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Superinsulation in Refrigerators and Freezers*

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INTRODUCTION

Two separate Cooperative Research and Development Agreements (CRADAs) have addressed the issue of refrigerator insulation improvements. Both were between the Appliance Industry-Government CFC Replacement Consortium, the Appliance Research Consortium (ARC), and the Oak Ridge National Laboratory.

The first CRADA's goal was to develop a lifetime testing procedure for powder-filled evacuated panels. The results presented here were obtained during Phase IV of that CRADA, which had the specific objective of determining the lifetime of superinsulations when installed in simulated refrigerator doors. Results for the first two years of tests were reported previously [Wilkes et al, 1996], this paper includes findings from the third year of aging.

The second CRADA was established to evaluate and test design concepts proposed to significantly reduce energy consumption in a refrigerator-freezer that is representative of approximately 60% of the U. S. market. The stated goal of this CRADA is to demonstrate advanced technologies which reduce, by 50 percent, the 1993 National Appliance Energy Conservation Act (NAECA) standard energy consumption for a 20 ft^3 (570 L) top-mount, automatic-defrost, refrigerator-freezer. For a unit this size, the goal translates to an energy consumption of 1.003 kWh/d. The general objective of the research is to facilitate the introduction of efficient appliances by demonstrating design changes that can be effectively incorporated into new products. In previous work on this project, a Phase 1 prototype refrigerator-freezer achieved an energy consumption of 1.413 kWh/d [Vineyard, et al., 1995]. Following discussions with an advisory group comprised of all the major refrigerator-freezer manufacturers, several options were considered for the Phase 2 effort, one of which was cabinet heat load reductions.

These two CRADAs included measurements for the same type of evacuated superinsulation panel. This paper compares the results for experimental data collected from simulated refrigerator panel sections to those for an entire refrigerator/freezer unit.

BACKGROUND

Greenhouse gases and their damaging effects on the atmosphere have received increased attention following the release of scientific data by the United Nations Environment Programme and World Meteorological Organization that show carbon dioxide to be the main contributor to increased global warming [UNEP, 1991]. For domestic refrigerator-freezers operating on alternative refrigerants such as HFC-134a, the indirect contribution to global warming potential resulting from the amount of carbon dioxide produced by the power plant in generating electricity to operate a unit over its lifetime is approximately one hundred times greater than the direct contribution of the refrigerant alone. Moreover, approximately 62 million new units are manufactured worldwide each year and hundreds of millions are currently in use [UNEP, 1995]. It is anticipated that the production of refrigerator-freezers will substantially increase in the near future as the result of an increased demand, especially in developing countries where growth is expected to be on the order of 10 to 15 percent per year for the next few years. Recent negotiations in Kyoto also emphasized the need for a renewed effort in improving energy efficiency wherever possible. Therefore, in response to global concerns over greenhouse gases, efforts are being made to produce refrigerator-freezers with low energy consumption [Fischer et al., 1991].

In addition to the concerns of the global community over greenhouse emissions, refrigerator-freezers are also required to meet certain minimum energy-efficiency standards set up by the U. S. Congress and administered by the U.S. Department of Energy (DOE)[NAECA, 1987]. The initial standards went into effect January 1, 1990 and had one revision, in 1993, which resulted in a cumulative 40% reduction in energy consumption. In the April 1997 revision, scheduled for implementation in July 2001, the standard requires an additional 30% reduction in energy consumption. [Appliance, 1997, Federal Register, 1997].

Customer expectations and competitive pressures impose an unwritten set of constraints on refrigerator-freezers produced in the United States. The excellent characteristics of CFC-12 and its use over the past fifty years have led to highly efficient and reliable compressors and other refrigeration system components [UNEP, 1991]. Studies have shown that refrigerator-freezers give satisfactory performance for approximately 13 years on average [Appliance, 1997]. This high degree of reliability has caused consumers to expect long lifetimes and trouble-free operation from refrigerator-freezers and all appliances in general. Additionally, refrigerator-freezers have become a relatively low cost commodity item. Therefore, increased costs associated with efficiency improvements must be justified on the basis of an improved environment and lower operating cost to the consumer. Unless consumers are motivated to spend more for efficiency, further improvements will be hard for manufacturers to justify based on existing market conditions. External forces, such as rebates, new selling techniques, or standards will be required to further reduce refrigerator-freezer energy consumption from existing levels and generate markets for high-efficiency products.

EXPERIMENTAL APPROACH

Composite Panel Specimen Descriptions

Composite panels were constructed to determine the lifetime of superinsulations when installed in refrigerator doors as shown in Fig. 1. One side of the panel is a sheet of 24 gauge [0.024 inch (.061 cm) thick] mild steel that represents the outside of a refrigerator cabinet. The other side is a 0.12 inch (0.3 cm) thick sheet of acrylonitrile-butadiene-styrene (ABS) plastic that represents the inside lining. The total thickness of the panels was 2.0 inches (5.1 cm), and the lateral dimensions were 24 by 24 inches (61 by 61 cm). In the past, the space between these two sheets would be completely filled with a polyurethane foam insulation. For the superinsulation/foam composite panels, a superinsulation panel was attached to the center of the inside surface of the steel sheet using double-sided foam tape, and the remaining space was filled with polyurethane foam. The edges were sealed using aluminum tape.

