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DEVELOPMENT OF A HIGH EFFICIENCY, AUTOMATIC DEFROSTING REFRIGERATOR/FREEZER

PHASE I — DESIGN AND DEVELOPMENT

FINAL REPORT

VOLUME II — R&D TASK REPORTS

Prepared by
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ARTHUR D. LITTLE, INC.
Acorn Park
Cambridge, Massachusetts 02140

February 1980

Work performed for

OAK RIDGE NATIONAL LABORATORY

Operated by

UNION CARBIDE CORPORATION

for the

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PREFACE

This report, prepared by Arthur D. Little, Inc. presents a study of technological improvements in refrigerator/freezer design. The report consists of two volumes.

Volume I, the Executive Summary, presents a summary of the technical improvements considered and a review of the important conclusions and recommendations. Volume II provides a comprehensive discussion of the results of the three major tasks in Phase I.

Task 2 Development of Improvement Target

Task 3 Prototype Design and Testing

Task 4 Field Demonstration Plan

ABSTRACT

Eighteen energy-saving design options were identified for the automatic defrost refrigerator/freezer unit. Projected energy savings and likely consumer acceptance of the design options were evaluated and seven promising options were selected for the development phase.

Computer and laboratory studies of: an improved condenser and evaporator design, new air flow path and fan housing design, improved defrost and refrigeration expansion valve control, and optimized cabinet insulation were performed. A prototype 16-cubic-foot automatic defrost refrigerator/freezer combining the seven energy saving design options were designed, built, and tested at Amana Refrigeration, Inc.

The Phase I prototype refrigerator/freezer had a 1.8 kwh per day energy consumption under the standard 90°F closed door energy test. This is an energy factor of over 10 cubic feet per kwh per day and it represents better than a 50% improvement in unit efficiency over the most efficient unit presently available.

A field test and market assessment (Phase II) is outlined. The test is designed to evaluate the unit performance in actual home use and marketability in a retail environment.

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TASK 2 REPORT

TARGET REFRIGERATOR DESIGN AND IMPROVEMENTS

Arthur D. Little, Inc.

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TARGET REFRIGERATOR DESIGN AND IMPROVEMENTS

INTRODUCTION

This task report summarizes: the analysis of food storage trends and their anticipated effect on refrigerator design, and energy-saving design features. These two sections are the design guides for the prototype development and testing task to follow. The first section outlines the feature desired in the refrigerator-freezer for the 1980-1985 market, and the second section outlines energy-saving improvements for this unit. A rating scheme was developed so that the best energy-saving options could be selected.

1. TARGET MARKET

The average size of a refrigerator purchased for the typical U.S. household depends in a large measure on the factors below:

1) The amount of food consumed by each individual in the household, discussed below in Section 1.1, and the number of people per household as discussed in Section 1.2.

2) The fraction of the food requiring refrigeration or freezing versus the amount of canned or otherwise shelf-stable processed food, discussed in Section 1.1

3) The timing of food purchases and the duration of storage, discussed in Section 1.3.

4) Trends in the selection of cabinet designs - side by side versus top/mount freezer.

1.1 Per Capita Food Consumption

An analysis of historical food consumption patterns (Table 1) shows nearly constant per capita levels with the large increase in soft drink and beer consumption offset by a decline in milk consumption. The dramatic decline in the consumption of milk has continued through the early 1970's and appears to be continuing, which may be tied to the declining birthrate. The birthrate trend may reverse itself in the next ten years, leading to a 20% increase in young children by 1985.

TABLE 1

PER CAPITA CONSUMPTION (In Pounds)
FRESH, FROZEN, AND CANNED FOODS

	MEAT, POULTRY AND FISH		DAIRY PRODUCTS				VEGETABLES			FRUIT AND JUICE				
Year	Fresh and Frozen	Canned	Milk and Cream (Fresh)	Butter and Cheese	Sherbert Ice Cream Ice Milk	Canned and Dry Milk	Fresh (Refrigerated)	Frozen	Canned	Fresh (Refrigerated)*	Frozen	Canned	Soft Drinks***	Beer***
1960	179.7	14.8	322	20.6	24.3	25.5	138.5	9.7	44.7	100.4	12.6	35.5	(122)	(120)
1965	187.8	15.6	302	20.7	26.6	22.7	130.1	13.8	48.7	88.8	12.2	34.4	169	134
1970	209.7	18.6	264	22.6	27.1	17.3	129.3	20.8	52.9	91.8	13.2	38.0	225	154
1975	206.3	16.2	248	24.0	27.8	14.6	131.1	23.7	55.5	89.7	16.2	33.7	262	180
1985	(205.0)	No est.	(275)**	25.0	(28.0)	No est.	(132.0)	(24.0)	No est.	(90.0)	(16.0)	No est.	(380)	(234)

*Includes chilled juices

**Reflects a 20% increase in 1-5 year olds in population representing about 1/2 of the milk consumption

Source: U.S. Department of Agriculture, 1976 Food Consumption Data and ADL estimates

***Source: Beverage Industry Annual Manual 1975-76, 1976-77
Straight Line Projection to 1985 see Appendix A

The recent approval of the retortable pouch by the Federal Drug Administration renewed speculation that it may eventually affect the packaging and storage of food. The retortable pouch has many advantages over the conventional three-piece can. The essence of its advantage is that the flexible pouch has a greater surface-to-volume ratio than the conventional three-piece can. Consequently, there is faster transfer of the heat necessary to commercially sterilize the pouched product. Apart from the savings in energy, it is claimed that pouched products exhibit improved color, flavor, and nutrient retention over their conventionally canned counterparts.

The retortable pouch presently consists of a three-layered film system: a heat sealable inner polyolefin liner, a middle aluminum foil layer for gas, moisture, and light impermeability, and an outer polyester layer for strength, scuff resistance, and printability. Presently designed equipment can form, fill, and seal between 30 and 60 pouches per minute. When one considers that conventional canning lines run at 200 to 400 cans per minute, this represents a serious limitation, even if several machines are set in parallel.

There remains considerable room for technical improvements in such areas as seal integrity, i.e., a reduction in the present rate of seal failures. We do not believe, however, that the retortable pouch will have a substantial impact on refrigerated and frozen food consumption in the next ten years. Presently, the consumer perceives fresh, refrigerated, and frozen foods to be more desirable than heat processed foods. The flexible pouch packaging will make a substantial impact on conventionally canned foods, but no reduction is anticipated in the consumption of refrigerated and frozen foods through 1990 as a result of the retortable pouch.

1.2 Household Size--Target Market

Although the per capita consumption rates and the types of food consumed are increasing, the average size of the household is declining. This decline is shown in Table 2, "Forecasts of Average Household Size." The projected decline in the number of persons per household is a result of two factors: the increasing number of older people who are living alone, and the increase in singles who postpone marriage. The average household is expected to consist of two to three persons in 1985, and this represents the target market for the 1985 high energy efficiency refrigerator-freezer.

These factors combined are expected to contribute to a decline of approximately 10% in household size over the next decade, leading to the projected household food consumption pattern shown in Table 3, showing a modest 3.5% increase in household refrigerated food consumption between 1975 and 1985. Included in this table is an estimate of the portion of the food eaten at home.

TABLE 2

FORECASTS OF AVERAGE HOUSEHOLD SIZE

Year	Average Number of Persons per Household	
1955	3.33	
1960	3.33	
1965	3.29	
1970	3.14	
1975	2.92	
1980	2.73	Target Household Size
1985	2.60	
1990	2.50	

Source: ADL estimates and U.S. Bureau of Census Current Population Reports, Series P-25

TABLE 3

TREND IN REFRIGERATED FOOD CONSUMPTION*
AND STORAGE VOLUME
(Based on Tables 1 and 2)

Year	Average Number of Persons per Household	% of Food Eaten at Home+	Refrigerated and Frozen		Sales Weighted Average Cubic Feet
			Lbs./Yr.-Person	Lbs./Yr.-Household	
1960	3.33	86.5	1049	3018.8	12.2
1965	3.29	85.3	1085	3045	13.6
1970	3.14	84	1158	3049	15.3
1975	2.92	83	1209	2929	15.7
Projected 1985	(2.60)	(82)	(1408)	(3001)	see Section 1.4

*Perishable food only--canned and dry foods not included

+Source: National Food Situation Nov. 1976

1.3 Nature of Household

More families will have both husband and wife in the work force, and this is expected to affect the household's food buying habits and their food storage needs. At the present time, of all husband-wife families, 50% are two-wage-earner households, and one-half of all mothers with children aged 6-17 years are working. Two-wage-earner households will approach 60% during the 1980's.

In 1976, the average number of food trips made per week was three.* (Note that it cannot be assumed that similar foods are purchased on each of the three trips, e.g., greater amounts of refrigerated and frozen foods may be purchased on one of the trips.)

There is reason to believe that this number will be lower in the future. A study by the Bureau of Advertising** revealed that working women are more likely than non-working women to shop for food only once a week increasing the storage volume requirements. Increasing use of microwave ovens, which would appeal to the same group of consumers (i.e., families with two wage earners and working mothers) will facilitate the use of frozen foods, and probably further increase demand for larger freezers in refrigerators--or separate freezers.

There are also some trends that indicate consumers will be more apt to select refrigerators that meet their needs.

Recent trends in household occupancy characteristics indicate increases in homeownership rather than renting. In 1975, 85% of single-family units were owner-occupied, whereas in 1970 the proportion was 81%. In the future, not much change is foreseen in the housing inventory characteristics in the 1975 to 1985 period (see Table 4). Basically, in the future, new construction will probably be dominated by single-family (detached and attached) with owner occupancy rates increasing by about 5% in each category of housing by 1985. This is due to the following factors:

- age structure of the population;
- increasing households with double incomes;
- real estate ownership viewed as a good investment versus renting; and
- tax advantages of ownership.

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*Frankel, M., What Do We Know About Consumer Behavior?, National Science Foundation, 1976.

**Working Women's Food Buying Habits Revealed, EDITOR AND PUBLISHER, September 30, 1972.

TABLE 4

HOUSEHOLD BY TYPE OF OCCUPANCY
('000's)

Housing Type	1975*			1985**		
	Total	Owner-Occupied	Renter-Occupied	Total	Owner-Occupied	Renter-Occupied
Single Family	46,869	39,787	7,082	54,392	48,953	5,439
Low Density	11,794	3,672	8,122	12,766	4,468	8,298
Multi-Family	10,517	585	9,932	12,458	1,246	11,212
Low Rise	7,319	450	6,869	8,970	958	8,012
High Rise	3,198	135	3,063	3,488	288	3,200
Mobile Homes	3,341	2,822	519	5,039	4,535	504
TOTAL	72,521	46,866	25,655	84,655	59,202	25,453

Sources:
*U.S. Department of Commerce and Bureau of Census estimates
**Arthur D. Little, Inc., estimates

Since homeowners are more likely to buy their refrigerators than renters, it can be expected that the refrigerators selected will be more closely matched to their needs.

Historically, family incomes have increased by about 25% with the addition of a second earner. However, in the future the added income should be greater than the historical rate as women increasingly develop a career orientation and maintain careers into marriage.

1.4 Composite of Factors Affecting Refrigerator Size

Purchased refrigerator sizes have mirrored trends in annual household refrigerated food consumption as shown in Figure 1. Factors which may have accounted for the steep rise in refrigerator size in the 1960-1970 period are:

1) Reduced number of shopping trips arising from increased number of working married women, (presently 50% of all married women) leading to larger storage requirements; and increased food storage times, particularly for storage of leftovers and foods to be reheated later arising from food packaging to meet the needs of a family larger than the average 2-3 person household.

2) The growing share of market captured by side-by-side units (shown in Figure 1) which are not offered in sizes below about 18 cubic feet. The side-by-side is a feature promoted by portraying it as two separate units (a freezer and a refrigerator) conventiently located near one another.

But in the near future we anticipate a slower growth in these trends toward increased refrigerator size, leading to a return to the growth rate of unit sizes evidenced in the late sixties (as shown in Figure 1).

1) The number of married working women having risen to 50% from 25% in 16 years will grow slowly to 60% in the 1980's.

2) The market share of side by side purchases may not increase markedly as consumers consciously consider energy consumption in their purchase decisions as evidenced in the findings of a study discussed below.

In a recent study, Labeling and Consumer Information Programs for Refrigerator-Freezers,* a sample of 104 subjects about to purchase refrigerators were divided into three groups as follows:

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*Worrall, J., Labeling and Consumer Information Program for Refrigerator-Freezers, Human Science Research, Inc., McLean, Virginia 22101.

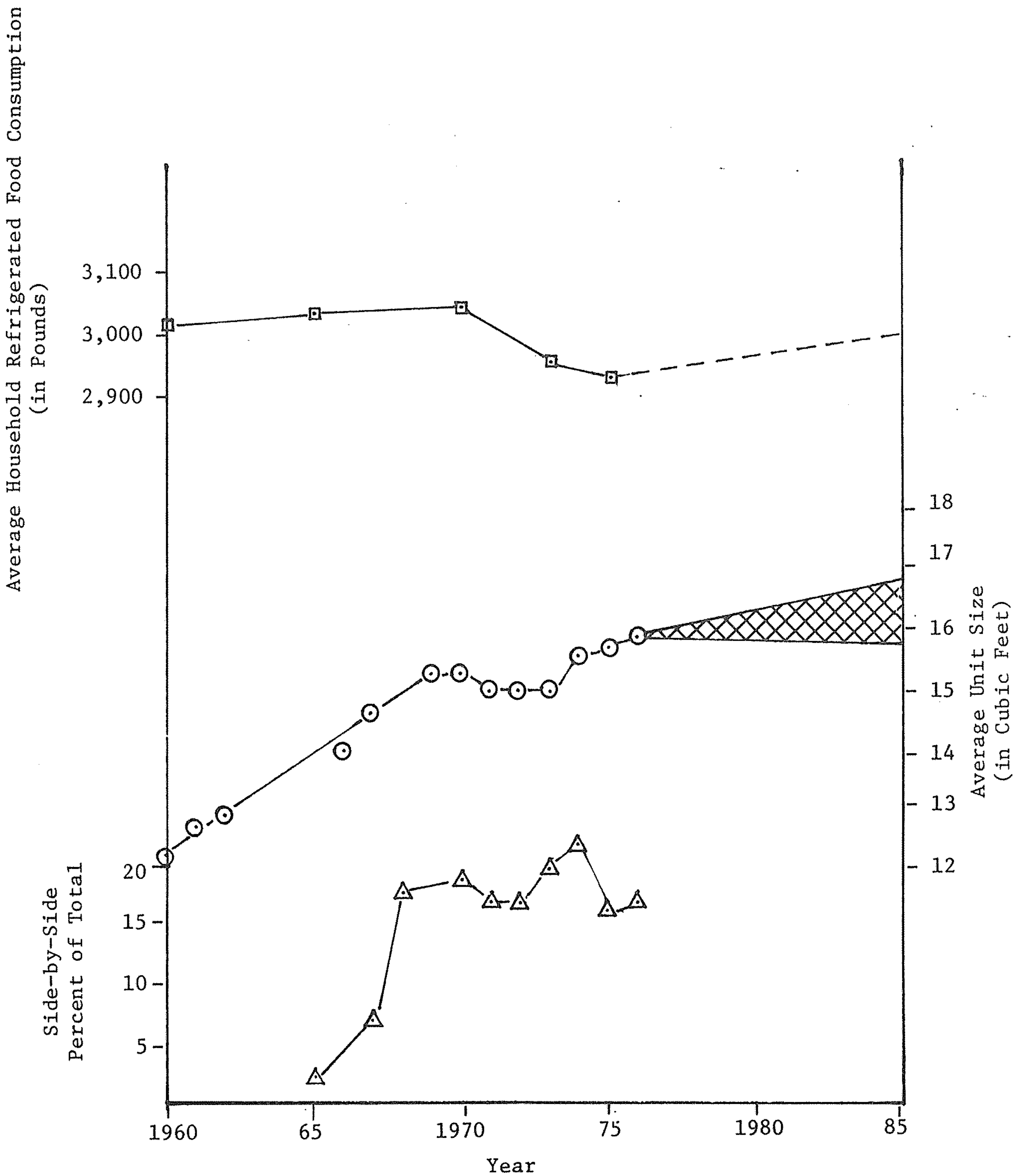

FIGURE 1 COMPOSITE FACTORS AFFECTING REFRIGERATOR-FREEZER SIZE

1) Subjects which received no energy-related information--the baseline condition.

2) Subjects receiving energy guide labels for models in which they expressed an interest.

3) Subjects receiving both labels and supplementary information dealing with the selection of energy efficient refrigerator-freezers.

The outcome of the experiment was highly significant, as shown in the following table.

Subject	Average Size Purchased	Average Unit Cost	Average unit kwh/Month	kwh/day
A - No energy information	19.2 Cubic Foot	$481	159.7	5.250
B - Labels only	18.8 Cubic Foot	$471	161.0	5.293
C - Labels plus supporting information	17.4 Cubic Foot	$441	135.5	4.438

We anticipate the federal appliance labeling and information programs (part of the National Energy Policy) to be in full swing by 1980 and that consumers will have similar information to the C-subjects, and as a result there will be a tendency toward the more energy efficient and smaller size units. Probably the largest effect on the average refrigerator-freezer size will come as a result of increased insulation (to achieve energy savings required by the California targets) in the side-by-side products, which will reduce the storage volume since most units are already at the limit for the outer cabinet size. Based on the projected food consumption trends, the likely storage volume reduction of the side-by-side and the influence of energy labeling and information programs, we see a leveling off in the trend toward larger refrigerator-freezers and would expect that the mean size will be in the 16-17 cubic foot range by 1985, as indicated in Figure 1.

1.5 Refrigerator Features

As discussed in the previous section, the sales-weighted mean size of the refrigerator-freezer by 1985 will probably be about 16-17 cubic feet.

In a recent survey* published in the August 1977 Merchandising magazine, "fifty-nine percent of the 300 retailers responding said that more consumers are specifically asking for energy-saving features on electrical appliances..Refrigerators ranked number one as the product area in which consumer requests for energy-saving features were most frequent." In a survey of 1500 consumers, 90% said energy efficiency was important, but less than half (48%) said that they would purchase the more energy-efficient model over the unit with more features.

The market share of the automatic defrosting features has grown to over 74% of the total market.** Automatic defrosting is clearly an important feature and should be offered in the high efficiency model. Additional features such as ice makers and hot water dispensers should also be considered, as long as the efficiency targets can be met. A number of other features and considerations that customers can observe are given in Table 5.

2. DESIGN OPTIONS

Eighteen design options for saving energy are considered in this chapter. These options are rated as the first step in selecting the best options for development and demonstration. Section 2.1 presents a rating methodology; Section 2.2 estimates refrigerator performance as it relates to energy consumption; and Section 2.3 presents the energy-saving options and their rating.

