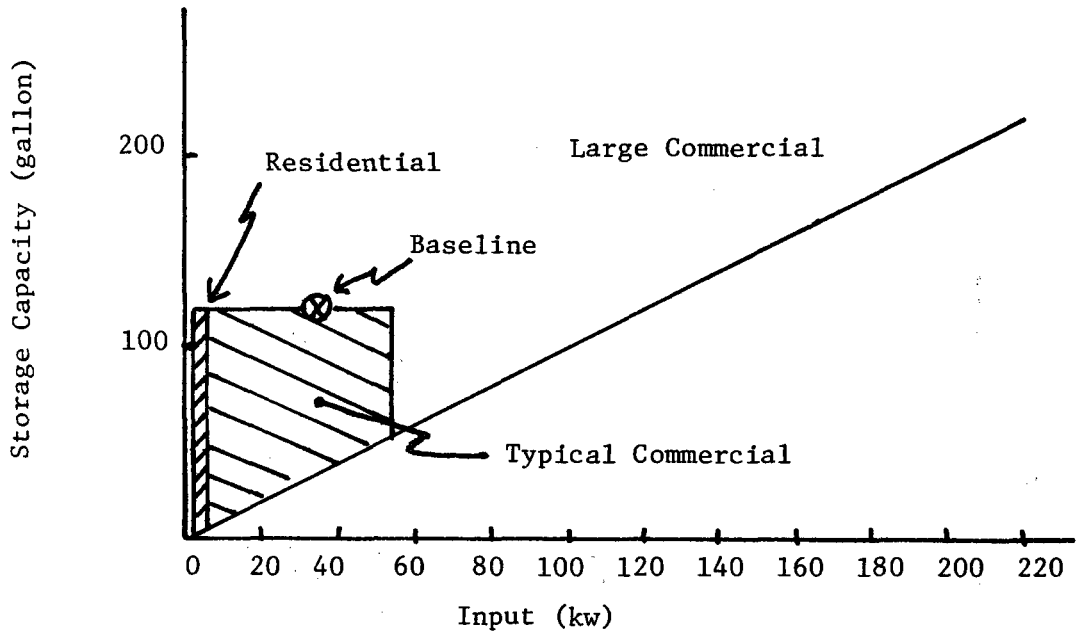

FIGURE 3.11 RELATION BETWEEN INPUT AND CAPACITY FOR ELECTRIC WATER HEATERS

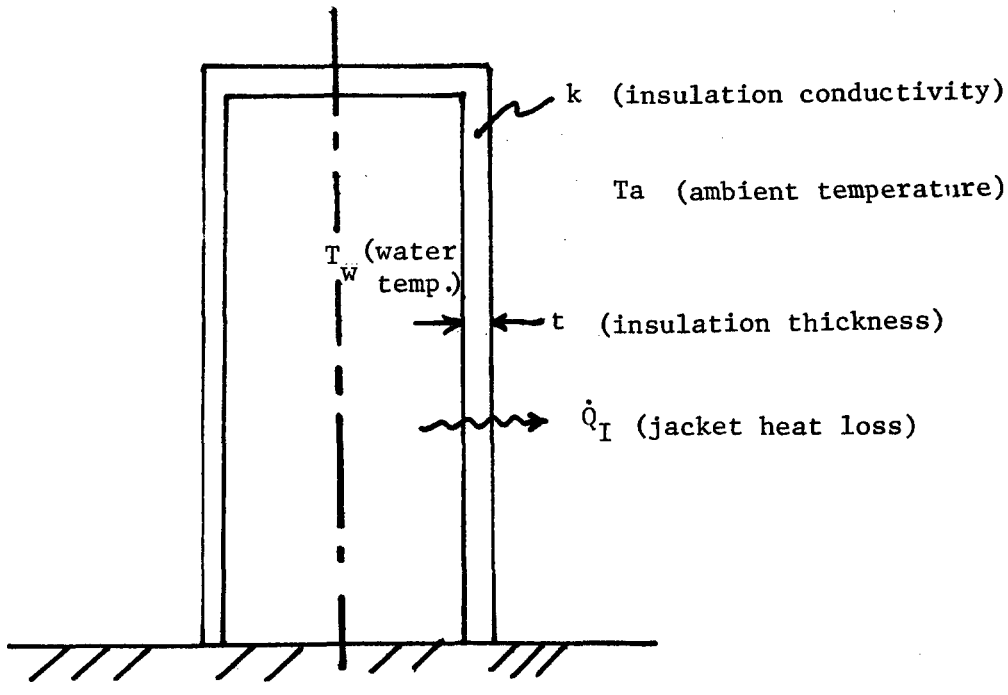

FIGURE 3.12 WATER HEATER NOMENCLATURE

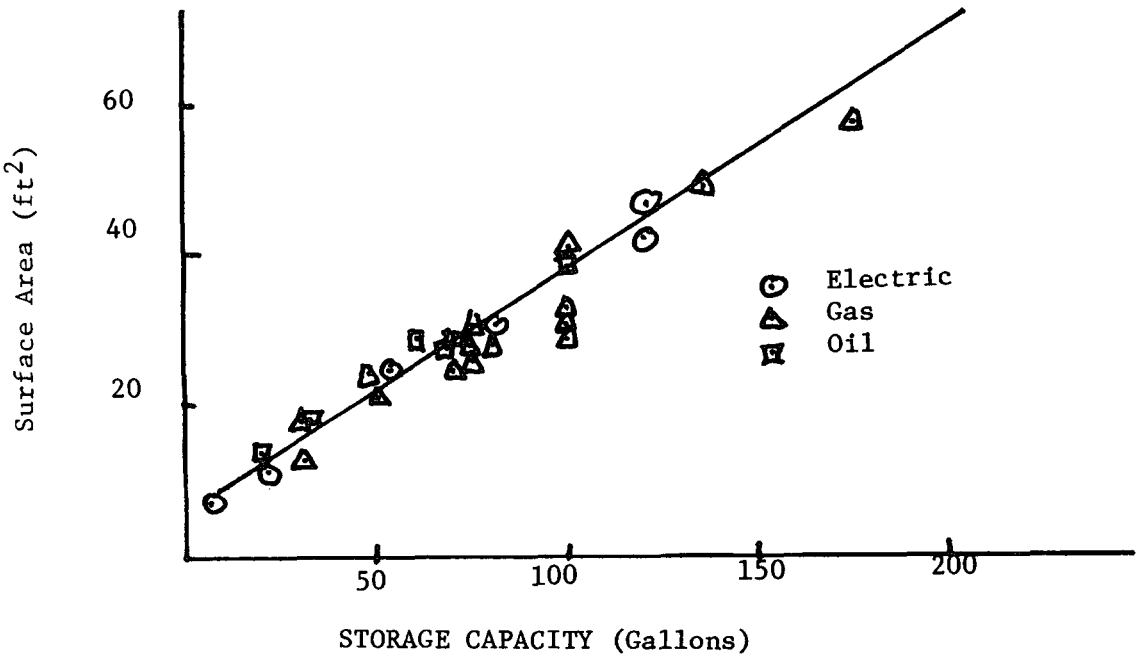

FIGURE 3.13 AREA OF WATER HEATERS JACKET AS A FUNCTION OF STORAGE CAPACITY

Using the data in Figure 3.13 and the relationship between heat loss and area shown above, estimates of losses and savings were made for units with different capacities as shown in Table 3.20. In addition to the baseline case, estimates were made for insulations with reduced conductivities. The energy savings for these units is plotted in Figure 3.14.