Nine superinsulation panels were furnished by each of four organizations, each using a different construction, as follows: silica powder filler encapsulated in a polymer barrier film (denoted as Type A); fibrous glass insulation filler encapsulated in a stainless steel barrier (denoted as Type B); an undisclosed insulation filler encapsulated in a stainless steel barrier (denoted as Type C); and panels containing radiation baffles within a polymer barrier film, and filled with krypton gas at atmospheric pressure (denoted as Type D). All superinsulation panels were approximately ½ inch (1.3 cm) thick, and had lateral dimensions of 14 by 14 inches (35.6 by 35.6 cm) (Types A and C) or 12 by 12 inches (30.5 by 30.5 cm) (Types B and D). The Type B superinsulation panel was the same kind of evacuated panel used in the refrigerator/freezer enhancements described below. However, the superinsulation panels used in the refrigerator/freezer enhancements were about twice as thick as those used in the composite panels.

Installation of the polyurethane foam into the composite panels was performed by three foam suppliers, each of which used a different foam blowing agent for these test specimens. The three types of foam blowing agents used were CFC-11, HCFC-141b, and HCFC-142b/22 blend. This procedure resulted in a matrix of specimens with each type of superinsulation and each type of blowing agent produced in triplicates. In addition to the superinsulation/foam composite panels, similar foamed panels were fabricated without the superinsulation panels. The purpose of these foam-only panels was to provide a baseline for comparison with the panels containing superinsulations. All of the composite panels, with or without superinsulation, were stored in closed cabinets maintained at 90°F (32°C) between the thermal measurements described below.

Refrigerator Cabinet Description

A 1996, 20 ft^3 (570 L) top-mount, automatic-defrost refrigerator-freezer was selected to evaluate and test design concepts proposed to significantly reduce energy consumption in a refrigerator-freezer that is representative of approximately 60% of the U. S. market. Two cabinet designs, one a base unit and one with enhanced insulation, and four door designs were tested. The base cabinet dimensions are given in Table 1. The base cabinet's insulation had a thermal conductivity of 0.125 Btu-in./h-ft^2-°F (0.018 W/m-K) and the mullion's thermal conductivity was 0.302 Btu-in./h-ft^2-°F(0.0435 W/m-K). In addition to the standard doors, which were 1.5-inch (3.81 cm) thick, three sets of doors with varying degrees of insulation improvements were tested on the base cabinet. The three improved door designs consisted of the following: thick doors [2.5 inches (6.35 cm)], 1-inch (2.5 cm) thick vacuum insulation panels surrounded by foam in standard doors, and 1-inch (2.5 cm) thick vacuum insulation panels surrounded by foam in thick doors. The vacuum panels used in the doors were 18.5 by 25.25 in. (47 by 64 cm), covering 15% of the external surface area of the whole refrigerator/freezer.

The enhanced cabinet was constructed with four vacuum insulation panels encased in foam around the freezer section. Each panel was 1 in. (2.5 cm) thick, three of the panels were 18.5 by 25.25 in. (47 by 64 cm), and one was 23.15 by 27.15 in. (59 by 69 cm). These four panels covered 64% of the freezer compartment's external surface area, or 22% of the external surface area of the whole refrigerator/freezer. For the tests with the enhanced cabinet, only standard doors and thick doors with no vacuum insulation panels were investigated.

Composite Panel Experimental Measurements

Thermal resistance measurements for the composite panels were made using a heat flow meter apparatus (HFMA), which conforms to ASTM C 518[ASTM, 1995]. In an HFMA, a flat rectangular specimen is sandwiched between hot and cold plates that are maintained at constant temperatures. The heat flux through the specimen is measured using a heat flux transducer (HFT), which is calibrated by making measurements on a standard specimen for which the thermal resistance is known. The HFMA accepted specimens with lateral dimensions of 24 in. by 24 in. (61 by 61 cm). This HFMA had an array of 30 4-inch (10.2 cm) square HFTs on the hot side. An average of the readings from the two HFTs nearest the center of the plate [giving an average heat flux over the central 4 inch by 8 inch (10.2 by 20.3 cm) area] was used in analysis of the data reported here. The composite panels were sandwiched between two layers of foam rubber to eliminate air gaps and to protect the plates of the HFMA. Temperature sensors were attached to the surfaces of the panels.

Refrigerator Cabinet Experimental Measurements

Several tests were conducted on the baseline and enhanced refrigerators to quantify the effects on energy consumption of cabinet design changes. The testing included reverse cabinet heat loss rate measurements and 90°F (32.2°C) closed-door, energy-consumption tests as specified in section 8 of the Association of Home Appliance Manufacturers (AHAM) standard for Household Refrigerators and Household Freezers [AHAM, 1985]. The tests were performed in environmental chambers with air flows and temperature fluctuations within the specifications of the AHAM standard or according to manufacturers' recommendations for tests where no standard is specified, such as the reverse heat loss rate tests.

Reverse cabinet heat loss rate measurements were made to assess the improvements in cabinet thermal performance from changes such as vacuum insulation or increased insulation thickness in the freezer section or doors. The procedure for measuring heat loss rate involves placing a cabinet in a cold chamber with controlled heat sources and small electrical chassis fans to maintain desired temperatures in both the freezer and fresh food compartments. The fans are run continuously during the test to prevent temperature stratification. Each fan draws approximately 6-7 watts of electricity and has an air circulation rate of 30 cfm (14 L/s), which is assumed to have negligible effects on the inside-surface heat transfer of the refrigerator-freezer. Temperature and watt measurements for both refrigerator-freezer compartments along with ambient temperature are recorded as the cabinet temperatures achieve desired levels. Once the cabinet temperatures achieve steady-state, data are compiled and averaged for a thirty-minute interval to determine overall heat loss rates for both compartments.