2.1 Rating Method

The rating of design options, presented here, reflects the consumers' perception of the value of the change and the cost. The rating uses consumer perception indices to quantify the following factors:

1) Annual Energy Savings
2) Added First Cost
3) Effect on Noise
4) Effect on Storage Volume
5) Effect on Unit Life

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* Opinion file - Marketing Energy Savings: The Sales Are There, August, 1977, Merchandising.

** Merchandising, 1976 Statistical and Market Report.

TABLE 5

SUBJECTIVE FEATURES

Type of Unit	Discussion
Refrigerator/freezer combination	Allows storage of both refrigerated and frozen foods in one appliance which is convenient to the food preparation area, and which is less bulky and less expensive than having two separate appliances (e.g., a small freezer/a small refrigerator).
Configuration	
Upright	A sizable appliance which fits into a relatively small amount of floor space with maximum access to its contents.
Side-by-Side	Good access to contents of frozen food comparment, and gives the appearance of having a separate freezer.
Doors	
See-through Doors	Smoked glass door panels with an outside switch for illumination of the interior could minimize door openings.
Inner doors	Allows even smaller compartments to be accessed without exposing entire unit to unchilled air.
Auxiliaries	
Ice maker	Provides continuous making of ice without the inconvenience of filling trays. Ones on the outside of doors provide ice without door openings and allow children access where they could not reach the freezer interior.
Cold water	Provides cold water without exposing refrigerator interior to unchilled air, and a convenient supply of cold water, particularly in regions where tap water temperatures exceed 65°F.
Hot water	Instant hot water without the inconvenience of boiling water on the stove.
Removable Compartments	
Passive	Allows separation of various foodstuffs (e.g., vegetables, meats, etc.) and easier access to items through a slide-out feature.
Active (plug in)	Allows separate temperatures within the refrigerator itself to keep items at their most appropriate temperature without harming other items which have different temperature needs. Also performs functions above.
Frost free	
Free convection coils vs. forced convection coils	Consumer unlikely to note differences except possibly that forced convection provides faster ice-cube formation. Frost-free refrigerators relieve owners from the messy, time consuming, and inconvenient manual defrosting of refrigerator-freezers. Furthermore, manual defrost may allow food to warm (or thaw) beyond a safe temperature, so the flavor, value, and healthfulness of the food is compromised.
Size	
Exterior dimensions	Most kitchens have one set location for their refrigerators; the refrigerator must fit in this space and still be large enough to hold a sizable quantity of foods. This puts the onus on the appliance to take up the minimum amount of space with its running apparatus and insulation.

Additional feature-related factors (e.g., ice maker, hot water, etc.) arising from any of the design changes should be noted, though they will not be used directly in rating the value of the energy-saving option.

To estimate consumers' responses to a design change, one must know the degree to which the change is perceived, plus the relative value the factor received in the final purchase decision. For example, in a recent Arthur D. Little, Inc., study in New York of air-conditioner purchases, it became apparent that for the majority of consumers a 15-20% price savings, combined with the smaller, lighter weight of a low efficiency unit, would more than offset a 20-30% savings in expected annual operating cost.* The savings must be perceived to outweight the cost of decline in convenience.

The bulk of consumers (2/3) are unable to perceive a directly observable change unless it is in excess of 12-18% (15% ± 3%). This is true with regard to simple phenomena such as light, sound, the length of a line, etc. This pattern is known as Weber's Law of Perception. When the opportunity to perceive change involves a long period of time (days or weeks), the amount of difference to be perceived must be even greater than 15%. (There are, of course, individual differences with some being able to perceive very small changes in the range of 3-6% and other failing to perceive differences until the change is 40-50%.) Thus, for factors affecting energy consumption, consumers in theory are not likely to perceive changes less than 18%. However, we anticipate that consumer sensitivity can be raised through government sponsored consumer information programs and appliance labelling. As discussed on page 11, consumers elected to purchase refrigerator/freezers consuming 15% less energy when labels and supporting information were provided. We conclude that a 15% ± 3% perception level for energy savings will be indicative of future consumer trends.

It is true that one can show "side-by-side" ratings such as one refrigerator costing $96 a year to run and another costing $116, but this is in the realm of a claim which, once the unit is in the home, is not perceivable to most and will have the effect of undermining the claim, i.e., "They said I'd save $20, but I haven't been able to find it!"

Another point is that once a first perceivable change is made, the next change must be of greater magnitude to be noticed--if operating cost is reduced 15% in one year, then for that consumer to perceive another change requires 15% of the <u>new</u> baseline of experience.

A time frame index of perception for the rating factors were assigned and given arbitrary weighting values in which immediately perceived factors are given the highest priority:

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*Arthur D. Little, Inc., "Efficacy of Incentives to Optimize Performance and Efficiency of Air Conditioners," prepared for the Administration and Management Research Association of New York City under NSF Grant #DL 42517, July 1975.

TABLE 6

EXAMPLE OF RATING CONSUMER REACTIONS TO ENERGY-SAVING OPTIONS

Factor*	Minimum Change to Perceive	Weight	Improved Insulation Effect	Improved Insulation Rating**
Annual Energy Savings	\$8±\$1	2	\$8 saved	+2
Added First Cost	\$48±\$10	3	\$14 added	0
Noise	3 db***	3	0 db	0
Storage	2±0.5 Cu.Ft.	3	0 Cu.Ft.	0
Unit Life	2.7±0.5 years	1	0 years	0
				+2

* Assumes subjective factors held constant for an average refrigerator. Many 16-cubic foot refrigerators will consume about 4.0 kwh/day (California 1979 standard), which means a \$58.40 energy cost per year (4¢/kwh) and 15% of this is \$8.76. Other factors are:

	Standard	Consumer Sensitivity Level	Minimum for Perceptable Change	Perceptable Change
Annual Energy Cost	\$58	Medium	15%±3%	\$8±\$1
First Cost	\$400	High	12%±3%	\$48±\$10
Storage	16 Cu.Ft.	High	12%±3%	2±0.5 Cu.Ft.
Unit Life	15 years	Low	18%±3%	2.7±0.5 years
Noise	-	High	NA	3 db***

** If the change is perceivable, the rating reflects the plus or minus weight value.

*** Minimum audible level.

- Added first cost--immediate, weight 3
- Effect on noise--immediate, weight 3
- Effect on storage volume--immediate, weight 3
- Annual energy savings--medium, weight 2
- Effect on unit life--long term, weight 1

Thus, an example of rating an energy-saving option would be as shown in Table 6. Options which have any adverse impacts are assigned a negative weighting value.

2.2 Estimation of Steady State Performance

The selection of promising improvements depends upon an estimate of the energy savings of each improvement. The method described below is the simple steady state model used in this analysis.

The steady state model of the refrigerator cabinet is from Volume 1* of a report prepared for the Federal Energy Administration by Arthur D. Little in Appendix A, pages 155 and 156, and is used throughout the screening of improvements. The governing equation for the daily energy consumption is:

$$\frac{kwh}{day} = .024 \left(\frac{Q_s}{Q_r - I}\right) \left(\frac{Q_r}{EER} + E\right)$$

where

Q_s = steady state cabinet heat load in Btu/hr,

Q_r = refrigeration unit cooling capacity in Btu/hr,

I = internal heat load accompanying refrigeration unit run time,

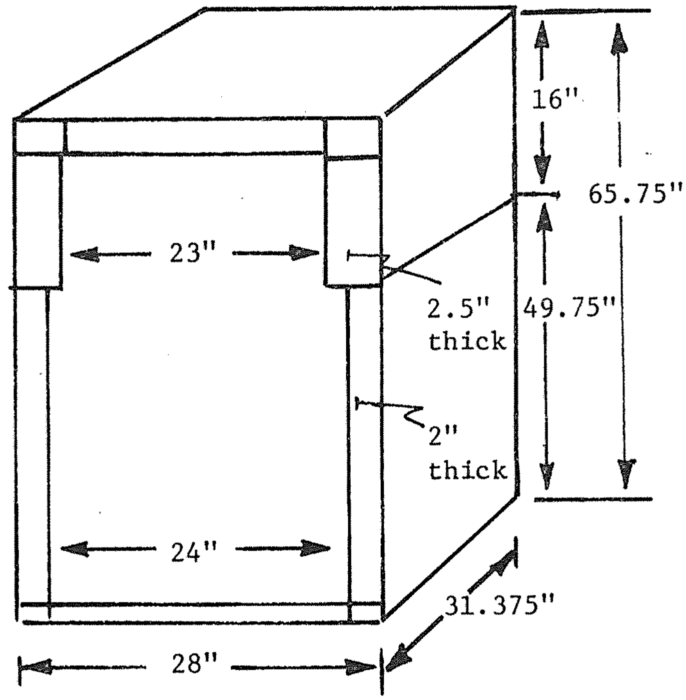
EER = refrigeration unit enegy efficiency ratio $\frac{\text{cooling capacity Btu/hr}}{\text{input energy watts}}$

E = auxiliary energy- such as fans, heaters, etc. in watts.

Baseline values for the 16-cubic-foot refrigerator-freezer used in this analysis are given in the following sections for an assumed 90° F room temperature.

---

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



	Area (Sq. Ft.)		Thickness (In.)		UA (K = .13) $\frac{Btu}{HR-°F}$	
	FF	FZ	FF	FZ	FF	FZ
Sides	20.33	6.38	2.0	2.5	1.014	.280
Back	8.99	2.85	2.0	2.5	.45	.125
Front	8.99	2.815	1.5	1.5	.55	.187
Top and Bottom	5.30	5.15	2.0	2.0	.28*	.273
					2.294	.865

---

* Underside compartment air is at an average temperature of about 50 °F above room temperature due to precooler and compressor.

FIGURE 2 BASELINE REFRIGERATOR-FREEZER DIMENSIONS

Q_s - The Steady Heat Load (in Btu/hr)

Freezer Walls (see Figure 2):

$$Q_s \text{ (Btu/hr)} = UA\ (T_o - T_i)$$
$$= .865\ (90°F - 5°F)$$
$$= 73.5$$

Refrigerator Walls:

$$Q_s \text{ (Btu/hr)} = \sum UA\ (T_o - T_i)$$
$$= 2.294\ (90°F - 38°F) + .26\ (50°F)^*$$
$$= 132.3$$

Where

$$UA = \sum_{i=0}^{6} (UA)_i$$

(i = each of the six outer walls of the cold compartments)

and

$$(UA)_i = \left(\frac{1}{\frac{\Delta X}{kA} + \left(\frac{1}{hA}\right)_{\text{inside}} + \left(\frac{1}{hA}\right)_{\text{outside}}} \right)$$

$$= \left(\frac{A}{\frac{\Delta X}{K} + \frac{2}{h}} \right)$$

where

*Reflects high temperature under refrigerator cabinet due to precooler and compressor (T ≈ 140°F).

		Freezer	Refrigerator
Nominal Insulation Thickness	$\Delta X =$	2.5 inches	2.0 inches
Insulation Conductivity	$K =$	$.13 \frac{\text{Btu-in}}{\text{Hr-Ft}^2\text{-}^\circ\text{F}}$	
Film Coefficient*	$h =$	$.57 \frac{\text{Btu}}{\text{Hr-Ft}^2\text{-}^\circ\text{F}}$	
Surface Area	$A =$	18.3 Ft^2	44.9 Ft^2

In addition to wall heat load is that heat contributed by the door area (gasket and throat area) where the following conductances have been used:

$$\text{Static} = .05 \frac{\text{Btu}}{\text{Hr-Ft-}^\circ\text{F}}$$

$$\text{Added dynamic (during evaporator fan run times)} = .028 \frac{\text{Btu}}{\text{Hr-Ft-}^\circ\text{F}}$$

The gasket length for the refrigerator is 12.50 feet, adding 30.6 Btu/hr; for the freezer, the gasket length is 6.98 feet, adding 29.6 Btu/hr.

The total cabinet steady-state heat load is:

$$Q_s = 266 \text{ Btu/hr}$$

I - Internal Heat Load (when compressor is on)

Items in this category include:

$$\text{Defrost } \left(\frac{500 \text{ watts x 15 min/cycle}}{8 \text{ hrs between cycles}}\right) = 53.3 \text{ Btu/hr}$$ (This accounts for the heat of fusion thermal load of the ice formation)

* McAdams, "Heat Transmission", page 173 for laminar heat transfer natural convection; Δt = temperature difference between the wall (about 20°F), 1 is the characteristic wall height in feet, and $h = .29\ (\Delta T/1)^{.25}$

Mullion Heater	= 18.8* Btu/hr	Off when in dry mode
Ref. Top Heater	= 17.1* Btu/hr	
Door Closure	= 42.8 Btu/hr	
Evaporator Fan	= 51.1 Btu/hr	
Hot Wall Condenser	= 42.8 Btu/hr	

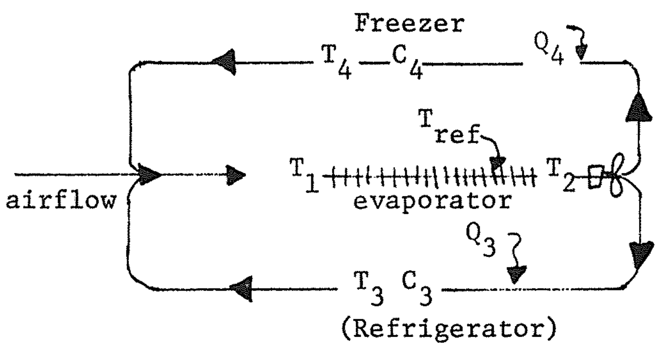
E - Electrical Load of Auxiliaries

Corresponding to the I Loads:

Mullion Heater	= 11* watts
Ref. Top Heater	= 5* watts
Defrost $\left(\frac{\text{500 watts x 15 min/cycle}}{\text{8 hrs between cycles}}\right)$	= 15.6 watts
Fan	= 15 watts

The Refrigeration Unit

The evaporator airflow circuit is shown below:



*Value of heater power when humidity switch is in the full on position, ½ this value with switch set to medium, and 0 with switch set to low.

where:

T_1 = mixed return air °F

T_2 = cold outlet air °F

T_3 = fresh-food compartment °F

T_4 = freezer food compartment °F

T_{ref} = refrigerant temperature

= the evaporator coil temperature °F (neglecting temperature difference between coil and fluid)

C_3 = the fresh food compartment air capacity rate $\dot{m}_3 C_p$ Btu/hr-°F

C_4 = the freezer air capacity rate $\dot{m}_4 C_p$ Btu/hr-°F

C_T = total air capacity rate Btu/hr-°F

Q_3 = fresh food refrigeration capacity in Btu/hr

Q_4 = freezer refrigeration capacity in Btu/hr

The governing equations are:

$$Q_4 = C_4 \ (T_4 - T_2) \tag{1}$$

$$Q_3 = C_3 \ (T_3 - T_2) \tag{2}$$

$$T_1 = T_3 \ \frac{C_3}{C_T} + T_4 \ \frac{C_4}{C_T} \tag{3}$$

$$C_T = C_3 + C_4 = (1.24)\mathrm{x}(\mathrm{Fan\ CFM}) \tag{4}$$

$$Q_4 + Q_3 = C_T \ \eta(T_1 - T_{ref}) \tag{5}$$

where the evaporator effectiveness $\eta = .75$

Also,

$$Q_4 + Q_3 = \alpha T_{ref} + \beta \tag{6}$$

where

α and β are known constants for a compressor-condenser unit.

and,

$$\frac{Q_4 - I_4}{Q_3 - I_3} = \text{ratio of refrigeration capacity} \tag{7}$$

= the known ratio of cabinet heat loads for the specified temperature.*

Equation 6 is the condensing unit (compressor-condenser) relation and α and β are constants for known pressure. For a 90°F room, we assume that the condenser is at 115°F and the total refrigeration capacity Δh_r is:

$$\Delta h_r = h_{gas} \text{ (90°F, gas at evap pressure)} - h_{sat.\ liq.} \text{ (at 115°F)}$$

= 53.4 Btu/lbm for a range of evaporator temperatures (0-20°F) and R-12

---

*Since

$(Q_4 - I_4)$ x percent on time = load in freezer

where

I_4 is freezer portion of I,

and

$(Q_3 - I_3)$ x percent on time = load in refrigerator,

then

$$\frac{Q_4 - I_4}{Q_3 - I_3} = \frac{\text{heat load in freezer}}{\text{heat load in refrigerator}}$$

$$= \frac{103.1}{164.8}$$

$$= .625.$$

From the compressor curve for the ¼ horsepower compressor (Tecumseh Compressor AE1360 B), we get:

$$\dot{m}\ (\text{lbs/hr}) = .325\ T_{ref} + 14.25$$

and the compressor capacity Q_r is:

$$Q_r = \Delta h_r\ \dot{m} = 17.355\ T_{ref} + 760.95\ \text{Btu/hr}.$$

The seven equations and the seven unknowns T_1, T_2, T_{ref}, C_3, C_4, Q_3, and Q_4, can be solved by iteration for C_T = 50 Btu/hr-°F yielding:

$$T_2 = -1.5\ °F$$

$$T_1 = 11.64°F$$

$$Q_r = 658\ \text{Btu/hr}$$

$$T_{ref} = -5.914°F$$

$\dot{m}_r$ = 12.3 lbm/h

The energy consumption of the compressor is then found from the compressor curves for watt consumption, yielding a compressor wattage of 184.9* and an EER of 3.56 Btu/hr-watt.

Daily Energy Consumption

These terms are combined into the formula:

$$\text{kwh/day} = .024\ \left(\frac{Q_s}{Q_r - I}\right)\ \left(\frac{Q_r}{EER} + E\right)$$

$$= .024\ \left(\frac{266}{658 - 190}\right)\ (184.9 + 30.6)$$

$$= 2.945^{**}\ \text{kwh/day}$$

2.3 Energy-Saving Options

Some 18 design improvements were identified for consideration, many were considered in the FEA study. Table 7 shows the list of design options examined. One page summaries of these options are given in the following

---

*watts (P_{hi} = 180 psig) = 203.49 + 3.184 T_{ref}

**humidity switch set to low

TABLE 7

ENERGY-SAVING OPTIONS

Non-Proprietary Options	Proprietary Options	Rating Value	Comments
Most Promising Prospect			
1. Optimized Insulation Thickness		+2	
2. Alternative Condenser Design		+3	
3. Door Seal Improvement		0	
4. Improved Evaporator Fan System		0	
	5. New Evaporator	+2	
	6. Hot Water Feature	+2	
	7. Improved Defrost Control	0	
Good Prospect			
8. Improved Static Condenser Design		+2	Significant design uncertainties
9. Expansion Valves		0	
	10. Sequential Control	+2	
Distant Prospect			
11. Evacuated Powder Insulation		+2	Development beyond the time frame of 1985 production
Not Promising Prospect			
12. Multiple Evaporator-Compressor		-2	Poor payback period
13. Thermal Storage		0/2	Small energy savings, uncertain peak electric cost
14. Mechanical Expander		-4	Too many uncertainties
15. Other Thermodynamic Cycles		-	No energy savings with known components
16. Hot Gas Defrost		-1	Poor payback periods
17. Inner Doors		-	Option #3 makes this obsolete based on present test procedures
18. Two-Speed Compressor		0	Poor payback period

NOTES ON CLASSIFICATIONS

- The classification of "Most Promising Prospects" are options which appear to have payback periods less than 3 years and are regarded by Amana as being most promising.
- "Good Prospects" are options which offer good payback periods but at a perceived higher risk. These concepts will be examined by ADL, though no explicit prototype is presently planned to incorporate these.
- "Distant Prospect" means that development is beyond the scope of this work.
- "Less Promising," see specific note.

pages. An estimate of the effect of the top 6 options is given at the end of this section.