Resistivity $\left(\frac{hr\ ft^2 °F}{Btu\ in}\right)$	80 Gallons Gas-Fired ($A = 31\ ft^2$)			120 GALLON Electric ($A = 44\ ft^2$)		
	Loss through Insulation	Savings		Loss through Insulation	Savings	
	(10^6 Btu/yr)	(10^6 Btu/yr)	($/yr)*	(10^6 Btu/yr)	(10^6 Btu/yr)	($/yr)**
3.3 (Baseline)	3.61	-	-	3.89	-	-
5	2.44	1.17	4.45	2.63	1.26	18.46
10	1.26	2.35	8.93	1.35	2.54	37.21
15	0.85	2.76	10.49	0.91	2.98	43.66
20	0.64	2.97	11.29	0.68	3.21	47.03
25	0.51	3.10	11.78	0.55	3.34	48.93

* Gas at $3.80/$10^6$ Btu

** Electricity at $14.65/$10^6$ Btu ($0.05/Kwh)

TABLE 3.20 SAVINGS FROM THE USE OF IMPROVED INSULATION IN COMMERCIAL WATER HEATERS

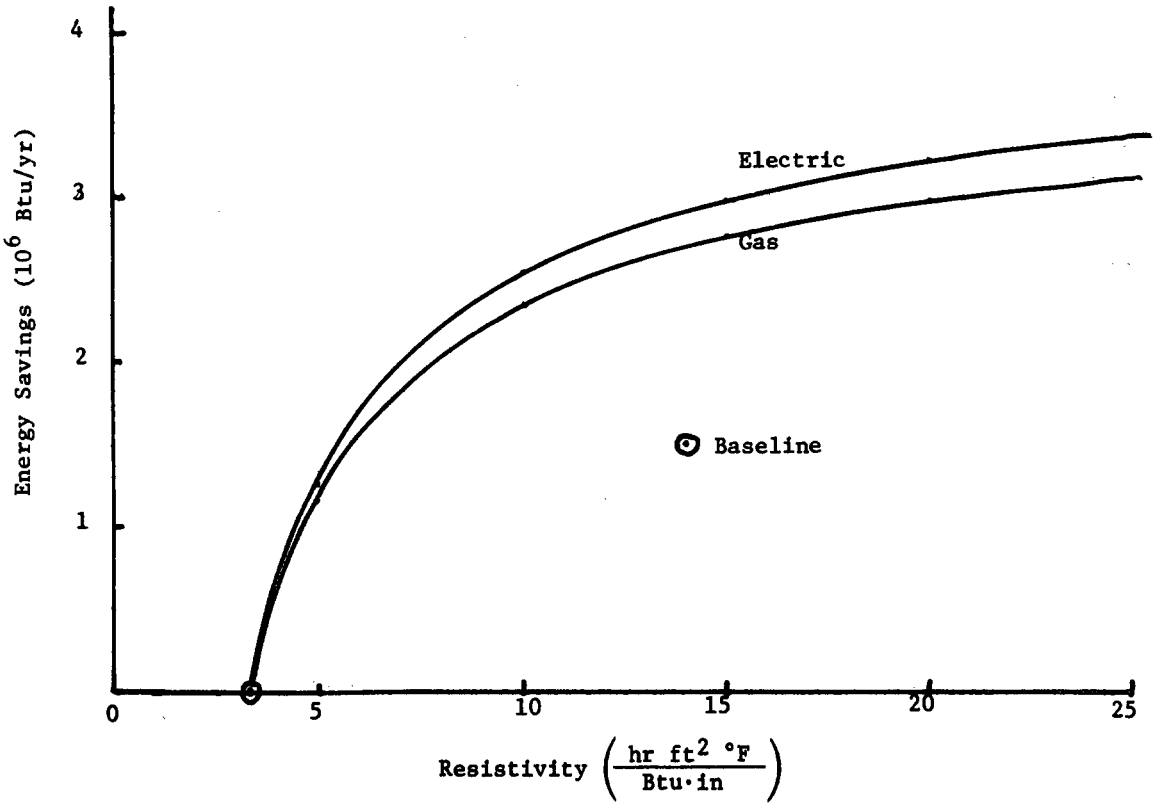

FIGURE 3.14 ENERGY SAVINGS FROM USE OF IMPROVED INSULATION IN COMMERCIAL WATER HEATERS

APPENDIX A

STATEMENT OF WORK

The objective of this program is to develop advanced insulation concepts for residential and commercial water heaters, refrigerators, freezers, and ovens. The threshold target for these insulations is a thermal resistance, for a 1-in. thickness, of 20 $hr \cdot ft^2 \cdot °F/Btu$. The program should also provide a path to commercialization of these new concepts by identifying one or more manufacturers who might be willing to participate in a demonstration program.

In order to accomplish this objective, Arthur D. Little, Inc., will complete the following tasks and subtasks for the following appliances.

Residential	Commercial
Water heaters[a]	Water heaters[a]
Refrigerator-freezer	Refrigeration systems
Freezers	Freezers
Ovens[a]	Ovens[a]

[a]Gas and electric.

Task I.1: Concept Identification

Develop a ranked listing of advanced insulation concepts for the selected appliances by performing the following subtasks.

A. Select baseline appliance (e.g., residential electric water heaters) based on their national primary energy consumption and, for each, characterize a best insulation practice for a representative model of the type responsible for the largest share of national energy consumption. If any appliance in the preceding list is excluded from the baseline set, the reason will be documented.

B. Estimate the magnitude (Btu/year/appliance) and net present value of energy savings resulting from reductions in the thermal conductivity of the insulation ranging from 2 to 10 for the representative models.

C. Develop innovative conceptual designs and estimate the cost of at least one design for each appliance model selected in Task I.1.A, utilizing advanced insulation systems with the target of achieving an insulating thermal resistance, for a 1-in. thickness, of 20 $hr \cdot ft^2 \cdot °F/Btu$. Initial design concepts will be generated by the project personnel with the assistance of at least two meetings of a concept team composed of Arthur D. Little staff to "blue sky" advanced concepts. Concepts will also be reviewed at an Industry Committee Meeting.