The heat loss rate is calculated in Btu/h (W) and plotted against the difference between temperatures inside each compartment and ambient air temperature. Heat loss rates for the freezer compartment were determined from the following equation:

$$Q_{FRZ} = UA_{FRZ} \times (T_{FRZ} - T_{AMB}) + UA_{MUL} \times (T_{FRZ} - T_{FF}) \tag{1}$$

where Q_{FRZ} is the heat loss rate for the freezer in Btu/h (W), UA_{FRZ} is the overall freezer compartment thermal transmittance in Btu/h-°F (W/°C), $(T_{FRZ}-T_{AMB})$ is the temperature difference between the air in the freezer and ambient in °F (°C), UA_{MUL} is the thermal transmittance of the mullion in Btu/h-°F (W/°C), and $(T_{FRZ}-T_{FF})$ is the temperature difference between the air in the freezer and fresh food compartments in °F (°C). In a similar manner, the fresh food heat loss rate was determined from the following equation:

$$Q_{FF} = UA_{FF} \times (T_{FF} - T_{AMB}) \;-\; UA_{MUL} \times (T_{FRZ} - T_{FF}) \tag{2}$$

where Q_{FF} is the heat loss rate for the fresh food compartment in Btu/h (W), UA_{FF} is the overall fresh food compartment thermal transmittance in Btu/h-°F (W/°C), and $(T_{FRZ}-T_{AMB})$ is the temperature difference between the air in the fresh food compartment and ambient in °F (°C).

Tests were initially run with the temperatures in both compartments essentially equal. This allowed the mullion heat transfer term to be dropped from both equations (1) and (2) so that freezer and fresh food compartment transmittances could be determined by dividing the power measurement ($\dot{Q}$) by the temperature difference in each compartment $(T_{FRZ}-T_{AMB})$ or $(T_{FF}-T_{AMB})$. Once the compartment thermal transmittances were known, tests were then performed with large temperature differences between the freezer and fresh food compartments to determine the mullion thermal transmittance. Equations 1 and 2 were then used to calculate the heat loss rates in both compartments for each cabinet and door configuration.

The tests were conducted using temperature differences across the cabinet walls comparable to those attained in the 90°F (32.2°C) closed-door test procedure where the refrigerator-freezer works to maintain cold internal temperatures in a warm room. In order to achieve the temperature differences, it was necessary to maintain the chamber at 0°F (-17.8°C). Since the thermal conductivity of insulating foam generally decreases with decreasing temperatures, this procedure could slightly underestimate actual cabinet heat loss rates [ASHRAE, 1989]. In addition, the reverse cabinet heat loss measurement employed in this study may not accurately measure the heat leakage through the gasket region. Heat leakage in the gasket area is a function of the airflow inside the freezer. Since the evaporator fan was not running, the heat leakage rate might be higher than the measured values for all the tests. However, the relative differences between the test results for the different insulation configurations should be approximately the same. The procedure used in this study was chosen because it allowed a determination of heat leakage rates for both the freezer and fresh food compartments.

EXPERIMENTAL RESULTS

Superinsulation Panels

Upon receipt of the superinsulation panels, their thermal resistances were measured in the HFMA. Center-of-panel thermal resistivities [measured over the center 4 in. by 8 in. (5 by 10 cm) area] are given in Table 2. It must be strongly emphasized that these center-of-panel values do not account for any heat conduction around the edge of the panels due, for example, to high thermal conductivity stainless steel skins, and hence do not represent a thermal value for a complete panel. The values do, however, serve as an indicator of the condition of the vacuum within the evacuated insulations, or of the fill gas in the gas-filled panels. Using the data shown in Table 2, the average resistivities were 28.0, 65.2, 48.2, and 11.1 h•ft^2•°F/Btu•in. (195, 453, 334, and 77.2 m-K/W) for Types A, B, C, and D, respectively.

Composite Panels Containing Superinsulations

Thermal measurements were performed on 36 composite panels that contained superinsulations. Measurements were made on all 36 panels at 0, 6, and 12 months of aging, on 12 panels at 24 months, and on all 36 panels at 36 months. Raw data on the actual test panels were analyzed using a computer model to normalize them for differences in the sizes of the superinsulation panels and to estimate the total effective thermal resistance of panels that would likely be used in refrigerators. The computer model was a three-dimensional finite-difference heat conduction model based on the HEATING code[Childs, 1993]. Analysis of the data consisted of two steps.

For the first step of analysis, a model was set up that included the composite panel as well as the foam rubber sheets that were laid between the panel and the plates of the HFMA.

Boundary conditions for the model consisted of the temperatures measured on the plates. Handbook values for thermal conductivities of several of the materials were used, viz., 480, 1.8, and 96 Btu•in./h•ft^2•°F (69, 0.26, and 14 W/m-K) for the steel sheet, the ABS plastic sheet, and the stainless steel superinsulation cladding, respectively. A measured value of 0.7 Btu•in./h•ft^2•°F (0.1 W/m-K) was used for the foam rubber sheets. The value used for the polyurethane foam insulation was the average value measured on the foam-only composite panels at each time period for each blowing agent. The thermal conductivity of the superinsulation was treated as the only unknown quantity. The thermal conductivity of the superinsulation was systematically varied in the calculations until the calculated heat flux over the central 4 by 8-in. (10 by 20-cm) area matched the value measured by the heat flux transducers..