These options have all been evaluated by estimating the unit performance under a closed door 90°F condition. This corresponds to the present DOE test procedure. Design options such as number 17, Inner Doors, would show greater promise in a door opening test. We recommend that the options (8 -- 18) which have not been chosen for the development and testing be reconsidered in light of a new test procedure which includes door openings.

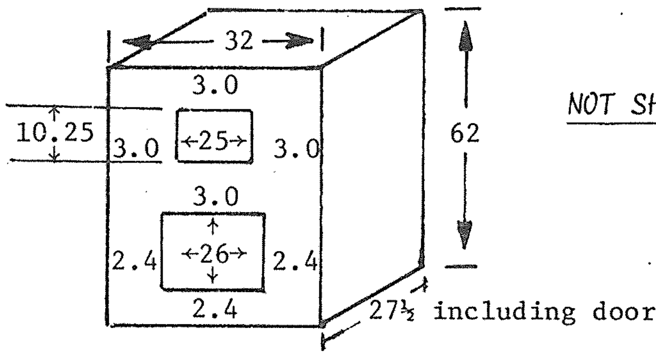
DESIGN OPTION 1.

Optimum Insulation Thickness

DESCRIPTION

Present 16-cubic-foot unit has 2-inch insulation in fresh-food compartment, 2.5-inches in freezer, and measured 28" wide, 66" high, and 31" deep. Widening to 32"x67"x31", an overall increase in insulation thickness may be achieved. This includes the foaming of a 2.4-and 3-inch thick door.

SCHEMATIC



NOT SHOWN TO SCALE

ENERGY PARAMETERS FOR 90°F ROOM

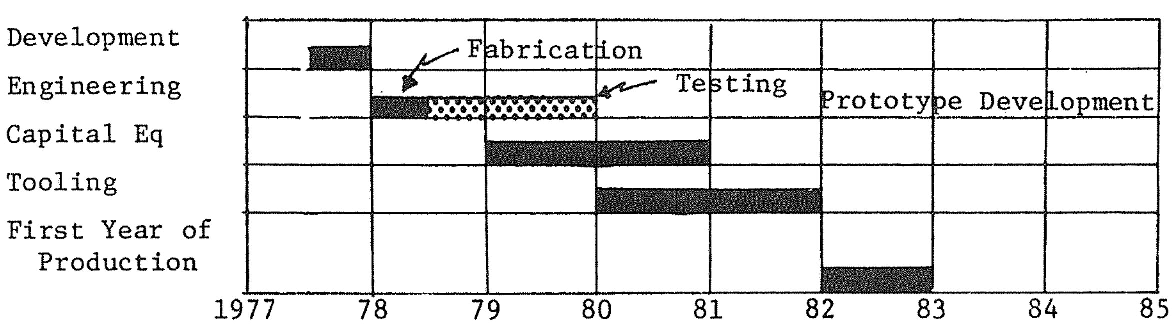
	Q_s Btu/Hr	Q_r Btu/Hr	I Btu/Hr	EER Btu/Hr-Watt	E Watts	Kwh/Day
Baseline	266	658	190	3.558	30.5	2.945
Design Option	223.1	658	190	3.558	30.5	2.46

SAVINGS: .48 Kwh/Day

ECONOMICS

Cost to Manufacture	Cost to Consumer	Annual Energy Savings	Years to Payback
$5.50	$14.00	$7.00	2

LEAD TIMES



+ See other side

CONSUMER ACCEPTANCE

	Consumer Perception Guide			
	Minimum Perceptable	Option Values	Standard Weighting	Option Evaluation
Annual Energy Savings	$8 ± $1	7	2	+2
First Cost	$48	14	3	0
Noise	3 db	0	3	0
Storage	2 cu. ft.	0	3	0
Unit Life	2.7 years	0	1	0
Rating				+2

ADDITIONAL NOTES

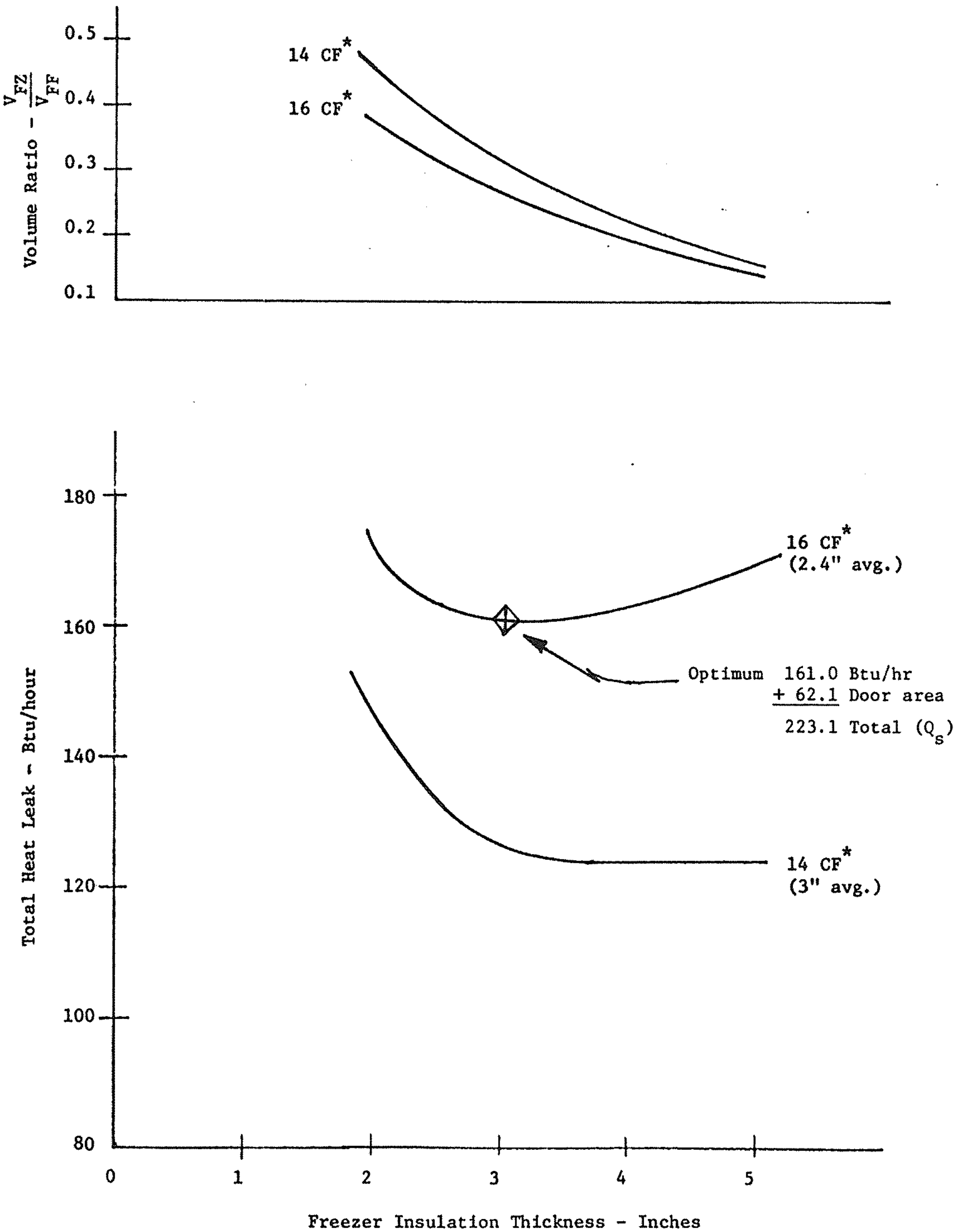
Optimized insulation thickness is based on the distribution of polyurethane foam in the refrigerator to maximize the foam utilization. Figure 3 shows the influence of the freezer thickness on minimizing the overall cabinet energy consumption for a fixed size refrigerator-freezer and for a fixed amount of insulation material. The amount of insulation material is set by specifying the outer cabinet dimensions (Figure 4) and the inner usable storage volume. Varying the freezer thickness automaticallly determines the fresh-food compartment insulation thickness. This figure shows the range of desirable freezer insulation thickness for optimal use of polyurethane foam insulation. A value of three inches of freezer insulation thickness was selected for the prototype in this program.

AREAS OF UNCERTAINTY

1. Increased foam thickness means longer cure times and larger chance of inconsistent foaming.

2. The consumer perception of the loss in storage volume from what was originally an 18-cubic-foot refrigerator cabinet to the 16-5 cubic-foot with thicker walls.

3. Measured savings of a non-optimized 3-inch thich insulation system over the 2, 2½ system is about .4 kwh/day.



*Based on mullion volume = 1 CF

FIGURE 3

INFLUENCE OF FREEZER INSULATION THICKNESS

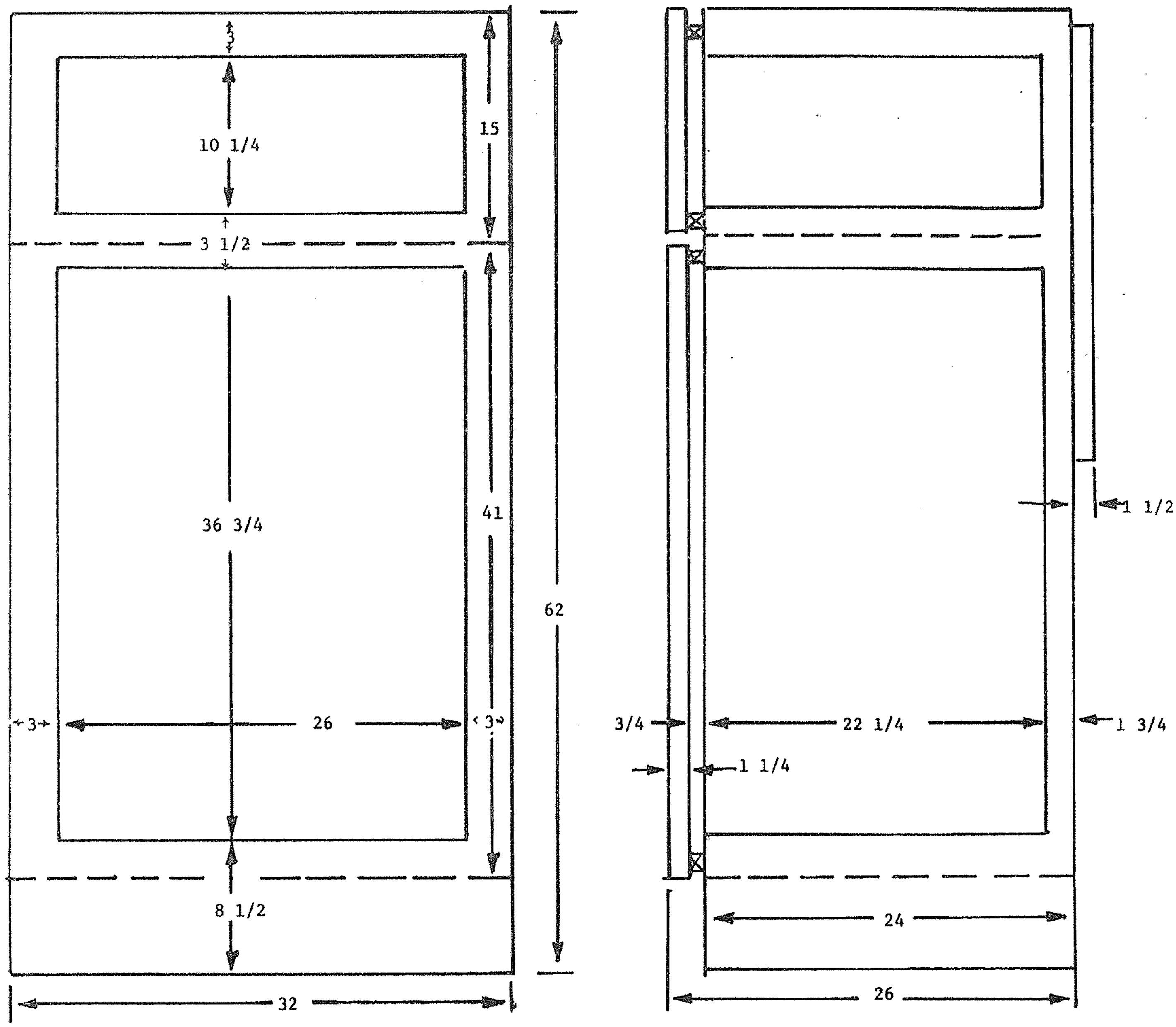


FIGURE 4

OUTER CABINET DIMENSIONS
(All in Inches)

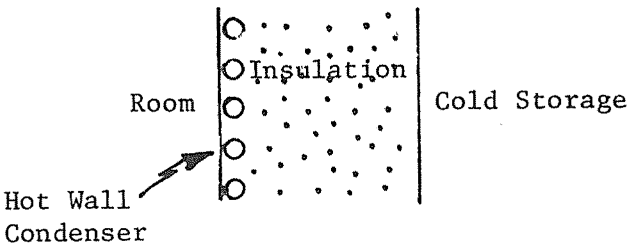
DESIGN OPTION 2.

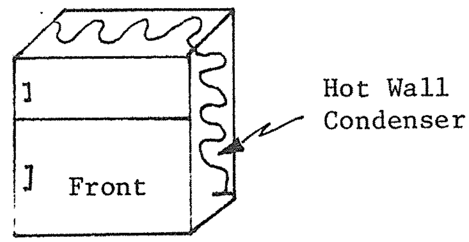
Removal of Hot Wall Condenser

DESCRIPTION

The condenser is fixed to the outside wall of the cabinet. We estimate a minimum heat addition to the cabinet of 43 Btu/hour+ due to the condenser. Removal of the hot wall condenser and replacement with a back-mounted condenser could reduce the internal heat (I) component by this entire amount, from 190 to 147.

SCHEMATIC





ENERGY PARAMETERS FOR 90°F ROOM

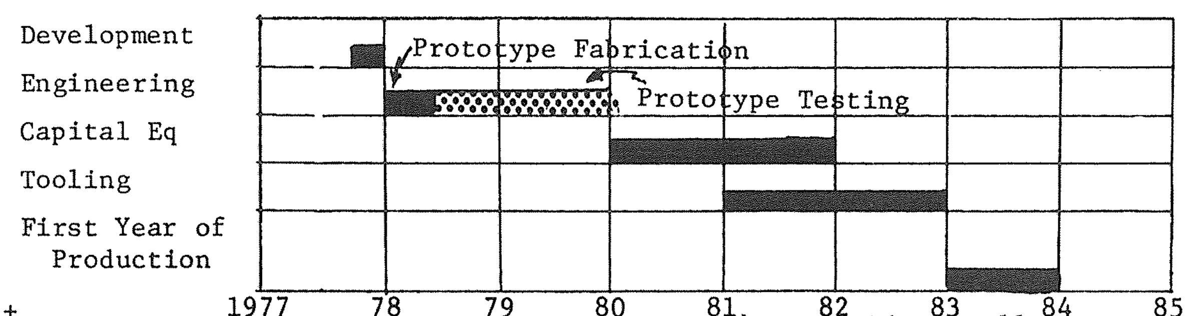
	Q_s Btu/Hr	Q_r Btu/Hr	I Btu/Hr	EER Btu/Hr-Watt	E Watts	Kwh/Day
Baseline	266	658	190	3.558	30.5	2.945
Design Option	266	658	147	3.558	30.5	2.70

SAVINGS: .25 Kwh/Day

ECONOMICS

Cost to Manufacture	Cost to Consumer	Annual Energy Savings	Years to Payback
$-10	$ -25	$ 3.65	NA

LEAD TIMES



+Based on assumption that the hot wall raises ½ the cabinet wall area temperature by 25°F, this amounts to a (2.58 + .843/2 (25°F) = 43 Btu/hr increase in internal thermal (I) due to the hot wall condenser.

OVER

CONSUMER ACCEPTANCE

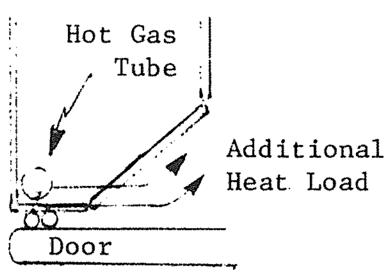
	Consumer Perception Guide: Minimum Perceptable	Option Values	Standard Weighting	Option Evaluation
Annual Energy Savings	$8	3.65	2	0
First Cost	$48 ± $10	−25	3	+3*
Noise	3 db	0	3	0
Storage	2 cu. ft.	0	3	0
Unit Life	2.7 years	0	1	0
Rating				+3

ADDITIONAL NOTES

*Though a $25 charge in first cost should not in theory be perceived, we judge that retail advertisement will highlight a first cost savings, so we assign the full weight advantage to this savings.

AREAS OF UNCERTAINTY

The actual amount of internal load due to the hot wall condenser is very uncertain. In addition to the thermal load on the plane wall surface, substantial heat gain from the condenser is experienced in the door closure area.



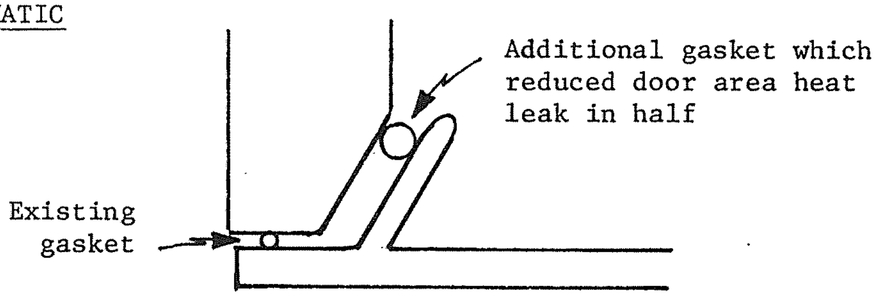
DESIGN OPTION 3.