D. Utilizing the results of Tasks I.1.B and I.1.C, select the concepts to be evaluated and ranked in detail based on the criterion that the net present value of savings must exceed the added first cost of the advanced insulation system. Evaluate and rank these concepts utilizing a weighting method (which Arthur D. Little, Inc., and industry provide input to), e.g., manufacturability, serviceability, durability effect on appliance warrantee, safety, etc., plus estimates of (for example) national energy savings, years to pay back, RD&D funds required, etc.

E. Conduct an experimental and analytical investigation of air-filled, nonevacuated, sub-micron sized powders and fibers to determine the potential for achieving an insulation which has a thermal resistance for a 1-in. thickness of 10 to 20 $hr \cdot ft^2 \cdot °F/Btu$. Identify and select the materials that are considered appropriate, based upon engineering judgment and approved by the ORNL-TM, and develop the methods for processing the materials. At least five prototype thermal insulation samples, each 1 in. thick and 12 by 12 in. in area, will be made. One of these samples will be of the Cab-O-Sil in its "as received" state. Perform room temperature thermal conductivity measurements on these samples (as proposed, to be done by a reliable outside laboratory). Prepare at least two photomicrographs of insulation samples tested. The experimental data will be evaluated and compared with the analytical predictions. Contact firms known to be active in the areas of micropore insulations to obtain information on their processes and their likely development plans. An Industrial Review Committee, consisting of representatives from insulation and appliance manufacturers, meeting shall then be held at the Arthur D. Little, Inc., facility. The meeting purpose is to obtain review and advice on the practicality of the concepts, e.g., manufacturability, serviceability, durability effect on appliance warrantee, and safety.

   Results obtained from the experimental investigation and advice obtained from the Committee will be included in the Task I.1.F report along with conclusions and recommendations.

F. Prepare a report on the concept evaluation (Tasks I.1.A through I.1.E) arranged so that it may be used as part of the final report.

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