With values for the thermal conductivity of each of the materials, another computer model was used to estimate the overall thermal resistance of composite panels of various sizes in which the superinsulation covered 60 percent of the total area. For this model, the steel and plastic boundary sheets were taken to be exposed to air with a heat transfer coefficient of 1.0 Btu/h•ft^2•°F (5.7 W/m^2-K). This arbitrary value was chosen because it was reasonable for natural convection and because this boundary condition allowed the surface temperature of the composite panel to vary. Overall thermal resistances obtained by this procedure are given in Table 3 and Figs 2 and 3. These figures show the average thermal resistance for each triplicate set of 24-inch (61-cm) panels with CFC-11 foam insulation. The results for the foam-only panels (with the same boundary sheets and air heat transfer coefficients) are included for reference. The type B superinsulation with CFC-11 is the most comparable to the panels used in the refrigerator system tests. These superinsulation panels were made by the same manufacturer as those used in the refrigerator/freezer modifications. Keep in mind, however, that the superinsulation panels in these specimens were only one-half as thick as those used in the refrigerator/freezer modifications. Although CFC-11 was not used to blow the insulating foam within the refrigerator/freezer, it represented a mature foaming technique when the composite panels were made, as opposed to the other foaming agents that were at that time experimental and not yet optimized.

The overall thermal resistance depends upon four factors in this study: the type of superinsulation, the blowing agent for the foam insulation, the aging time, and the size of the simulated panels. The effect of panel size is only significant for those superinsulations that have stainless steel claddings [Wilkes, et al, 1996]. As was found for the foam-only composite panels, the overall resistances with superinsulation show a very slow decrease of resistance with time. During the three-year period, the foam-only panels decreased in thermal resistance from 4 to 7%. Considering the averages of the triplicate sets of data at the 24-inch (61-cm) panel size, the composite panel resistance changes over the three-year period range from an increase of 0.2 percent to a decrease of 9.0 percent. The thermal resistivity of the superinsulation panels themselves is estimated as described above and the average for each type of panel is shown in Figs. 4 and 5. Although none of the thermal resistance changes are large, the panels with stainless steel barriers (types B and C) appear to show less degradation than those with the polymer barrier films (type A and D).

Averaging all the results for each type of superinsulation after three years of aging gives average composite panel thermal resistances of 20.1, 19.8, 20.2, and 16.9 h•ft^2•°F/Btu (3.54, 3.49, 3.56, and 2.98 m^2K/W) for panels with Types A, B, C, and D superinsulation. Thus panels with Types A, B, and C superinsulations are similar, while the panels with Type D superinsulation have resistances that are about 15 percent lower than the other three types. This is in agreement with the center-of-panel results on the original superinsulation panels, where the average thermal resistivity of Type D panels was 40 percent less than those of Type A panels. The thermal resistance of composite panels with Types A, B, and C superinsulations are remarkably similar, even though the center-of-panel thermal resistivities were greatly different. The higher center-of-panel thermal resistivities for Type B and C superinsulations were offset by heat conduction through their stainless steel encapsulation material.

Looking at Fig. 2, the Type B panel (which incorporates a ½-inch thick superinsulation panel covering 60% of a 24 by 24 inch space) offers about 18% more thermal resistance than the foam-only panel.

Refrigerator Cabinet Modifications

Cabinet heat loss rates for the baseline cabinet with the standard doors and door insulation improvements are shown in Table 4. The heat loss rates are determined from equations 1 and 2 using compartment and mullion thermal transmittances calculated from measurements made under steady-state conditions. Table 4 also shows the cabinet heat loss results for the enhanced cabinet with the standard and thick doors. The experimental results indicate that the baseline cabinet heat loss rate was reduced 6.4% by replacing the standard doors with thick doors. Using 1-inch (2.5 cm) thick vacuum panels surrounded by foam in a standard door (thereby covering 15% of the total exterior surface area) resulted in the cabinet heat loss rate being reduced by 11.0%. Finally, when 1-inch (2.5 cm) thick vacuum panels were encased in foam in a thick door, the cabinet heat loss rate was reduced by 12.3%.

For the enhanced cabinet, vacuum panels surrounded by foam around the entire freezer section resulted in an overall cabinet heat loss rate 15.0% lower than the baseline cabinet. Remember that these vacuum panels were 1-inch (2.5 cm) thick and covered only 21.5% of the cabinet's exterior surface. Tests were also performed with thick doors on the enhanced cabinet resulting in a 20.4% reduction in the overall cabinet heat loss rate.

Direct comparisons between the composite panel and refrigerator cabinet measurements are not possible because the vacuum panels were of different thicknesses. However, the model used to calculate the overall composite panel thermal resistivities shown in Table 3 was also used to estimate the effect of the increased vacuum panel thickness. These results were then fed into a widely-distributed refrigerator model to calculate cabinet heat gain values appropriate for comparison to the refrigerator cabinet reverse heat loss measurements[EPA, 1993]. The refrigerator model results are included in Table 4 and are within 6% of the measured values for the configurations without superinsulation panels. For the superinsulation cases, the model over predicts energy use by 12 to 20%. More rigorous modeling efforts are planned.

SUMMARY

Small composite panels that contain superinsulations along with polyurethane foam are being used to address questions related to long-term reliability and heat transfer degradation. It was demonstrated that both gas-filled and vacuum superinsulations can withstand the processes necessary to fabricate refrigerator/freezer walls and doors, including the foaming of polyurethane insulation around the superinsulations. The overall range of resistance for the 2-in (5-cm) thick composite panels was from 16 to 23 h•ft^2•°F/Btu (2.9 to 4.2 m^2•K/W). Composite panels incorporating stainless-steel evacuated panels were measured over a three-year time period and showed less than a 5% reduction in overall thermal resistance. Similar results on composite panels without superinsulation showed thermal resistance reductions of about 6%. These small changes with time indicate that the bounding surfaces of the simulated refrigerator walls or doors hinder the movement of air into and the blowing agent out of the cells of the foam. Longer experimental time periods are still needed and are in progress.