Improved Door Seal

DESCRIPTION

Cold air wiping through the door closure area is a significant heat leak. Addition of an inner seal could inhibit much of the heat leak. Presently, the heat load is 62.1 Btu/hr steady with an additional 42.8 Btu/hr with the fan on. Eliminating 1/2 of the door area heat leak would reduce Q_B to 236 Btu/hr and I to 168.6 Btu/hr.

SCHEMATIC



ENERGY PARAMETERS FOR 90°F ROOM

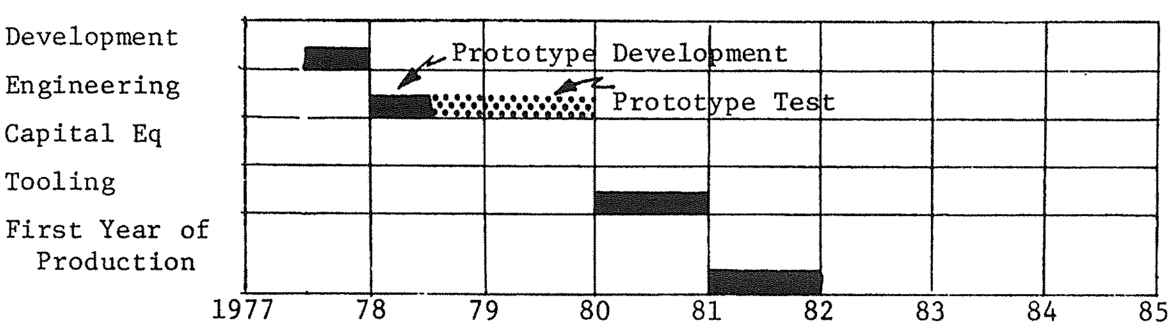
	Q_s Btu/Hr	Q_r Btu/Hr	I Btu/Hr	EER Btu/Hr-Watt	E Watts	Kwh/Day
Baseline	266	658	190	3.558	30.5	2.945
Design Option	236	658	168.6	3.558	30.5	2.50

SAVINGS: .45 Kwh/Day

ECONOMICS

Cost to Manufacture	Cost to Consumer	Annual Energy Savings	Years to Payback
$ 0.80	$ 2.00	$ 6.57	0.3

LEAD TIMES



OVER

CONSUMER ACCEPTANCE

	Consumer Perception Guide			Option Evaluation
	Minimum Perceptable	Option Values	Standard Weighting	
Annual Energy Savings	$8 ± $1	6.57	2	0
First Cost	$48	$2.00	3	0
Noise	3 db	0	3	0
Storage	2 cu. ft.	0	3	0
Unit Life	2.7 years	0	1	0
Rating				0

ADDITIONAL NOTES

AREAS OF UNCERTAINTY

1. How to develop a good inner seal held by the force of the magnetic outer gasket. The material must be highly compliant, yet durable.

2. The actual magnitude of door area heat load is not well known, and the savings from an additional gasket is very uncertain.

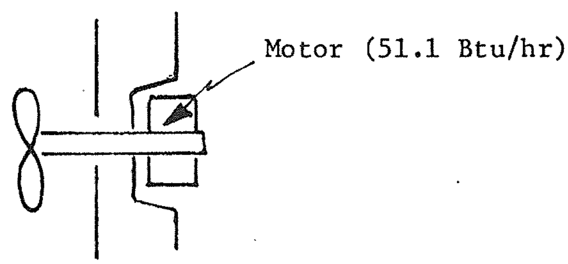
DESIGN OPTION 4.

Improved Evaporator Fan

DESCRIPTION

Presently the evaporator fan and motor are wholly within the cold space. An improved fan design could place the motor outside of the cold space.

SCHEMATIC



ENERGY PARAMETERS FOR 90°F ROOM

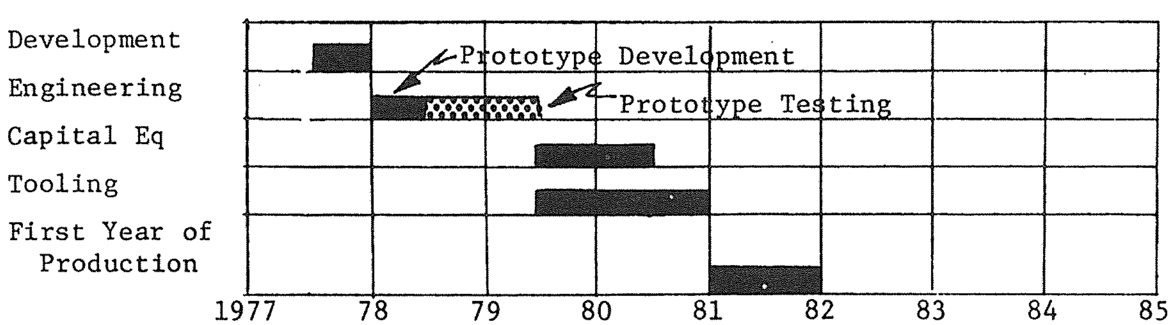
	Q_s Btu/Hr	Q_r Btu/Hr	I Btu/Hr	EER Btu/Hr-Watt	E Watts	Kwh/Day
Baseline	266	658	190	3.558	30.5	2.945
Design Option	266	658	138.9	3.558	30.5	2.65

SAVINGS: .3 Kwh/Day

ECONOMICS

Cost to Manufacture	Cost to Consumer	Annual Energy Savings	Years to Payback
$ 0.5	$ 1.25	$ 4.38	0.3

LEAD TIMES



OVER

CONSUMER ACCEPTANCE

	Consumer Perception Guide			Option Evaluation
	Minimum Perceptable	Option Values	Standard Weighting	
Annual Energy Savings	$8	4.38	2	0
First Cost	$48	1.25	3	0
Noise	3 db	0	3	0
Storage	2 cu. ft.	0	3	0
Unit Life	2.7 years	0	1	0
Rating				0

ADDITIONAL NOTES

AREAS OF UNCERTAINTY

1. Fan and airflow configuration.
2. Bearing losses due to additional overhung load.
3. Heat leak and warm air infiltration through shaft area.

DESIGN OPTION 5.

New Evaporator

DESCRIPTION

A new evaporator designed to improve the energy efficiency.

SCHEMATIC

ENERGY PARAMETERS FOR 90°F ROOM

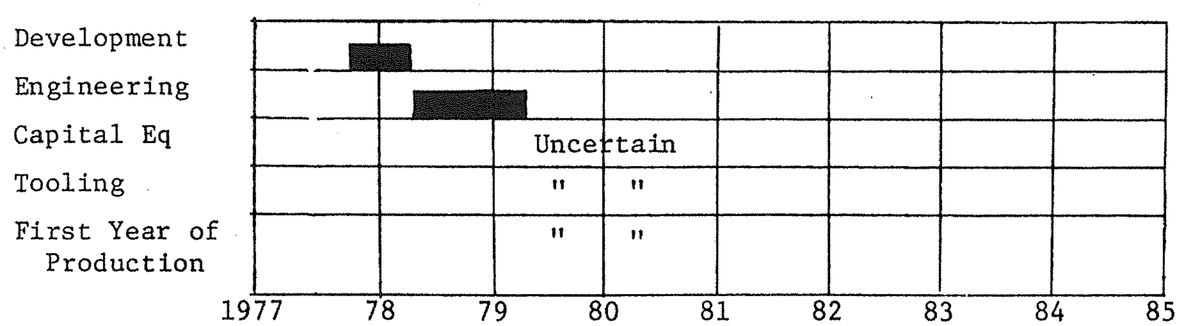
	Q_s Btu/Hr	Q_r Btu/Hr	I Btu/Hr	EER Btu/Hr-Watt	E Watts	[redacted]	Kwh/Day
Baseline	266	658	190	3.558	30.5	[redacted]	2.945
Design Option	266					[redacted]	2.392

SAVINGS: .552 Kwh/Day

ECONOMICS

Cost to Manufacture	Cost to Consumer	Annual Energy Savings	Years to Payback
$ <20	$ <38	$ 8.06	<5

LEAD TIMES



OVER

CONSUMER ACCEPTANCE

	Consumer Perception Guide: Minimum Perceptable	Consumer Perception Guide: Option Values	Consumer Perception Guide: Standard Weighting	Option Evaluation
Annual Energy Savings	\$8 ± \$1	8.41	2	+2
First Cost	\$48 ± \$10	< 38	3	0
Noise	3 db	0	3	0
Storage	2 ± .5 cu. ft	<1.5	3	0
Unit Life	2.7 years	0	1	0
Rating				+2

ADDITIONAL NOTES

AREAS OF UNCERTAINTY

DESIGN OPTION 6.

Hot Water Feature

DESCRIPTION

A system to provide hot water using heat recovered from the refrigerator.

SCHEMATIC

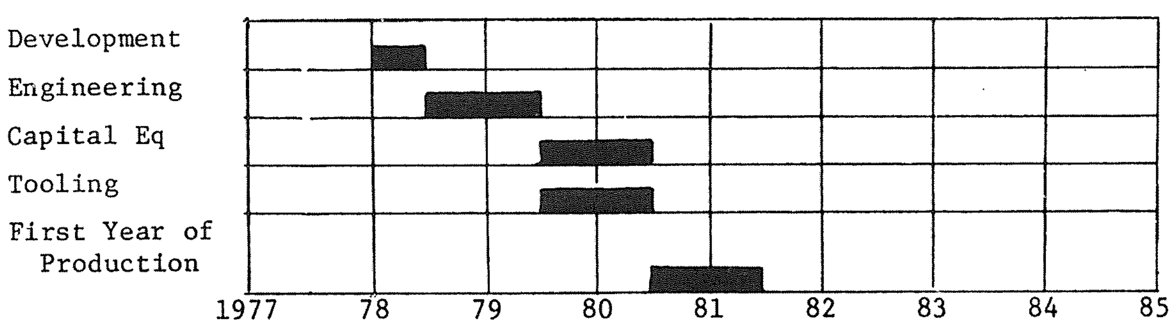
ENERGY PARAMETERS FOR 90°F ROOM

	Q_s Btu/Hr	Q_r Btu/Hr	I Btu/Hr	EER Btu/Hr-Watt	E Watts	Kwh/Day
Baseline	266	658	190	3.558	30.5	3.09
Design Option	-----No change (6,380 Btu/day H_2O heating)-----					--------

ECONOMICS

Cost to Manufacture	Cost to Consumer	Annual Energy Savings	Years to Payback
$<20	$32[+]	$ 27.31[++] (14.20)[+++]	1.17 (2.26)[++]

LEAD TIMES

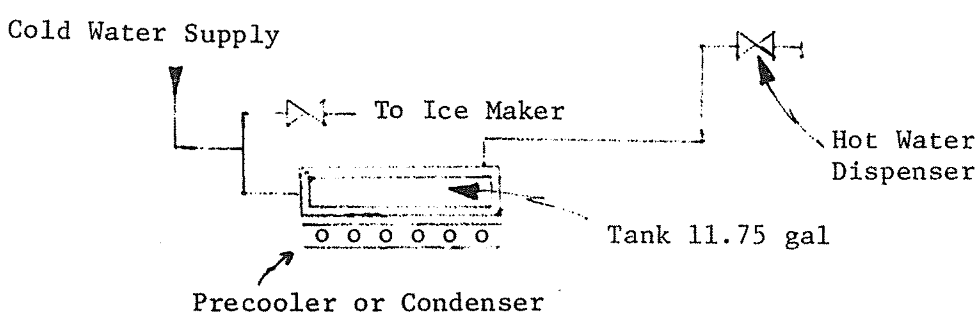


[+]Assumes ice maker installed, cold water already plumbed to refrigerator.
[++]Assumes no useful space heating by the refrigerator.
[+++]Accounting for refrigerator space heating displaced by water heating, and all estimates depend on customer usage in the kitchen.

CONSUMER ACCEPTANCE

	Consumer Perception Guide: Minimum Perceptable	Option Values	Standard Weighting	Option Evaluation
Annual Energy Savings	$8	14.20	2	2
First Cost	$48	32.0	3	0
Noise	3 db	0	3	0
Storage	2 cu. ft.	less than 2 cu.ft.	3	0
Unit Life	2.7 years	0	1	0
Rating				+2

ADDITIONAL NOTES



AREAS OF UNCERTAINTY

DESIGN OPTION 7.

Improved Defrost Control

DESCRIPTION

Increased compressor run time between defrost cycles can reduce energy consumption while providing the necessary defrost.

SCHEMATIC

ENERGY PARAMETERS FOR 90°F ROOM

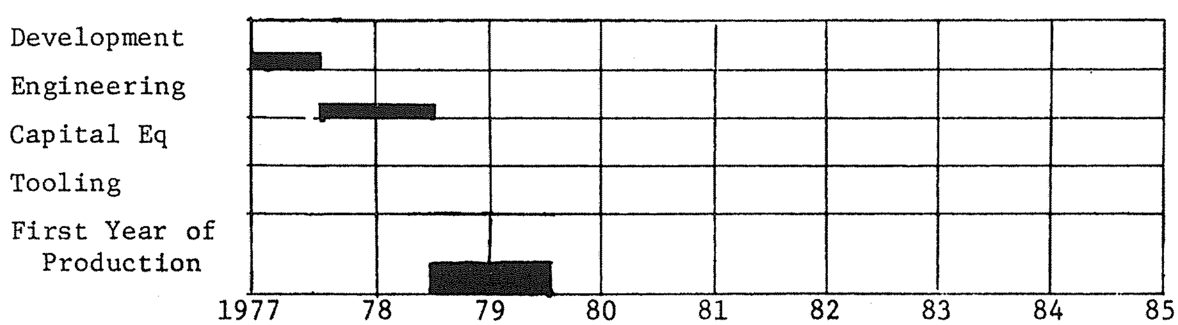
	Q_s Btu/Hr	Q_r Btu/Hr	I Btu/Hr	EER Btu/Hr-Watt	E Watts	Kwh/Day
Baseline	266	658	190	3.558	30.5	2.945
Design Option	266	658	172.2	3.558	25.31	2.762

SAVINGS: .183 Kwh/Day

ECONOMICS

Cost to Manufacture	Cost to Consumer	Annual Energy Savings	Years to Payback
$<20	$<38	$ 2.67	<14

LEAD TIMES



OVER

CONSUMER ACCEPTANCE

	Consumer Perception Guide: Minimum Perceptable	Consumer Perception Guide: Option Values	Consumer Perception Guide: Standard Weighting	Option Evaluation
Annual Energy Savings	$8 ± $1	2.67	2	0
First Cost	$48 ± $10	<38	3	0
Noise	3 db	0	3	0
Storage	2 cu. ft.	0	3	0
Unit Life	2.7 years	0	1	0
Rating				0

ADDITIONAL NOTES

AREAS OF UNCERTAINTY

Defrost performance under high humidity usage (so-called Gulf States Test).

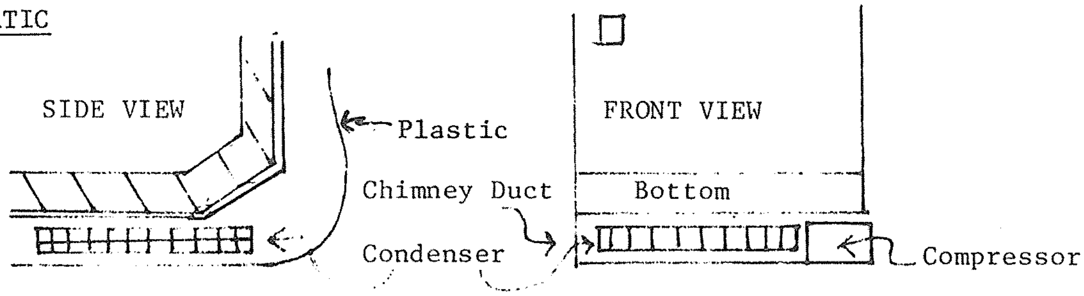
DESIGN OPTION 8.

Improved Static Condenser

DESCRIPTION

Present back-mounted static condensers draw air (buoyant air) from the bottom of the refrigerator up to the top. With a refrigerator in an alcove, the supply air to the condenser is pre-heated by the machine compartment (compressor). Back-mounted condensers add to the overall unit depth though space beneath the refrigerator is unused. An improved condenser would be mounted below and appropriate ducting would provide the natural draft.

SCHEMATIC



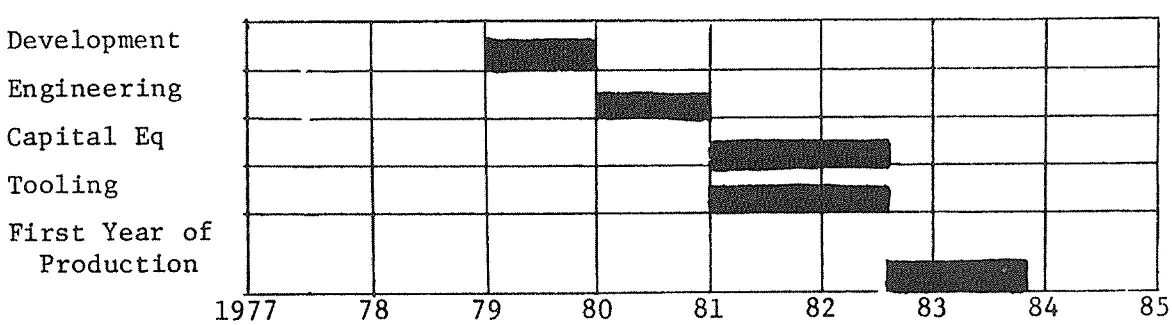
ENERGY PARAMETERS FOR 90°F ROOM

	Q_s Btu/Hr	Q_r Btu/Hr	I Btu/Hr	EER Btu/Hr-Watt	E Watts	Kwh/Day
Baseline	266	658	190	3.558	30.5	2.945
Design Option	266	758	147+	4.219	30.5	2.195
						.749

ECONOMICS

Cost to Manufacture	Cost to Consumer	Annual Energy Savings	Years to Payback
$ <20	$ <38	$ 10.93	<4

LEAD TIMES



+Includes elimination of hot wall condenser which accounts for .25 kwh/day savings. See Areas of Uncertainty on reverse side.

CONSUMER ACCEPTANCE

	Consumer Perception Guide: Minimum Perceptable	Option Values	Standard Weighting	Option Evaluation
Annual Energy Savings	\$8 ± \$1	10.93	2	+2
First Cost	\$48 ± \$10	<38	3	0
Noise	3 db	0	3	0
Storage	2 cu. ft.	0	3	0
Unit Life	2.7 years	0	1	0
Rating				+2

ADDITIONAL NOTES

Assume that the maximum effect is to lower the condensing temperature from 115° (90°F room) to 105°F. Then $\dot{m}$ = 13 lbs/hr and $\Delta h_{refrigeration}$ = 58.31 Btu/lbm; therefore Q_{ref} = 758.03 Btu/hr and also from AE 1360B data, watts = 179.63.