The refrigerator system enhancements evaluation has shown that it is technically feasible to significantly reduce cabinet heat gain using vacuum panel insulation. Although the costs of vacuum panels are still of concern, the new regulatory requirements for improved efficiency may necessitate their use. In addition to its energy-saving potential, vacuum panel insulation may be attractive for refrigerator-freezers because it can augment the food storage volume by reducing the insulation volume in areas where it is thickest, such as the doors.

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Table 1. Base Refrigerator Cabinet Dimensions

	(in.)	(cm)
Height	61.2	156
Width	32.8	83.2
Depth (including door and gasket)	28.9	73.4
Gasket Thickness	0.75	1.91
Gasket Width	1.63	4.13
Door Edge Thickness	1.25	3.17
Wedge Depth	3.0	7.62
Wedge Flange Width	1.64	4.17
Compressor Compartment		
Top Depth	5.0	12.8
Bottom Depth	11.2	28.4
Height	5.14	13.1
Freezer Compartment Insulation Thickness		
Top Wall	2.9	7.3
Side Wall	2.9	7.3
Back Wall	3.0	7.62
Door	1.5	3.81
Fresh Food Compartment Insulation Thickness		
Side Wall	1.9	4.83
Back Wall	2.5	6.35
Bottom Wall	1.75	4.44
Door	1.5	3.81
Mullion		
Distance to Top	18.2	46.2
Thickness	1.5	3.81

Table 2. Center-of Panel Thermal Resistivities of Superinsulation Panels

Panel	Resis-tivity	Panel	Resis-tivity	Panel	Resis-tivity	Panel	Resis-tivity
A-1	26.7	B-1	68.3	C-1	51.1	D-1	11.0
A-2	27.9	B-2	71.3	C-2	47.3	D-2	11.1
A-3	29.3	B-3	57.6	C-3	46.5	D-3	11.4
A-4	26.8	B-4	54.6	C-4	52.5	D-4	11.4
A-5	27.9	B-5	64.8	C-5	46.2	D-5	11.0
A-6	29.4	B-6	71.6	C-6	47.2	D-6	11.0
A-7	28.1	B-7	69.1	C-7	48.5	D-7	10.9
A-8	29.0	B-8	71.1	C-8	47.8	D-8	11.2
A-9	27.0	B-9	58.7	C-9	46.6	D-9	11.4

Note: Thermal resistivities measured before installation into composite panels. Thermal resistivities have units of h•ft^2•°F/Btu•in. Multiply by 6.933 to obtain units of m-K/W.

Table 3. Assembly R-values for 24 by 24 inch by 2 inch composite panels containing superinsulating vacuum panels. R-values have units of h-ft^2-°F/Btu.

Panel Type	Blowing Agent	Superinsulation Panel ID	Initial	6 months	12 months	24 months	36 months
A	CFC-11	MTA-1	20.2	23.0	22.5	-	20.9
A	CFC-11	MTA-4	23.1	23.1	23.0	-	21.2
A	CFC-11	MTA-8	23.0	22.5	22.3	22.3	21.1
A	HCFC-142b/22	MTA-03	21.1	20.3	20.0	21.0	20.1
A	HCFC-142b/22	MTA-06	22.2	19.8	20.7	-	20.1
A	HCFC-142b/22	MTA-13	20.8	20.7	20.4	-	19.6
A	HCFC-141b	MTA-11	21.1	20.5	21.1	-	19.4
A	HCFC-141b	MTA-9	20.8	20.7	20.4	-	19.1
A	HCFC-141b	MTA-15	21.1	20.6	20.8	19.9	18.9
B	CFC-11	6007-014-02	21.3	21.9	21.6	-	20.9
B	CFC-11	6007-029-02	21.9	21.0	21.1	20.9	21.3
B	CFC-11	6007-030-03	19.9	20.5	19.6	-	18.8
B	HCFC-142b/22	6007-029-03	20.6	20.3	20.3	20.3	19.7
B	HCFC-142b/22	6007-012-02	21.1	21.0	20.9	-	20.2
B	HCFC-142b/22	6007-014-01	20.1	19.3	20.2	-	20.0
B	HCFC-141b	6007-014-03	19.5	19.0	19.4	-	18.2
B	HCFC-141b	6007-030-02	20.8	19.7	20.6	-	19.4
B	HCFC-141b	6007-013-03	20.9	19.5	20.3	19.6	19.6
C	CFC-11	123	22.5	21.4	22.2	-	21.3
C	CFC-11	128	21.5	22.3	22.3	-	21.1
C	CFC-11	121	22.3	20.7	21.2	21.7	20.8
C	HCFC-142b/22	132	20.6	20.3	20.8	-	20.1
C	HCFC-142b/22	122	19.9	19.6	20.3	20.8	20.2
C	HCFC-142b/22	131	19.7	20.6	20.5	-	20.1

Table 3. (cont.)