AREAS OF UNCERTAINTY

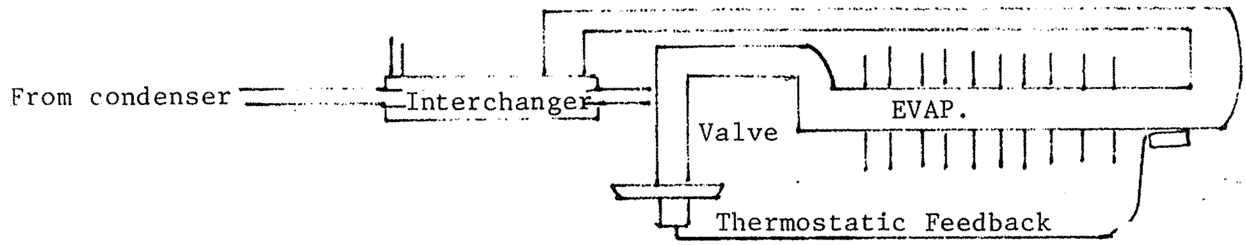
Unknown effect of condenser heat transfer to cabinet.

Expansion Valves

DESCRIPTION

Under real use conditions and for alternative refrigeration unit designs, expansion valves may provide improved system balancing and improved performance. This will be particularly true for alternative refrigeration unit design in which evaporator performance is intentionally modified significantly during running conditions.

SCHEMATIC



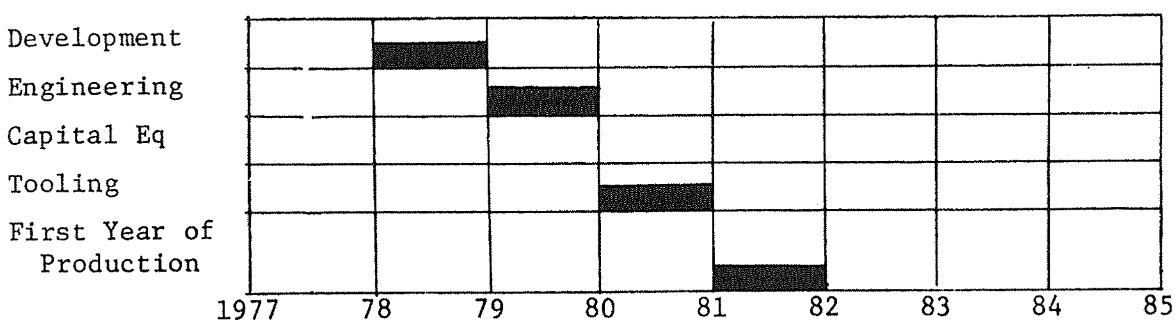
ENERGY PARAMETERS FOR 90°F ROOM

	Q_s Btu/Hr	Q_r Btu/Hr	I Btu/Hr	EER Btu/Hr-Watt	E Watts	Kwh/Day
Baseline	266	658	190	3.558	30.5	2.945
Design Option			unknown			

ECONOMICS

Cost to Manufacture	Cost to Consumer	Annual Energy Savings	Years to Payback
$4.54	$ 11.35+	$<5	<3

LEAD TIMES



+Based on a Singer thermostatic Valve F-12 6,000 Btu/hr rating

OVER

CONSUMER ACCEPTANCE

	Consumer Perception Guide			Option Evaluation
	Minimum Perceptable	Option Values	Standard Weighting	
Annual Energy Savings	$8	probably <5	2	0
First Cost	$48 ± $10	11.35	3	0
Noise	3 db	0	3	0
Storage	2 cu. ft.	0	3	0
Unit Life	2.7 years	<2.2	1	0
Rating				0

AREAS OF UNCERTAINTY

1. The pressure flow characteristic necessary for a balanced system (flooded evaporator) design over a wide range of operating conditions.

2. The pressure flow characteristics for alternative evaporator (Design Option # 5) designs.

3. Effect on unit life.

DESIGN OPTION 10.

Sequential Control

DESCRIPTION

Operation of the evaporator in sequence between the freezer and the fresh-food compartment, permitting partial operation at high evaporator temperatures (and higher efficiency) during fresh-food cooling.

SCHEMATIC

Not Applicable

ENERGY PARAMETERS FOR 90°F ROOM

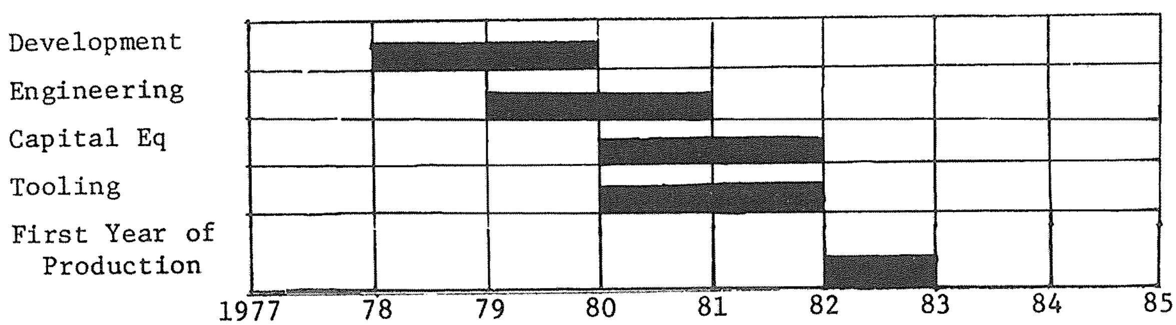
		Q_s $\frac{Btu}{Hr}$	Q_r $\frac{Btu}{Hr}$	I $\frac{Btu}{Hr}$	EER $\frac{Btu}{Hr-Watt}$	E Watts	$\frac{Kwh}{Day}$	
Baseline		266	658	190	3.55	30.5	2.945	
Design Option	FF	164.8	970.94	160.8	4.0	30.5	1.33	2.372
	FZ	10.31	579.59	133.13	3.40	30.5	1.042	

SAVINGS = .573 kwh/day

ECONOMICS

Cost to Manufacture	Cost to Consumer	Annual Energy Savings	Years to Payback
$ <20	$< 38	$ 8.36	<5

LEAD TIMES



OVER

CONSUMER ACCEPTANCE

	Consumer Perception Guide			Option Evaluation
	Minimum Perceptable	Option Values	Standard Weighting	
Annual Energy Savings	$8 ± $1	8.36	2	+2
First Cost	$48	< 38	3	0
Noise	3 db	0	3	0
Storage	2 cu. ft.	0	3	0
Unit Life	2.7 years	0	1	0
Rating				+2

AREAS OF UNCERTAINTY

1. Ability of present cap tube/condenser to maintain proper liquid level in evaporator during fresh-food operation.

2. Adverse effect of thermal inertia of evaporator.

3. Adverse effect of air leakage between spaces.

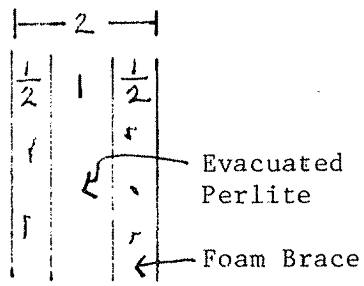
DESIGN OPTION 11.

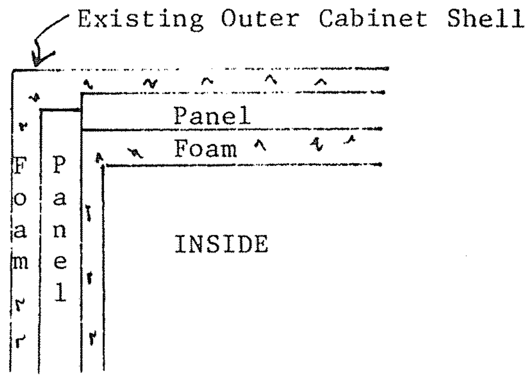
Evacuated Powder Insulation Panels

DESCRIPTION

Evacuated (1-100 micron) panels are inserted between existing foam-in-place walls as pre-assembled and tested panels. 5 mil mylar/al barrier is used.

SCHEMATIC





ENERGY PARAMETERS FOR 90°F ROOM

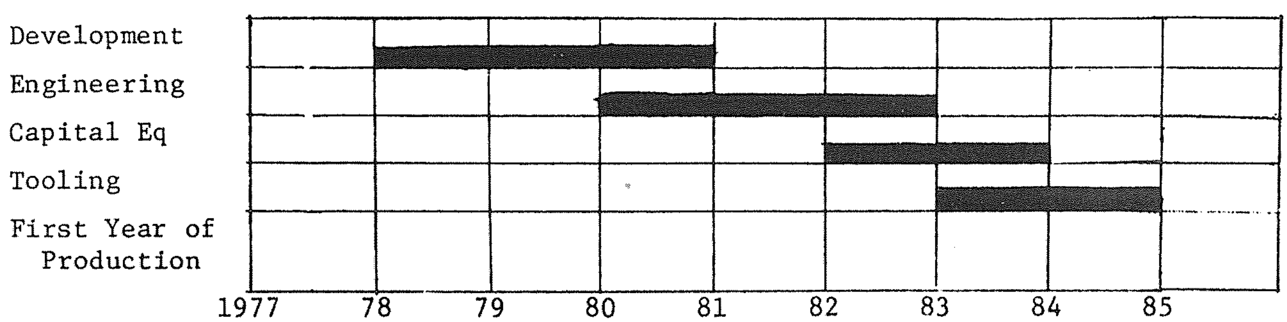
	Q_s Btu/Hr	Q_r Btu/Hr	I Btu/Hr	EER Btu/Hr-Watt	E Watts	Kwh/Day
Baseline	266	658	190	3.55	30.5	2.945
Design Option	75.1	658	125.8	3.55	30.5	.729

SAVINGS: 2.20 Kwh/day

ECONOMICS

Cost to Manufacture	Cost to Consumer	Annual Energy Savings	Years to Payback
$ 30.52	$ 76.30	$ 32.24	2.36

LEAD TIMES



OVER

CONSUMER ACCEPTANCE

	Consumer Perception Guide			
	Minimum Perceptable	Option Values	Standard Weighting	Option Evaluation
Annual Energy Savings	\$8 ± \$1	32	2	+2
First Cost	\$48 ± \$10	76	3	-3
Noise	3 db	0	3	0
Storage	2 ± .5 cu. ft.	+2	3	+3
Unit Life	2.7 years	0	1	0
Rating				+2

AREAS OF UNCERTAINTY

1. Vacuum integrity of insulating panels is a highly speculative prospect. A high vacuum, sensitive to component outgassing and microscopic leaks, will be difficult to maintain.

2. In the event of a tiny leak, the sealing quality of the surrounding poured-in-place foam is unknown.

3. The system will be cost-effective only with thin 5 mil. aluminum walls or plastic. Manufacturing vacuum-tight panels of these materials will require a new technological jump.

DESIGN OPTION 12.

Two Compressors with Two Evaporators

DESCRIPTION

Provide two compressors (or compressor cylinders) and two evaporators. Fresh-food evaporator can operate at a higher temperature than freezer evaporator to achieve higher EER. Free convection fresh-food evaporator can be used to achieve additional savings via elimination of fan and dynamic throat losses in fresh-food compartment.

SCHEMATIC

See Attached.

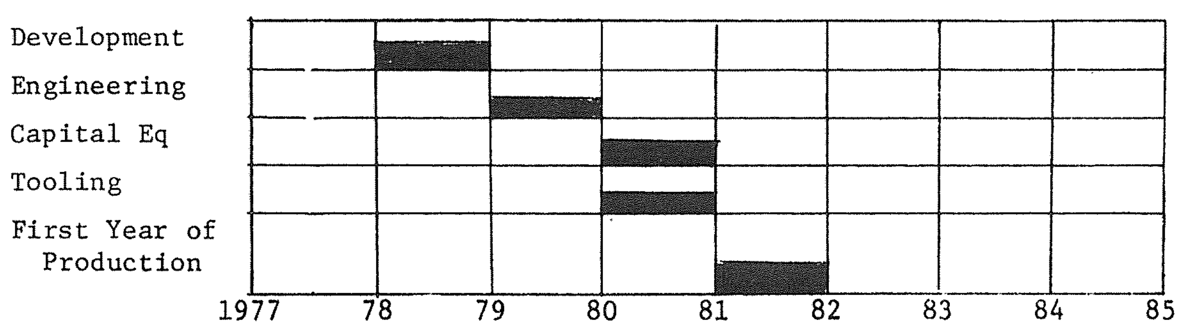
ENERGY PARAMETERS FOR 90°F ROOM

	Q_s Btu/Hr	Q_r Btu/Hr	I Btu/Hr	EER Btu/Hr-Watt	E Watts	Kwh/Day	SAVINGS Kwh/Day
Baseline	266	658	190	3.56	30.5	2.95	Baseline
Design Option 1*	266	855	140	3.43	23	2.44	.51
2**	266	623	140	3.97	23	2.38	.57

ECONOMICS

	Cost to Manufacture	Cost to Consumer	Annual Energy Savings	Years to Payback
Design Option 1	$21	$ 51.93	$ 7.45	7

LEAD TIMES



* Two available 360 Btu/hr compresssors in place of one 600 Btu/hr unit.
** Two 310 Btu/hr compressors with EER's like available 600 Btu/hr units.

OVER

CONSUMER ACCEPTANCE

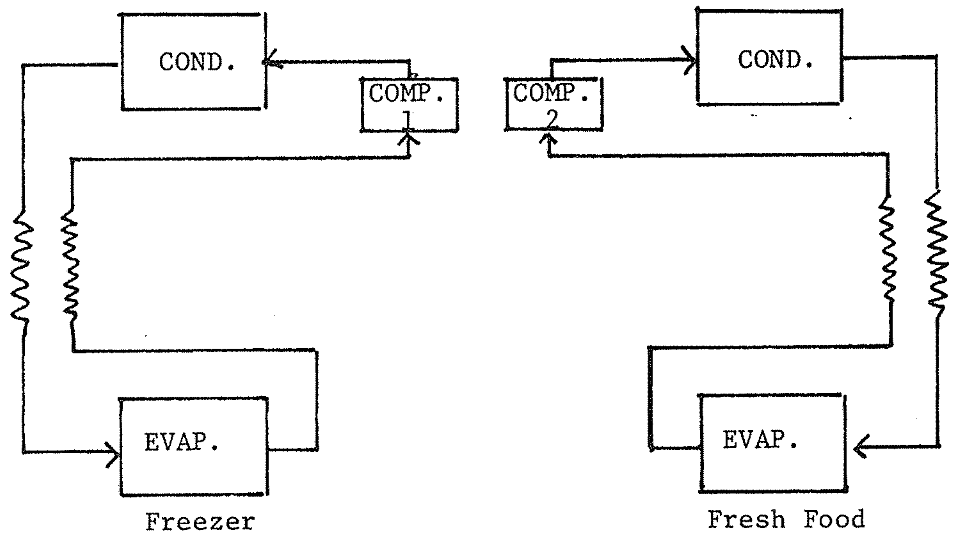
	Consumer Perception Guide			
	Minimum Perceptable	Option Values	Standard Weighting	Option Evaluation
Annual Energy Savings	\$8 ± \$1	7.45	2	+2
First Cost	\$48 ± \$10	50	3	-3
Noise	3 db	0	3	0
Storage	2 cu. ft.	0	3	0
Unit Life	2.7 years	Unknown	1	-1
Rating				-2

ADDITIONAL NOTES

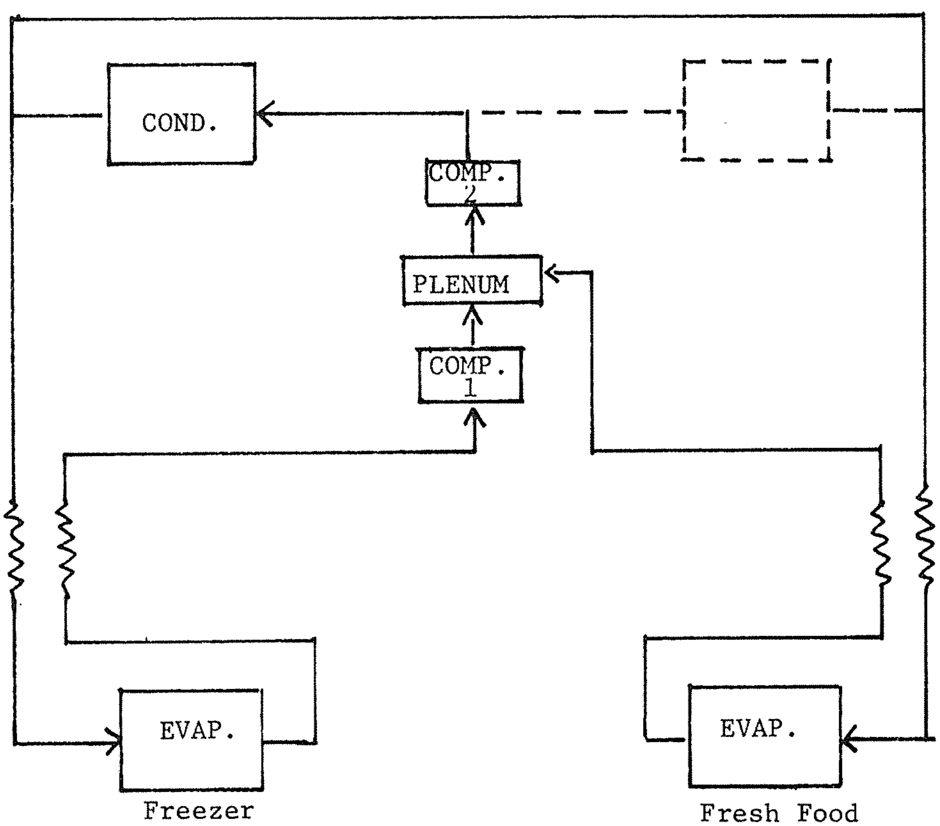
AREAS OF UNCERTAINTY

DESIGN OPTION 12. Attachment

Two Compressor/Evaporator Systems



A. SEPARATE CIRCUITS



B. TWO-STAGE COMPRESSOR

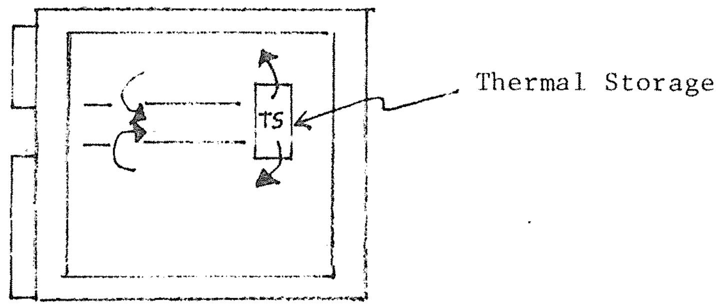
DESIGN OPTION 13.

Thermal Storage

DESCRIPTION

The refrigeration unit is run at night when the ambient temperature is lower and the EER is high--"cold" is stored in a phase change material.

SCHEMATIC



ENERGY PARAMETERS FOR 90°F ROOM

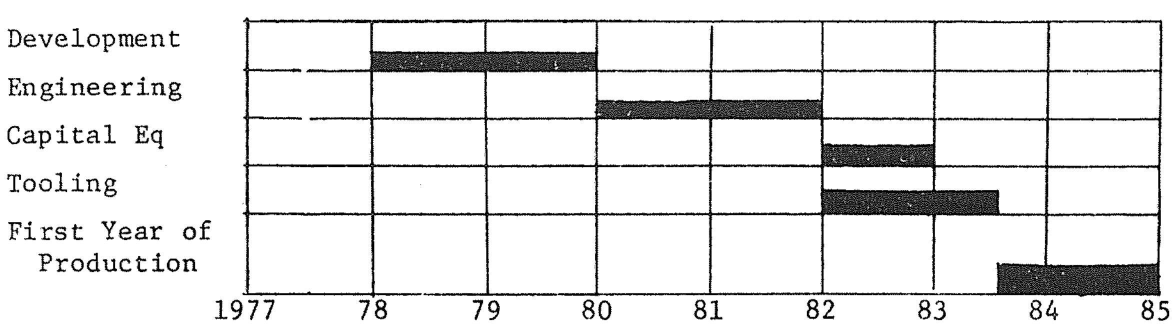
	Q_s Btu/Hr	Q_r Btu/Hr	I Btu/Hr	EER Btu/Hr-Watt	E Watts	Kwh/Day
Baseline NIGHT 80°F	223	670	184	3.679	30.6	2.751
Baseline DAY 90°F	266	653	190	3.555	30.6	
Design Option	266	653	190	3.555	30.6	2.698

SAVINGS .053 Kwh/Day

ECONOMICS

Cost to Manufacture	Cost to Consumer	Annual Energy Savings	Years to Payback
$ <20	$ <38	$1.00 STD Electric Rates	<38
		$11.00 Peak Electric Rates	3.4

LEAD TIMES



OVER

CONSUMER ACCEPTANCE

	Consumer Perception Guide: Minimum Perceptable	Option Values	Standard Weighting	Option Evaluation
Annual Energy Savings	\$8 ± \$1	1, 11	2	0, 2
First Cost	\$48	< 38	3	0
Noise	3 db	0	3	0
Storage	2 ± .5 cu. ft.	1	3	0
Unit Life	2.7 years	0	1	0
Rating				0, 2

ADDITIONAL NOTES		NIGHT	DAY	TOTAL
Experimental Time of Day Rates ¢/kwh		1.4	4.4	
Baseline	Run time, hrs	3.67	9.19	12.86
	E_{in}, Kwh	.781	1.970	2.751 (\$35.63)
Thermal Storage	Run time, hrs	8.0	4.65	12.65
	E_{in}, Kwh	1.701	.997	2.698 (\$24.69)

AREAS OF UNCERTAINTY

1. Typical storage material at -6°F refrigerant temperature and about -3 to 0°F storage temperature could be an ethylene glycol-water mixture. Assuming 4370 Btu/cubic ft. storage, an overnight storage of 2104 Btu would mean .48 cubic ft. of storage material. There would be additional storage volume loss for the container and thermal storage heat exchanger, it could amount to nearly 1 cubic ft. loss of useful food storage.

2. The actual usage pattern in a home may result in larger savings but the 90°F Closed Door Standard Test would not account for any of the anticipated savings.

3. Other thermal storage techniques such as the recondenser design of Altman (University of Pennsylvania) may raise energy savings to 6-10%.

4. The actual value of time of day electric rates and the period of the day falling into the "off peak" category is unknown. Figures used above reflect experimental rates proposed in recent public utilities submissions in Massachusetts.

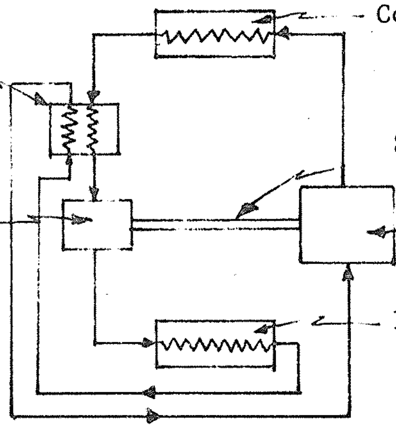
DESIGN OPTION 14.

Mechanical Expander

DESCRIPTION

Replace capillary tube with a mechanical expander--use expander output to assist in compressor drive or to drive fan.

SCHEMATIC



ENERGY PARAMETERS FOR 90°F ROOM

	Q_s Btu/Hr	Q_r Btu/Hr	I Btu/Hr	EER Btu/Hr-Watt	E Watts	Kwh/Day
Baseline	266	658	190	3.558	30.5	2.945
Design Option	266	679	190	3.80	30.5	2.73

SAVINGS .22 Kwh/Day

ECONOMICS

Cost to Manufacture	Cost to Consumer	Annual Energy Savings	Years to Payback
$ Unknown	$ Unknown	$3.21	Unknown

LEAD TIMES

	1977	78	79	80	81	82	83	84
Development								
Engineering								
Capital Eq								
Tooling								
First Year of Production								

85

Beyond Scope of this program

OVER

CONSUMER ACCEPTANCE

	Consumer Perception Guide Minimum Perceptable	Option Values	Standard Weighting	Option Evaluation
Annual Energy Savings	$8	3.21	2	0
First Cost	$48	Unknown	3	0
Noise	3 db	Unknown	3	-3
Storage	2 cu. ft.	0	3	0
Unit Life	2.7 years	Unknown	1	-1
Rating				-4

ADDITIONAL NOTES

AREAS OF UNCERTAINTY

DESIGN OPTION 15.

Other Thermodynamic Cycles

DESCRIPTION

Other thermodynamic cycles have been considered. These include:

Reverse Brayton Cycle (Air)
Reverse Brayton Cycle (Closed Helium)
Stirling Cycle

With high component efficiencies (of 90% or greater) the Air Brayton Cycle may result in a savings of up to 20%. For common machine effectiveness (75% to 85%) there is a loss in energy efficiency as compared to the convention unit. The same is true for the Closed Helium Brayton Cycle.

Stirling Cycle engines are not promising at common refrigerator temperatures. They are more appropriate for temperatures below -50°F.

ENERGY PARAMETERS FOR 90°F ROOM

	Q_s $\frac{Btu}{Hr}$	Q_r $\frac{Btu}{Hr}$	I $\frac{Btu}{Hr}$	EER $\frac{Btu}{Hr\text{-}Watt}$	E Watts	$\frac{Kwh}{Day}$
Baseline	266	658	190	3.558	30.5	2.945
Design Option	See comments above					

ECONOMICS

Cost to Manufacture	Cost to Consumer	Annual Energy Savings	Years to Payback
Unknown	Unknown	Unknown	Unknown

LEAD TIMES

	1977–78	78–79	79–80	80–81	81–82	82–83	83–84	84–85
Development								
Engineering								
Capital Eq								
Tooling								
First Year of Production								

OVER

CONSUMER ACCEPTANCE

	Consumer Perception Guide			Option Evaluation
	Minimum Perceptable	Option Values	Standard Weighting	
Annual Energy Savings	$8		2	
First Cost	$48		3	
Noise	3 db		3	
Storage	2 cu. ft.		3	
Unit Life	2.7 years		1	
Rating				Unknown

ADDITIONAL NOTES

AREAS OF UNCERTAINTY

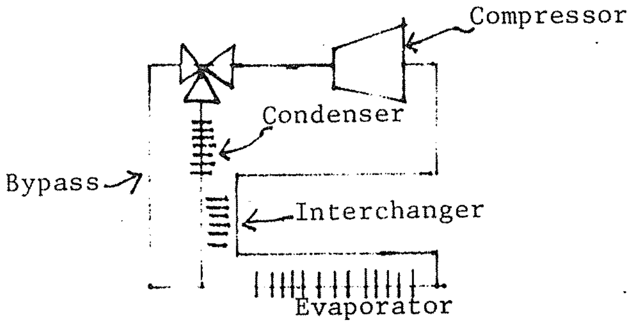
DESIGN OPTION 16.

Hot Gas Defrost

DESCRIPTION

After the 8-hour run interval, a solenoid opens a hot gas bypass from the compressor to the evaporator. Heat is removed from the compressor housing with an effective EER of 5.18 Btu/hr-Watt melting the ice in the evaporator.

SCHEMATIC



ENERGY PARAMETERS FOR 90°F ROOM

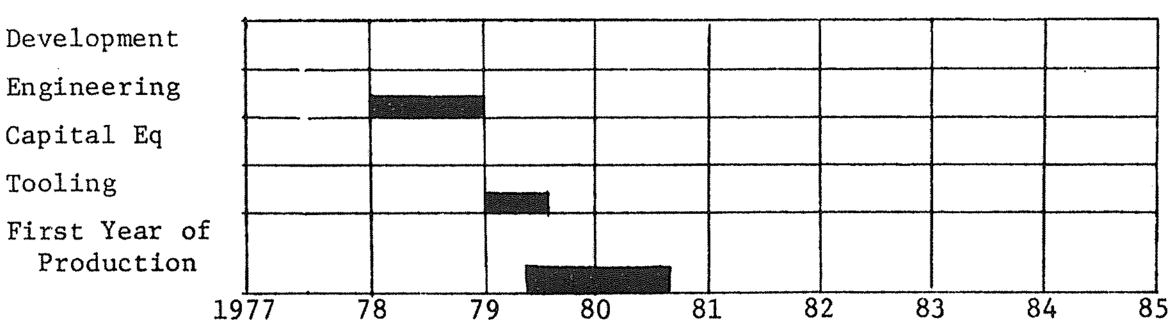
	Q_s Btu/Hr	Q_r Btu/Hr	I Btu/Hr	EER Btu/Hr-Watt	E Watts	Kwh/Day
Baseline	266	658	190	3.558	30.5	2.945
Design Option	266	658	190	3.558	25.1	2.864

SAVINGS: .08 Kwh/Day+

ECONOMICS

Cost to Manufacture	Cost to Consumer	Annual Energy Savings	Years to Payback
$ 10.58	$ 26.45	$ 1.20	22

LEAD TIMES



+Based on AHAM Closed Door Test.

OVER

CONSUMER ACCEPTANCE

	Consumer Perception Guide			Option Evaluation
	Minimum Perceptable	Option Values	Standard Weighting	
Annual Energy Savings	\$8 ± \$1	1	2	0
First Cost	\$48 ± \$10	26	3	0
Noise	3 db	0	3	0
Storage	2 cu. ft.	0	3	0
Unit Life	2.7 years	?	1	-1
Rating				-1

AREAS OF UNCERTAINTY

The return of liquid refrigerant to the compressor in this simple bypass scheme will probably lower the compressor life. A four-way reversing valve and additional capillary tube may be used to provide a heat pump type of circuit which has the "condenser" adding heat before the refrigerant returns to the compressor and should minimize liquid slugging, but double the initial cost.

DESIGN OPTION 17.

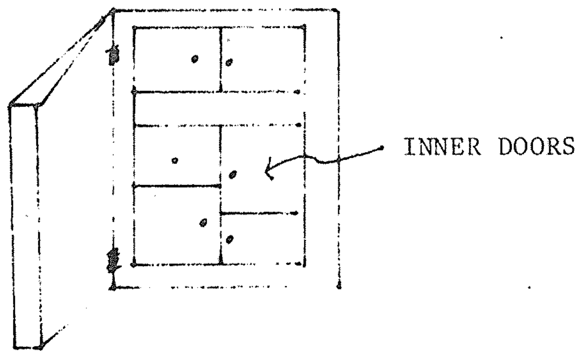
Inner Doors

DESCRIPTION

Several inner doors are added to :

1. Inhibit air exchange during door openings.
2. Reduce gasket air leakage.

SCHEMATIC



ENERGY PARAMETERS FOR 90°F ROOM

	Q_s Btu/Hr	Q_r Btu/Hr	I Btu/Hr	EER Btu/Hr-Watt	E Watts	Kwh/Day
Baseline	266	658	190	3.558	30.5	2.945
Design Option	In a 90°CD test, energy savings is same as Option 3--Improved Door Seal but at a much greater cost.					

ECONOMICS

Cost to Manufacture	Cost to Consumer	Annual Energy Savings	Years to Payback
Design Option 3 makes this concept obsolete in a 90°F Closed Door test.			

LEAD TIMES

	1977–78	78–79	79–80	80–81	81–82	82–83	83–84	84–85
Development								
Engineering								
Capital Eq								
Tooling								
First Year of Production								

1977 78 79 80 81 82 83 84 85

OVER

CONSUMER ACCEPTANCE

	Consumer Perception Guide Minimum Perceptable	Option Values	Standard Weighting	Option Evaluation
Annual Energy Savings	$8			
First Cost	$48			
Noise	3 db			
Storage	2 cu. ft.			
Unit Life	2.7 years			
Rating				Not evaluated

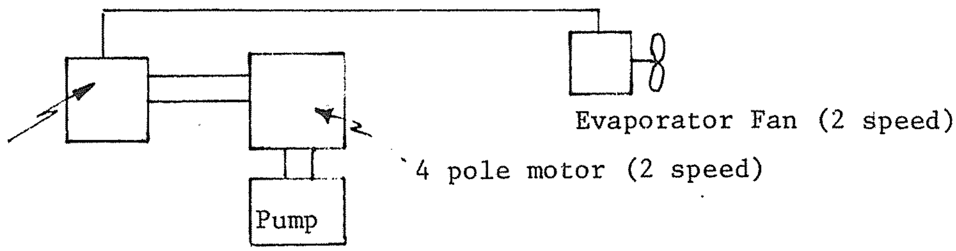
DESIGN OPTION 18.

Two-Speed Compressor

DESCRIPTION

Lower speed compressor operation during shorter run times (lower room ambient) could take advantage of the impact (by an estimated 12%) efficiency of the pump at lower speeds. A four pole motor run as both a two pole and four pole winding with appropriate capacitors could be used. A two-speed evaporator fan may also be required to achieve energy savings.

SCHEMATIC



ENERGY PARAMETERS FOR 90°F ROOM

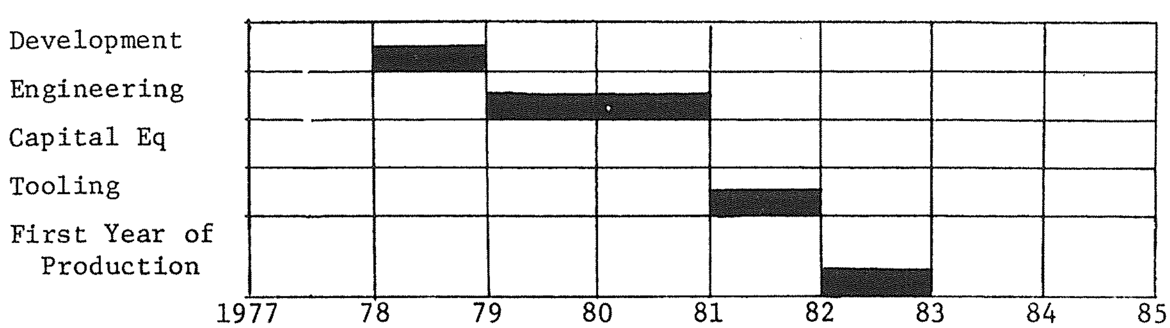
	Q_s Btu/Hr	Q_r Btu/Hr	I Btu/Hr	EER Btu/Hr-Watt	E Watts	Kwh/Day
Baseline	266	658	190	3.558	30.5	2.945
Design Option	Assuming ½ the time at ½ the speed = 6% savings					2.765

SAVINGS .18 Kwh/Day

ECONOMICS

Cost to Manufacture	Cost to Consumer	Annual Energy Savings	Years to Payback
$ <20	$ <38	$ 2.63	<15

LEAD TIMES



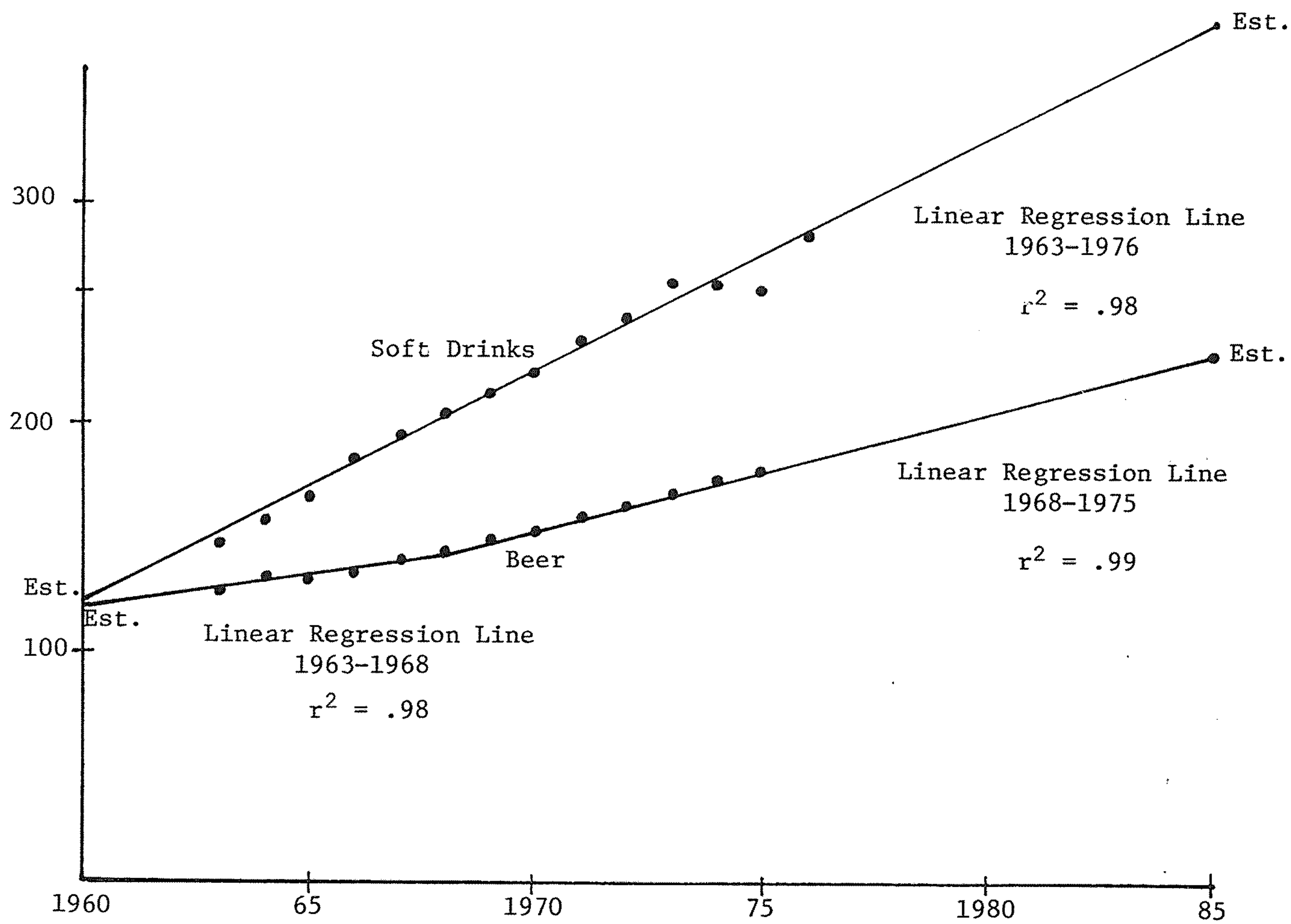
OVER

CONSUMER ACCEPTANCE

	Consumer Perception Guide			
	Minimum Perceptable	Option Values	Standard Weighting	Option Evaluation
Annual Energy Savings	\$8 ± \$1	2.63	2	0
First Cost	\$48 ± \$10	< 38	3	0
Noise	3 db	0	3	0
Storage	2 cu. ft.	0	3	0
Unit Life	2.7 years	0	1	0
Rating				0

ADDITIONAL NOTES

AREAS OF UNCERTAINTY



APPENDIX A STRAIGHT LINE PROJECTION OF BEER AND SOFT DRINK CONSUMPTION

DEVELOPMENT OF A HIGH EFFICIENCY, AUTOMATIC DEFROSTING REFRIGERATOR/FREEZER

TASK 3 REPORT

PROTOTYPE DESIGN AND TESTING

Arthur D Little, Inc.