Panel Type	Blowing Agent	Superinsulation Panel ID	Initial	6 months	12 months	24 months	36 months
C	HCFC-141b	129	20.6	19.9	20.3	19.7	20.2
C	HCFC-141b	127	20.5	19.4	20.6	-	19.2
C	HCFC-141b	124	20.2	19.7	20.0	-	19.1
D	CFC-11	177	19.3	18.7	18.8	-	18.1
D	CFC-11	163	19.2	18.3	18.9	-	16.8
D	CFC-11	170	19.1	18.9	18.8	18.8	18.0
D	HCFC-142b/22	173	17.4	17.0	17.1	-	16.6
D	HCFC-142b/22	176	17.7	17.1	17.3	-	16.9
D	HCFC-142b/22	180	18.1	17.3	17.1	18.0	17.4
D	HCFC-141b	166	18.3	17.5	17.8	-	16.6
D	HCFC-141b	167	17.2	16.8	16.3	17.0	15.4
D	HCFC-141b	172	17.5	17.1	17.2	-	16.0

Table 4. Summary of Reverse Heat Loss Tests (90° F Ambient, 5° F Freezer, 45° F Fresh Food Compartment) and Comparable Model Results

	Reverse heat loss tests				Model	
Description	$Q_{freezer}$ [Btu/hr(W)]	$Q_{fresh\ food}$ [Btu/hr(W)]	Q_{total} [Btu/hr(W)]	Savings (%)	Q_{total} [Btu/hr(W)]	Savings (%)
Base Cabinet						
thin doors	103. (30.3)	91.8 (26.9)	195. (57.2)	----	208. (60.8)	
thick doors	94.8 (27.8)	87.9 (25.8)	183. (53.5)	6.4	193. (56.5)	7.4
vacuum panels in thin doors	95.1 (27.9)	78.6 (23.0)	174. (50.9)	11.0	195. (57.1)	6.4
vacuum panels in thick doors	98.4 (27.8)	72.8 (21.3)	171. (50.2)	12.3	186. (54.4)	10.9
Enhanced Cabinet (Vacuum Panels Around Freezer Section)						
thin doors	86.4 (25.3)	79.5 (23.3)	166. (48.6)	15.0	198. (58.1)	4.8
thick doors	80.3 (23.5)	75.0 (22.0)	155. (45.5)	20.4	183. (53.6)	12.2
vacuum panels in thin doors					185. (54.2)	11.2