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PROTOTYPE DESIGN AND TESTING

INTRODUCTION

In Task 2, a list of eighteen design options were examined for technical feasibility, energy savings and likely consumer acceptance. Seven (7) of options, including a hot water delivery feature, showed considerable promise. During the early stages of Task 2 it became apparent that the hot water feature had areas of uncertainty while new advantages from a concept (thermostatic expansion valve) not selected in Task 2 were identified. The hot water feature was dropped and the concepts considered in Task 3 were:

1) optimized insulation,
2) improved gaskets,
3) alternative condenser design,
4) new evaporator,
5) improved fan system,
6) improved defrost control, and
   thermostatic expansion valve.

Eight individual design studies involving both computer simulation and laboratory preprototype testing were undertaken to better understand the benefits and design trade-offs for the design concepts under examination. The discrete concept evaluation studies are outlined in Table 1. The approach and results for each of these analyses is summarized in the first section of this Task Report. In the second section, the design of the Phase I prototype and the results of testing are presented.

TABLE 1

Pre-Prototype Studies

	Study	Activity	Concept Under Study
1.	Baseline Analysis	Model and test of Amana ESRF 16	none
2.	Double Gasket	Two laboratory tests and analysis	# 2
3.	Insulation Optimization	Computer Modeling	# 1
4.	Fan Tests	Wind tunnel tests and analysis	# 5
5.	Bench Test	Model Validation	NA
6.	Bench Tests	Thermostatic expansion valve	# 6
7.	Design Guidance	Combined design options	# 3, 4, 5, 6
8.	Calorimeter Tests	Cabinet heat flow	# 1

1. PRE-PROTOTYPE ANALYSIS AND TESTING

The following eight subsections include the purpose, approach and findings of each of the individual design studies.

1.1 Baseline Analysis

1.1.1 Purpose

Experimentally, isolate cabinet heat flow partitioning.

1.1.2 Approach

The ESRFC3-16 (3" thick wall) production model was used as a baseline. The heat flow through the parts of the cabinet was determined by analysis and gasket area tests. The refrigeration unit was modelled and used with the cabinet data to simulate cabinet performance. The predicted results were compared with the data.

The thermal load in the baseline refrigerator is comprised of heat flow through the following major elements:

- Plane walls and corners
- Wedge
- Flange
- Infiltration
- Hot wall condenser

Plane Walls and Corners

The plane (parallel) walls, including corners, accounts for most of the thermal load on the refrigeration unit. A computer program was developed to calculate the thermal load based on outside cabinet dimensions. The program corrects for corner effects based on the method of Langmuir.[1]

The program was used to calculate the thermal loads for the baseline 16-cubic-foot ESRF refrigerator for the nominal dimensions of Figure 1. The results were:

[1] Langmuir, I., E. Adams, and S. Meike, "Flow of Heat Through Furnace Walls: The Shape Factor," Trans American Electrochemical Society, 24:54-84, 1913.

TABLE 2

Plane Wall and Corner Head Load - ESRF C3-16

Compartment	Heat Load (Btu/hr)
Freezer	50.3
Mullion	11.1
Fresh-food	111.1

Wedge

The cabinet wall must taper at the door to permit interference-free opening. The heat flow in the taper is a complex flow. Through a transformation of coordinates, a simple equation governing the wedge conduction was developed. This equation is given below for the geometry of Figure 2. Please see Appendix for details.

$$\text{Wedge Heat Flow} = k\,\ell\,\frac{\ln a/b}{\alpha}\left(T_{\text{room}} - T_{\substack{\text{inside}\\ \text{cabinet}}}\right)$$

For the baseline case (assuming fiberglass insulation in the wedge area), the wedge conduction amounted to:

Wedge conduction = 30.2 Btu/hr

Flange and Gasket

The flange heat leak is the conduction of heat across the flange metal. This heat flow is affected by the evaporator fan air motion and therefore differs in the fan on and fan off conditions.

A flange test* apparatus was built and tests with four element thermopiles were made on the refrigerator unit. The experimental arrangement is shown in Figure 2.

The test data for the model showed the following:

TABLE 3

	Heat Flow in Btu/Hr		
	Total	Fresh Food	Freezer
Thermocouple-Flange Heat Load with fan on	68.1	9.1	59.0
Thermocouple-Flange Heat Load without fan	43.5	11.3	32.2

*These tests were performed in the ADL test facility.

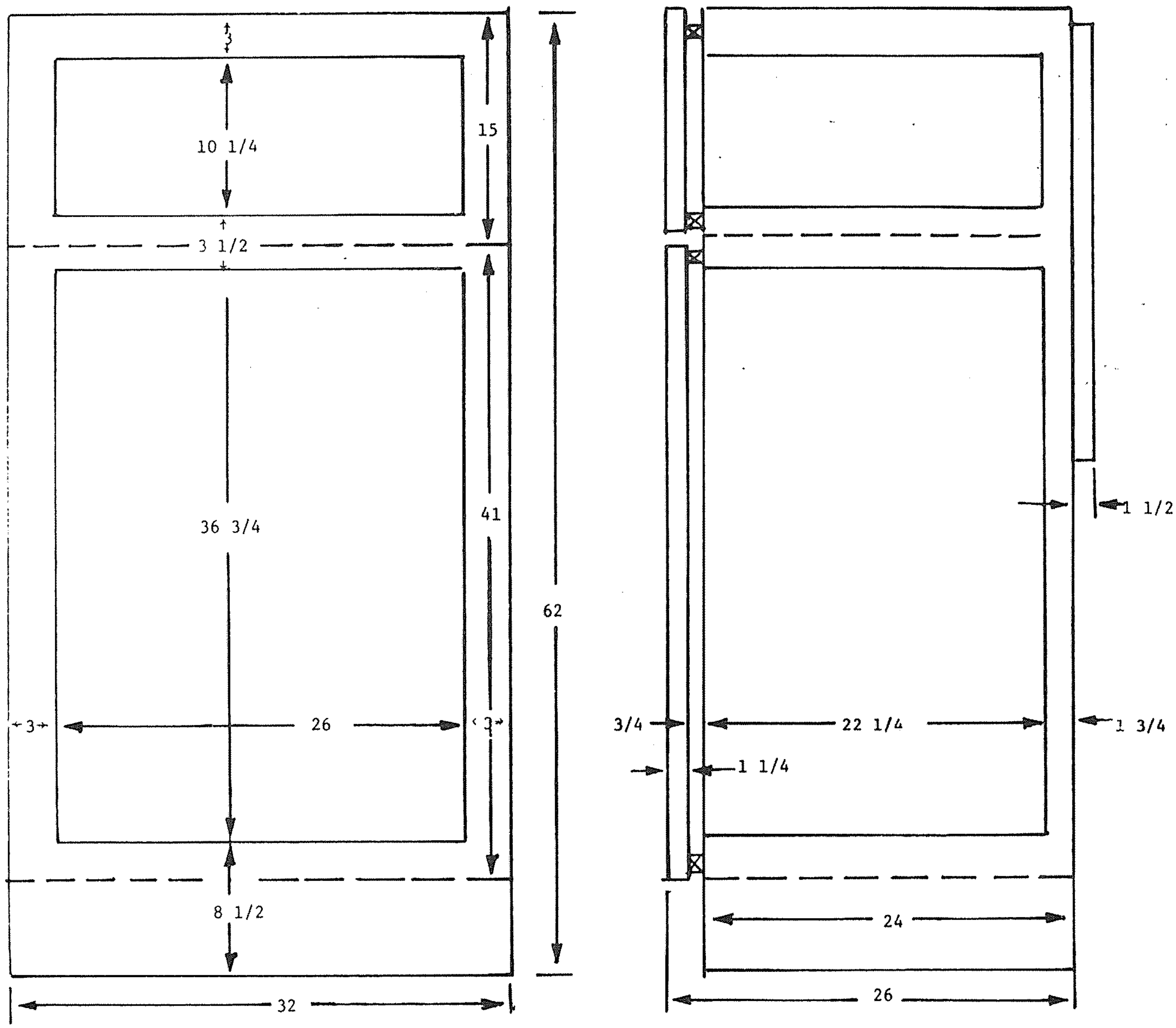

FIGURE 1
ESRFC3-16
OUTER CABINET DIMENSIONS
(All in Inches)

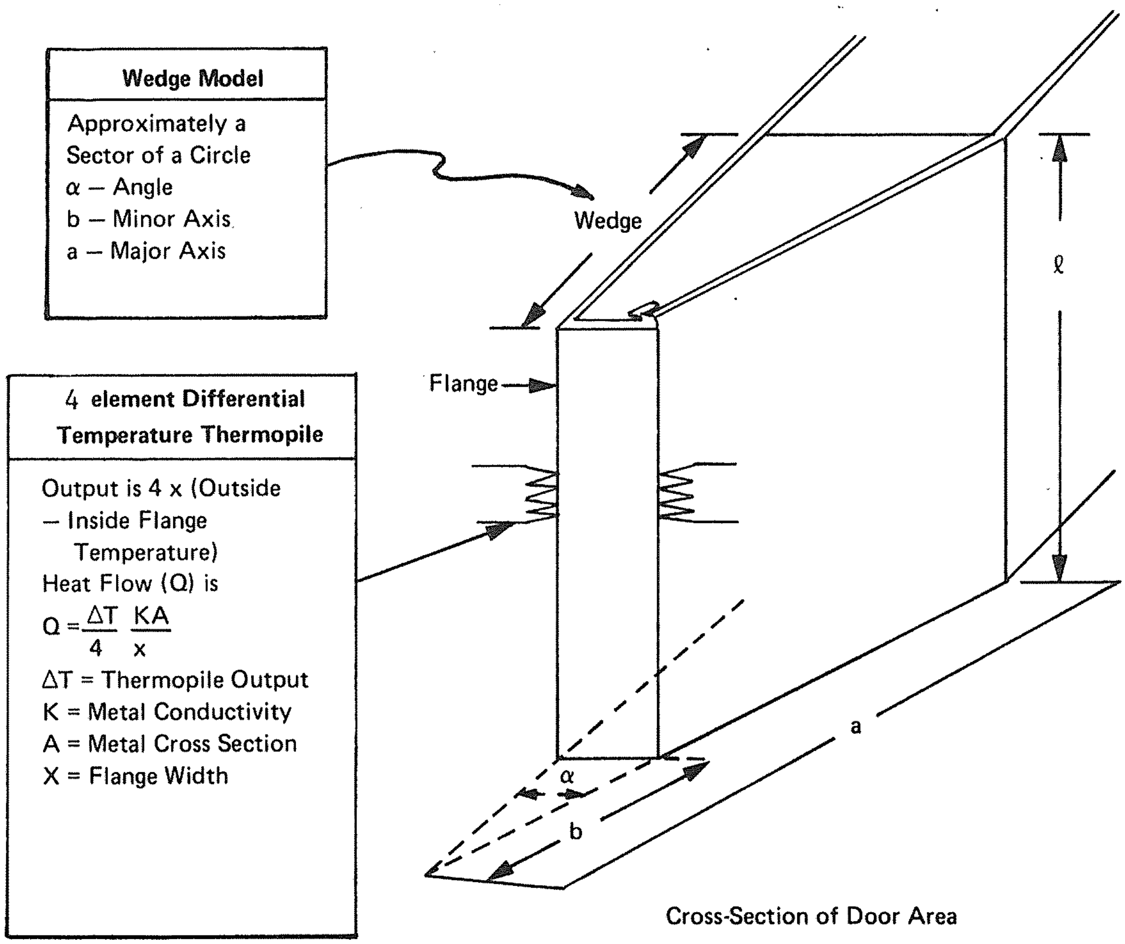

FIGURE 2 FLANGE AND WEDGE SCHEMATIC

These values are slightly higher than that predicted by subtracting the calculated wall heat load from the laboratory test cabinet heat load calorimeter value (see Section 1.8) which yielded 62.5 Btu/hr. vs. 68.1 Btu/hr. for the fan on condition. The lower (62.5) value is more representative because we were unable to separate all of the hot wall heat flux component from the flange heat flow in our tests, leading to the following representation of flange heat data:

Flange Heat Design Values For Baseline Unit

	Heat Flow	Conductance* Fresh Food	$\frac{Btu}{Hr. °F ft.}$ Freezer
Flange Heat Load with Fan	62.5 Btu/hr.	.022	.092
Flange Heat Load without Fan	43.5 Btu/hr.	.028	.055

Infiltration

Infiltration of moisture-laden air was thought to account for some of the thermal load on the unit. Two tests were performed to isolate the effect of air/moisture flow through the gasket area.

In a test of an 18 cubic foot unit, the entire door area was taped tightly, and the unit energy consumption was compared to the unit energy consumption without tape. The findings are shown in Table 3. The conclusion is that infiltration in the baseline unit in a closed door test is not a significant heat load as the effect on run time is only 1 - 2%.

In another test in a 90°F room, a Production ESRFC3-16 cubic foot refrigerator was run continuously for 16 days, and 250 ml. of defrost water was collected; equivalent to a steady infiltration flow of 5 Btu/hr., once again indicating that infiltration heat flow is insignificant.

Hot Wall Condenser

In recent years, the hot wall condenser used in chest freezers has been applied to the refrigerator-freezer. The condenser tubes are clamped to the inside of the outer cabinet shell. The space normally required for the conventional back-mounted condenser is considerably reduced, though additional wall clearance is required on the side walls to assure sufficient air circulation. A section of the hot wall condenser seen from the inside of the outer shell is shown in Figure 3.

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* Based on 10.548' gasket length in the fresh food compartment and 6.8333' in the freezer.

TABLE 4

Measured Cabinet Infiltration in a 90°F Room
(No Door Openings)

	Untaped Doors	Taped Doors
16-Cubic-Foot Unit		
Percent Run Time	48.6	47.9
Daily Energy Consumption (kwh/day)	2.9	2.88
Average Freezer Temperature	4.5°F	4.5°F
Average Fresh-Food Temperature	39°F	38.5°F
18-Cubic-Foot Unit		
Percent Run Time	43.8	43.2
Daily Energy Consumption (kwh/day)	3.82	3.75
Average Freezer Temperature	3.2°F	2.8°F
Average Fresh-Food Temperature	38°F	37.8°F

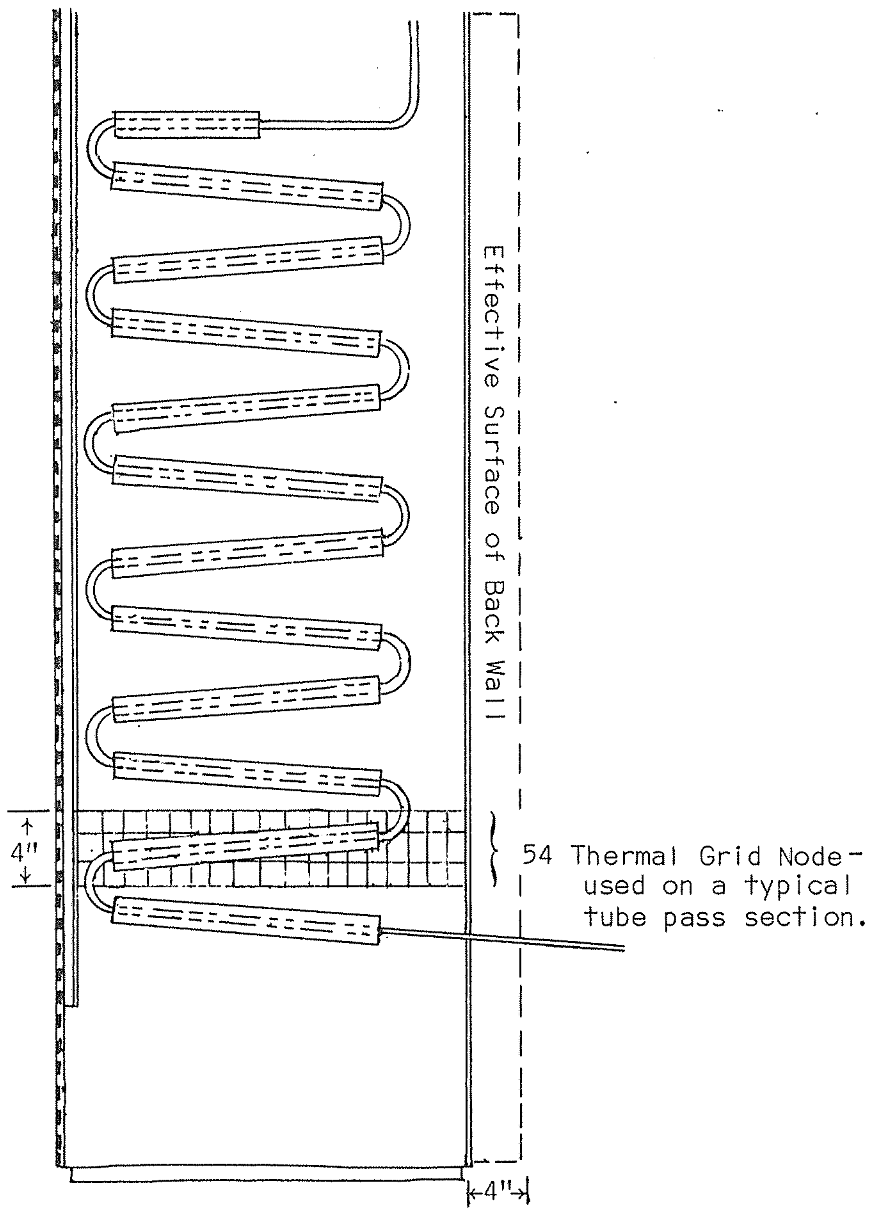


FIGURE 3 VIEW OF HOT WALL CONDENSER SIDE WALL

A thermal analysis was performed to establish the partitioning of heat flow from the hot wall condenser. The analysis consisted of a 54 node finite difference analysis on one of the tube sections. The results of the analysis are shown in Figure 4. Of the 778 Btu/hr of condenser heat rejection, some 67 Btu/hr is (undesirably) returned to the cold space.