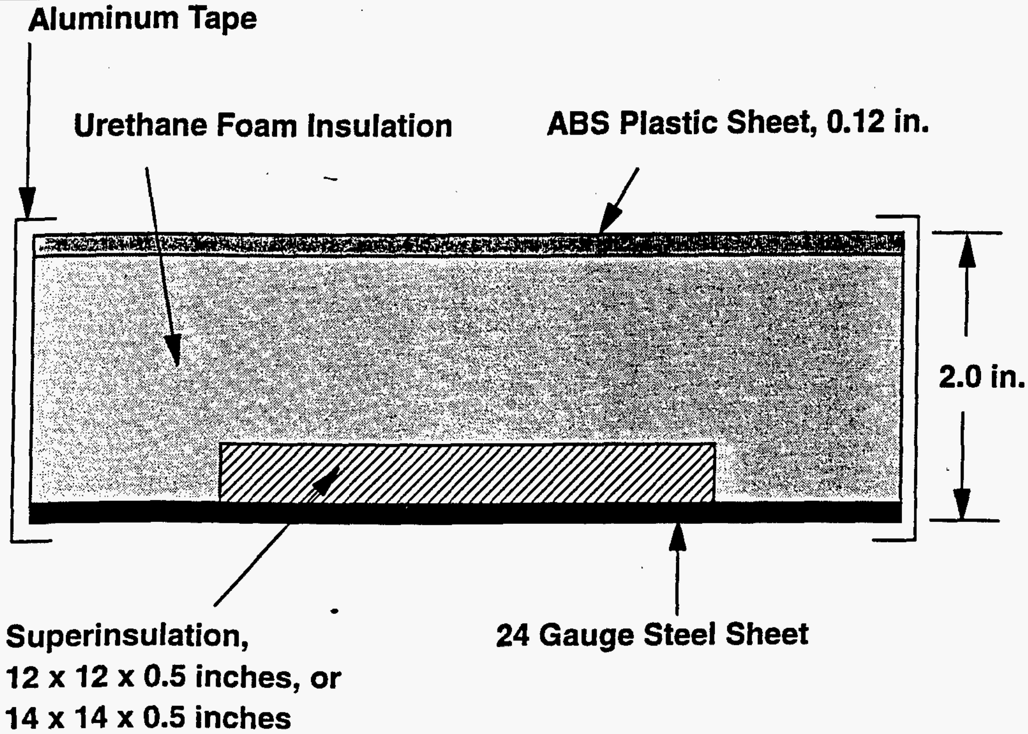

Figure 1. Construction of Composite Panels

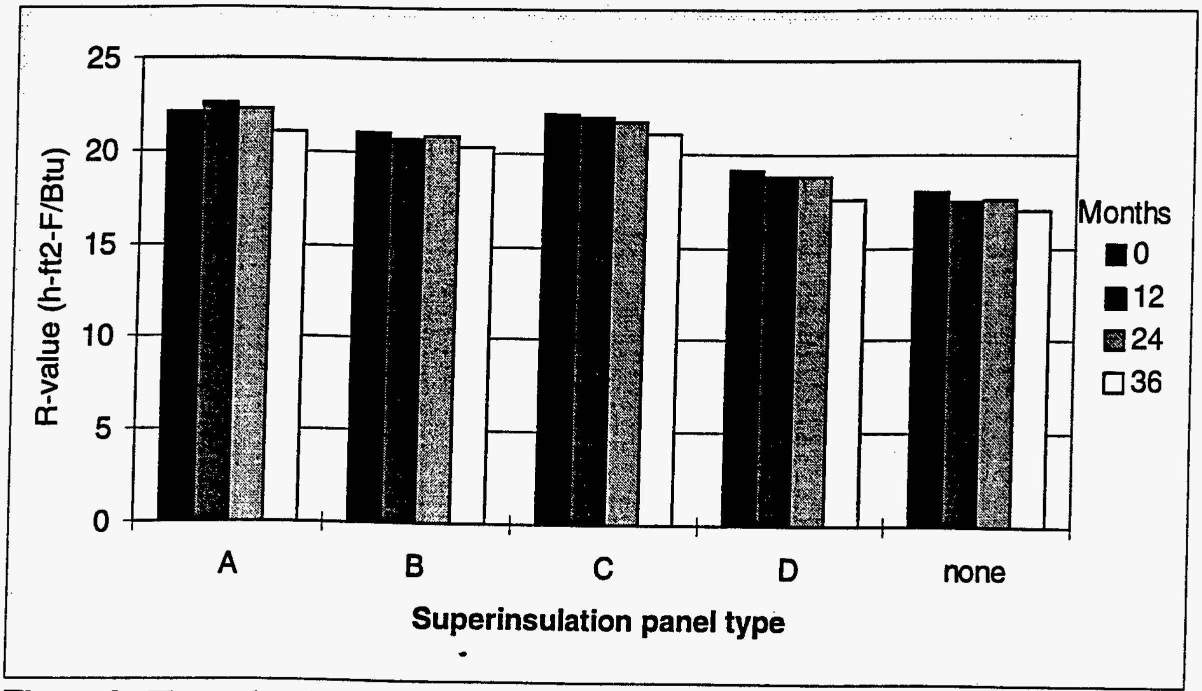

Figure 2. Thermal resistance for 24 by 24 by 2 inch panels incorporating ½ in. thick superinsulation panels encased within CFC-11-blown foam over a 36 month period.

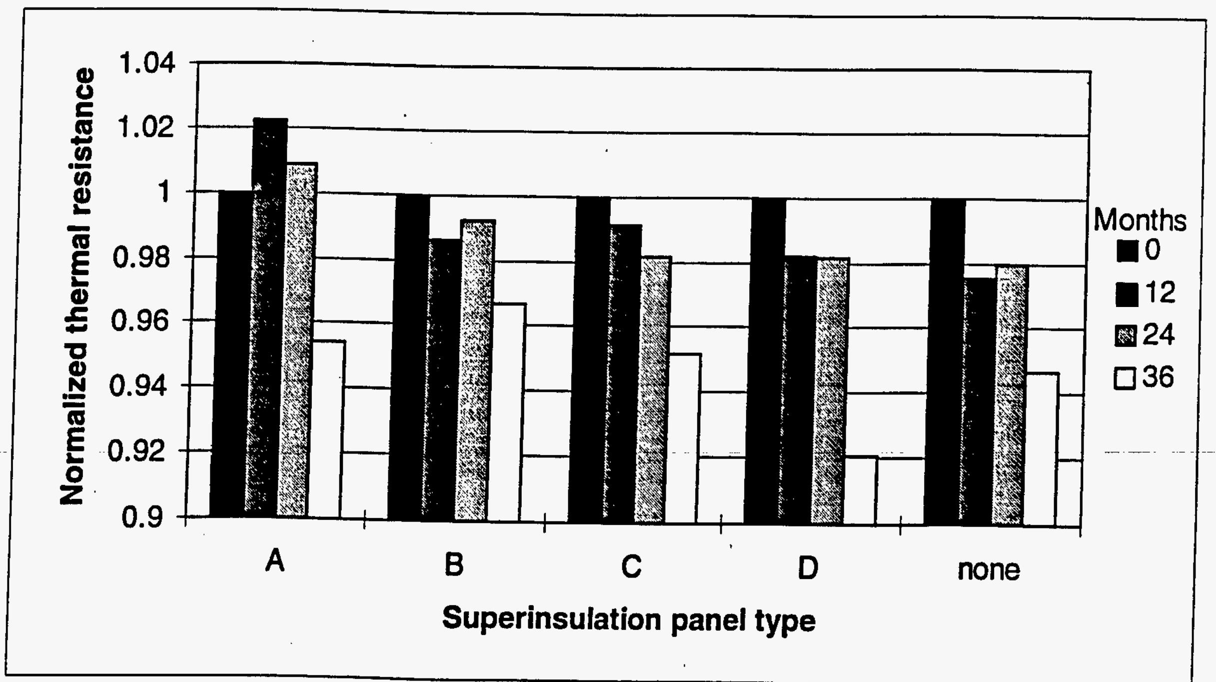

Figure 3. Thermal resistance for 24 by 24 by 2 inch panels incorporating ½ in. thick superinsulation panels encased within CFC-11-blown foam over a 36 month period, normalized relative to the initial value.

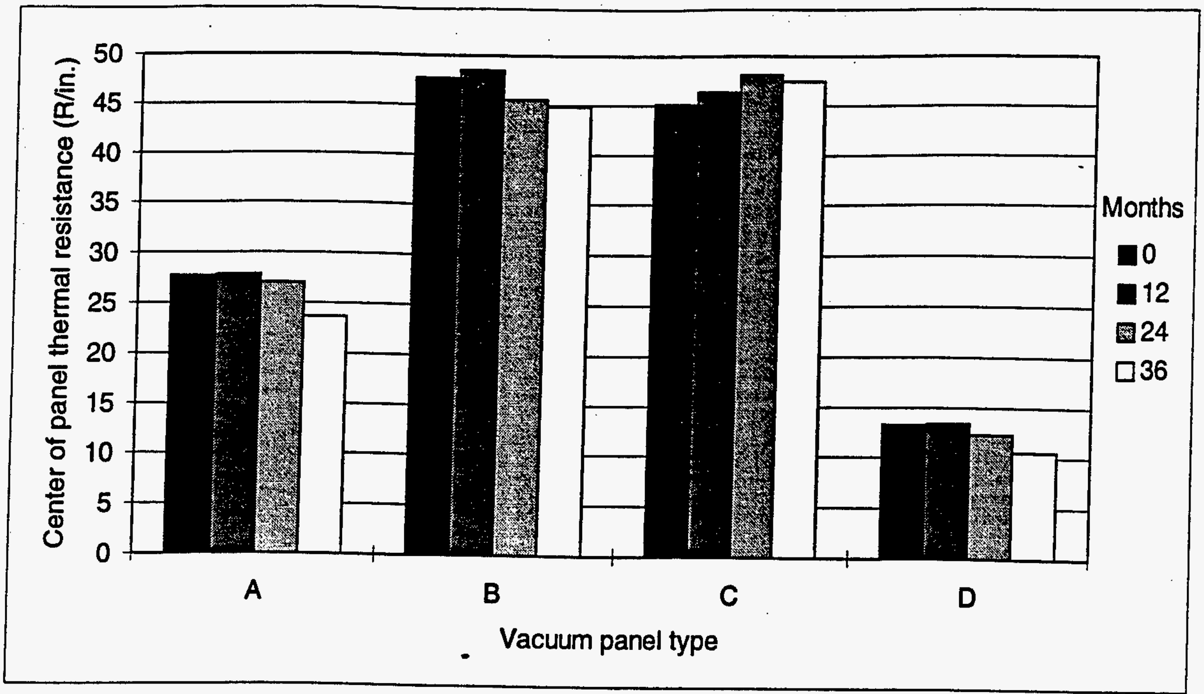

Figure 4. Calculated superinsulation center-of-panel thermal resistance (i.e. thermal resistance in the absence of edge effects) based on composite panel measurements over a 36 month period.

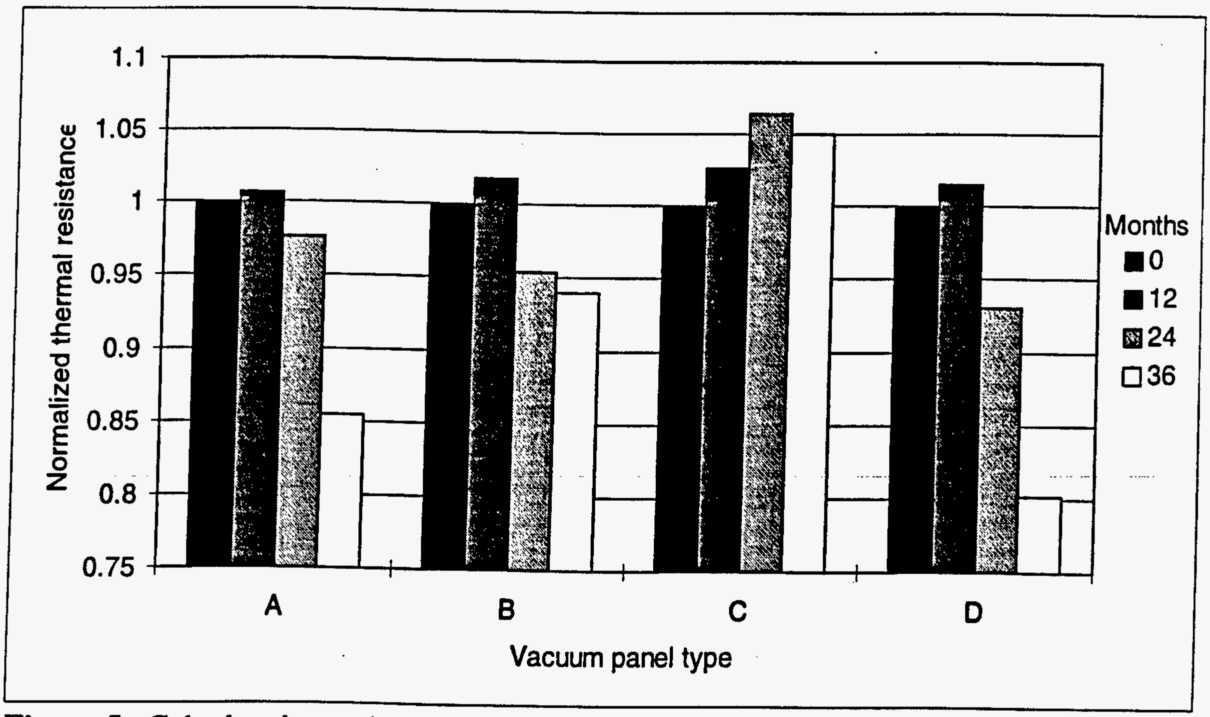

Figure 5. Calculated superinsulation center-of-panel thermal resistance (i.e. thermal resistance in the absence of edge effects) based on composite panel measurements over a 36 month period, normalized relative to the initial value.

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