In summary, the cabinet heat flow is:

TABLE 5

Cabinet Heat Flow Partitioning

BASELINE	ESFR-16C	HEAT FLOW	BTU/hr
Heat Load Component	Freezer	Fresh Food	Total
wall and door	50.3	111.1	161.4
mullion	11.1	-	11.1
wedge	10.0	20.2	30.2
flange (fan on)	53.5	9.0	62.5
Subtotal	124.9	140.3	265.2
flange (fan off)	32.2	11.3	43.5
hot wall condenser	31.7	35.2	66.9

The cabinet heat flow partitioning together with an estimated refrigeration capacity can be used to predict the run time of the Baseline unit in the 90°F room test for comparison with test data (see Findings).

The refrigeration unit consists of:

TABLE 6

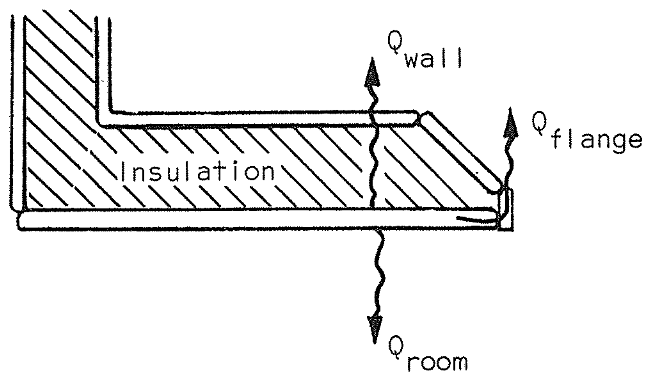
Refrigeration Unit

- A 20.5" x 10" x 2" evaporator (UA = 64.89 Btu/hr°F)
- A 38.3 CFM evaporator air flow.
- A Tecumseh AE1360A compressor
- Hot wall condenser

The estimated refrigeration capacity is:

$\dot{Q}$ refrigeration = 626.5 Btu/hr*

*Based on the measured evaporator operating temperature in a 90°F room test of - 14°F.



	Btu/Hour
Heat to Room	711
Heat through Flange to Cold Space	48.7
Heat through Wall Insulation to Cold Space	18.2

FIGURE 4 PARTITIONING OF HOT WALL CONDENSER HEAT FLOW

1.1.3 Findings

The predicted run time based on the calculated heat flow and refrigeration capacity is:

$$\begin{array}{l}\%\text{ Run Time} \\ \text{Std 3" Foam Production} \\ \text{Model}\end{array} = \frac{\overset{\text{walls}}{202.7} + \overset{\text{flange}}{43.5}}{\underset{\substack{\text{refrigeration} \\ \text{capacity}}}{626.5} - \underset{\substack{\text{hot} \\ \text{wall} \\ \text{condenser} \\ \text{effect}}}{66.9} - \underset{\substack{\text{evapor-} \\ \text{ator fan} \\ \text{motor losses}}}{51.5} - \underset{\text{heaters}}{52.8} - \underset{\substack{\text{incremental load} \\ \text{for flange with} \\ \text{fan on}}}{19}} = \frac{246.2}{436.3} = 56.4\%$$

which compares with the measured percent run time of 54.3%.* The heat additions to the standard unit were calculated as:

TABLE 7

Heater Inputs

	gross watts	Net internal heat load in Btu/hr.
Mullion	11	18.75
Liner Top	10	34.1
Defrost (none used)	0	0
TOTAL	21 watts	52.8 Btu/hr.

* ESRFC3-16 Test June 28, 1978 in a 90.5° F Room.

1.2 Double Gasket

1.2.1 Purpose

Evaluate the potential for energy savings with additional door gaskets.

1.2.2 Approach

A series of tests were performed to evaluate the savings potential of improved door gaskets to prevent air convection in the throat area shown in Figure 5. In the first tests the throat was sealed as well as possible with large soft urethane seals. In the second tests, (pilot) vinyl gaskets were used. The test results were:

Table 8 *

Throat Gasket Test Results

		MEASURED		CALCULATED		
		% Run Time	Evaporator Temperature °F	Refrigeration Capacity (Btu/hr)	% Run Time	% Flange Heat Elimination
A	Std single gasket	48.2	-11.5	699.7	48.2	N.A.
B	Seal freezer only	45.5	-13.5	660.8	43.6	48.4
C	Seal fresh food only	48.0	-13.0	670.5	49.0	0
D	Seal both	44.0	-12.5	680.2	39.7	57.9
E	Vinyl-gasket, freezer only	47.6	-13.0	670.5	42.7	42.7

The influence of the gaskets on the run time can be thought of as a reduction of the flange heat leak. In the limit, the complete elimination of the flange heat leak should result in a reduction of unit run time to:

$$\text{\% run time through elimination of flange heat leak} = \frac{202.7}{(680.2)-(66.9)-(103)} = 39.7\%$$

The measured percent run time for the test with both compartments sealed was 44%, which would be equivalent to a partial reduction of the flange heat to 58% of the minimum expected value.

*Compressor - AE 1380; 90°F Room 5 and 38 Cabinet, 2/2/78

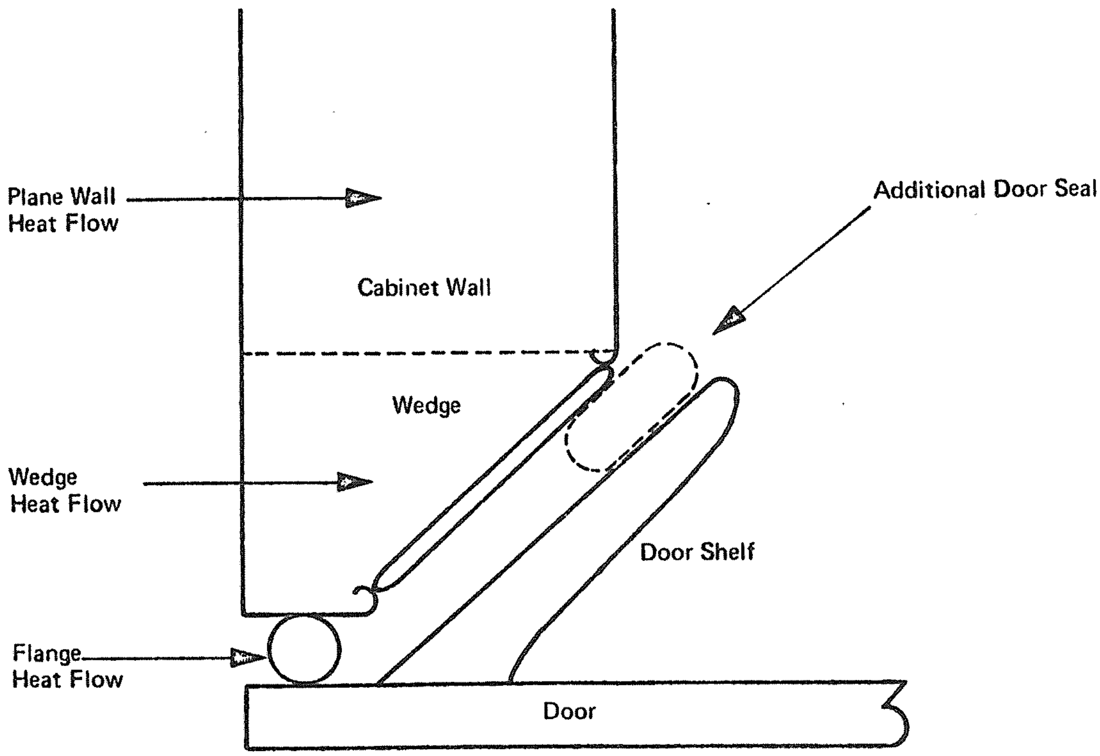


FIGURE 5 CROSS-SECTION OF DOOR CLOSURE AREA

1.2.3 Findings

Only the freezer compartment double gasket (B) effectively reduces the unit energy consumption, and should be incorporated into the prototype. The air flow in the freezer is approximately six times that of the fresh food compartment causing greater flange area heat flow there.

A prototype vinyl throat gasket was tested (E). While not achieving the energy reduction of the completely blocked throat test (B), this practical application of the concept does look promising, and does reduce the throat heat flow. It will save about $1.90 of electric cost per year at a fraction of a dollar cost increase.

The vinyl gasket appears to reduce the calculated gasket area heat load to 0.427 of its normal value (see Table 7). Applying this factor to the freezer gasket conductances (page 6) gives anticipated flange/gasket conductance with a vinyl secondary gasket.

Table 9

Flange Conductance

	Conductance $\frac{BTU}{Hr.\ ft.\ ^\circ F}$ Fresh Food Single Gasket	Freezer Double Gasket
Flange conductance with fan on	.022	.039
Flange conductance with fan off	.028	.0235

The predicted heat loads to be used in component sizing analysis for the Phase I prototype are:

Table 10

Predicted Flange Heat Flow

	Fresh Food Btu/hr	Freezer Btu/hr	Total Btu/hr
Flange heat flow with fan on	13.2	28.7	41.9
Flange heat flow with fan off	13.1	15.6	28.8

1.3 Optimized Insulation

1.3.1 Purpose

Develop cabinet outer dimension limitations and determine polyurethane foam insulation configuration to minimize heat leak.

1.3.2 Approach

Outer cabinet dimensional limits were established by applying the following criteria:

- A single insulation formula (doors & wall thicknesses) was to be carried through the 16, 18, 20, 23 cubic foot family
- The width would be 32" for the 16 and 18 cubic foot units
- The maximum dimensions of any unit would be:

  height 67"
  width 35"
  depth 31"
- The freezer to fresh food volume ratio (see Table 10) will be the same as existing models.
- Standard internal volume deductions for nonusable storage volume would be applied to the volume calculation

Computer analysis of numerous insulations systems led to the system offering the minimum heat flow while meeting the design criteria. A system of 2.5" thick doors, 3.0" thick walls in the freezer unit, 2.25" thick walls in the fresh food compartment was found to meet the criteria while providing the minimum heat flow. The cabinet dimensions are given in Table 10.

1.3.3 Findings

Table 11 summarizes the optimum insulation designs. The insulation thickness of the several walls are given along with the heat flow through each. Cabinet storage volumes are calculated with and without deductable (unusable space defined by ANSI standard) adjustments.

TABLE 11

Dimensions for 2.25/3.0*Series

Unit Size in Cubic Feet	Dimensions in Inches Height	Width	Depth	Freezer Volume / Total Volume
16	65.00	32	27.00	.23
18	67.25	32	28.75	.24
20	67.25	32	31.50	.28
23	67.00	35	30.75	.284

* 2.25 inch Fresh food walls, 3.0 in Freezer and 2.5 inch Door.

Table 12

Cabinet Specifications Prototype Series

Cabinet Size	Insulation Thickness (Inches)							Heat Flow (Btu/hr)						Volumes (cu ft) Unadjusted		Adjusted			
	Freezer				Fresh Food														
	Side	Front	Back	Top	Side	Front	Back	Freezer	Mullion	Fresh Food	Gasket	Wedge	Total	Freezer	Fresh Food	Freezer	Fresh Food	Total	Ratio ($\frac{Frz.}{Total}$)
16 cubic foot Prototype	3.00	2.50	3.00	3.00	2.25	2.50	2.25	43.08	13.28	68.39	41.9	10.06	176.7	3.88	12.66	3.60	12.22	15.82	0.227
18 cubic foot	3.00	2.50	3.00	3.00	2.25	2.50	2.25	47.89	13.94	73.34	40.03	20.62	195.82	4.55	14.13	4.35	13.72	18.07	0.241
20 cubic foot	3.00	2.50	3.00	3.00	2.25	2.50	2.25	56.89	14.98	73.71	40.37	19.81	205.76	5.87	14.90	5.67	14.49	20.16	0.281
23 cubic foot	3.00	2.50	3.00	3.00	2.25	2.50	2.25	58.46	15.28	77.23	56.28	19.85	227:103	6.48	16.45	6.43	16.00	22.42	0.283

1.4 Fan Test

1.4.1 Purpose

Develop the optimum air flow design to reduce unit energy consumption.

1.4.2 Approach

Two analyses were undertaken. In the first, a fan-motor unit optimization was performed to arrive at the optimum air flow rate. A second task was performed in the laboratory to achieve this air flow rate with the minimum fan motor power.

Optimum Evaporator Airflow

There is an optimum fan airflow rate for minimum refrigerator energy consumption. As airflow decreases from that value, effective refrigeration decreases and as airflow increases, rising fan power (which puts normal energy into the air stream) also causes net system refrigeration to decrease. The actual shape of the energy consumption versus airflow curve determines the range of acceptable airflow rates for a refrigerator design.

The following analysis assumes:

- all fan and motor power is absorbed in the cold space,
- a constant heat exchanger effectiveness,
- pressure drop (system resistance) based on pipe analysis,
- a constant room temperature,
- a constant condenser temperature.

The refrigerator energy consumption can be written as:

$$\frac{\text{KWH}}{\text{DAY}} = 0.024 \quad Qs \left[\frac{Pc + Pf}{Qr - Pf}\right] \qquad \text{(Quantities in watts)} \qquad (1)$$

where

Pc = Compressor power input
Pf = Fan-motor input $\propto$ (air mass flow)^3
Qr = Refrigeration capacity
Qs = Steady cabinet heat load

An expression for the power consumption for the standard 600 btu/hr* compressor was developed in Task 2.

$$P_c = 3.184\ T_e + 203.49 \text{ (watts)} \quad \text{(See page 21 of Task 2 Report)} \quad (2)$$

$$P_f = K\ \dot{M}_a^{\ 3} \quad \text{where 12 watts} = K\ (35\text{CFM x .08 lbm/ft}^3) \text{ for the standard air flow configuration} \quad (3)$$

$$= 0.546\ \dot{M}_a^{\ 3}$$

$$Q_r = \dot{M}_a\ C_p\ E\ (T_a - T_e) \quad (4)$$

where

T_a = evaporator inlet air temperature

T_e = evaporator (refrigerant) temperature

E = evaporator heat exchanger effectiveness (typically E = .75)

$\dot{M}_a$ = mass flow rate of air in lbm/hr

Also given in the Task 2 was an equation for the compressor mass flow rate ($\dot{M}_r$) which is used to calculate the refrigeration capacity (Q_r).

$$Q_r = \Delta h_r \times \dot{M}_r \quad (5)$$

where

Δh_r = refrigerant cooling capacity

= 53.4 Btu/lbm for a condensing temperature of 115°F (See page 21 of Task 2 Report)

$\dot{M}_r$ = refrigerant mass flow rate

= .325 T_e + 14.25 (lbm/hr) for a AE 1360 at 115°F condensing temperature

if relations (1), (2), (3), (4) are combined, then

$$\frac{\text{kwh/day}}{.024\ Q_s} = \frac{(3.184)T_e + 203.49 + (0.546)\ \dot{M}_a^{\ 3}}{\dot{M}_a\ c_p E\ (T_a - T_e) - (0.546)\ \dot{M}_a^{\ 3}}\text{, and} \quad (6)$$

*A Tecumseh AE 1360 compressor

equating (4) and (5):

$$T_e = \frac{\dot{M}_a\ C_p\ E\ T_a - (14.24)\ \Delta h_r}{\dot{M}_a\ C_p\ E + (0.326)\ \Delta h_r} \tag{7}$$

These relations were used to determine the optimum air flow rate which was found to be in the range of 3.3 lbm/min (38 CFM). Excerpts from the analysis near the optimum range are given in the table below.

TABLE 13

Fan Air Flow Optimization

$\dot{M}_a$ in lbm/min.	T_e in °F	$\frac{\text{kwh/day}}{.024\ Qs}$
3.0	-9.0	1.15
3.1	-8.37	1.11
3.3	-8.0	1.07
3.5	-7.0	1.13
4.0	-5.7	1.18

Fan Air Flow Tests

The air path and fan characteristics to provide the desired air flow (38-41 CFM) with a minimum of fan power were determined with a bench test. Two areas were examined:

- Reduce air flow resistance
- Improve fan-motor efficiency

Rear Spacing

The effect on system airflow of the spacing (Figure 6) between the fan and the evaporator was investigated. The results with and without the tapered entrance contour are shown in Figure 7. It can be seen that rear spacing has little effect on system airflow and the tapered entrance has no effect. The 3-inch rear spacing used in the current Amana design is adequate and was used in the remaining tests.

Intake Air Configuration

The inlet passage design (see Figure 8) was examined. A horizontal slot was varied from a 3 square inch inlet area to 48 square inches. An angled slot of 3.9 square inches was tested, as well as a baffle . A series of slot and baffle combinations were tested, and the findings are summarized in Figure 9. A step (the baffle) is desired to provide space in the fresh food compartment for a light, and a multi-slotted, minimum area intake is desired for strength, appearance, and to prevent items from falling into the mullion space.

The effect of system CFM per watt on the refrigerator-freezer unit energy consumption can be seen in Figure 10. An increase in energy consumption of about 14% separates the worst (4 square inch slot area) from the best (18 square inch slot area). This dramatic effect on energy consumption is due to the compounding effect of the evaporator fan energy which adds to the run time by increasing the refrigeration load and to the running watts.

Alternative Fan/Motor Designs

The following concepts were tested as possible energy-saving designs:

Fan	Motor
Tube Axial Fan	Integral Motor
Reverse Curve Centrifugal Fan	Shaded Pole Motor
Standard Propeller Fan	Permanent Split (Capacitor) Motor

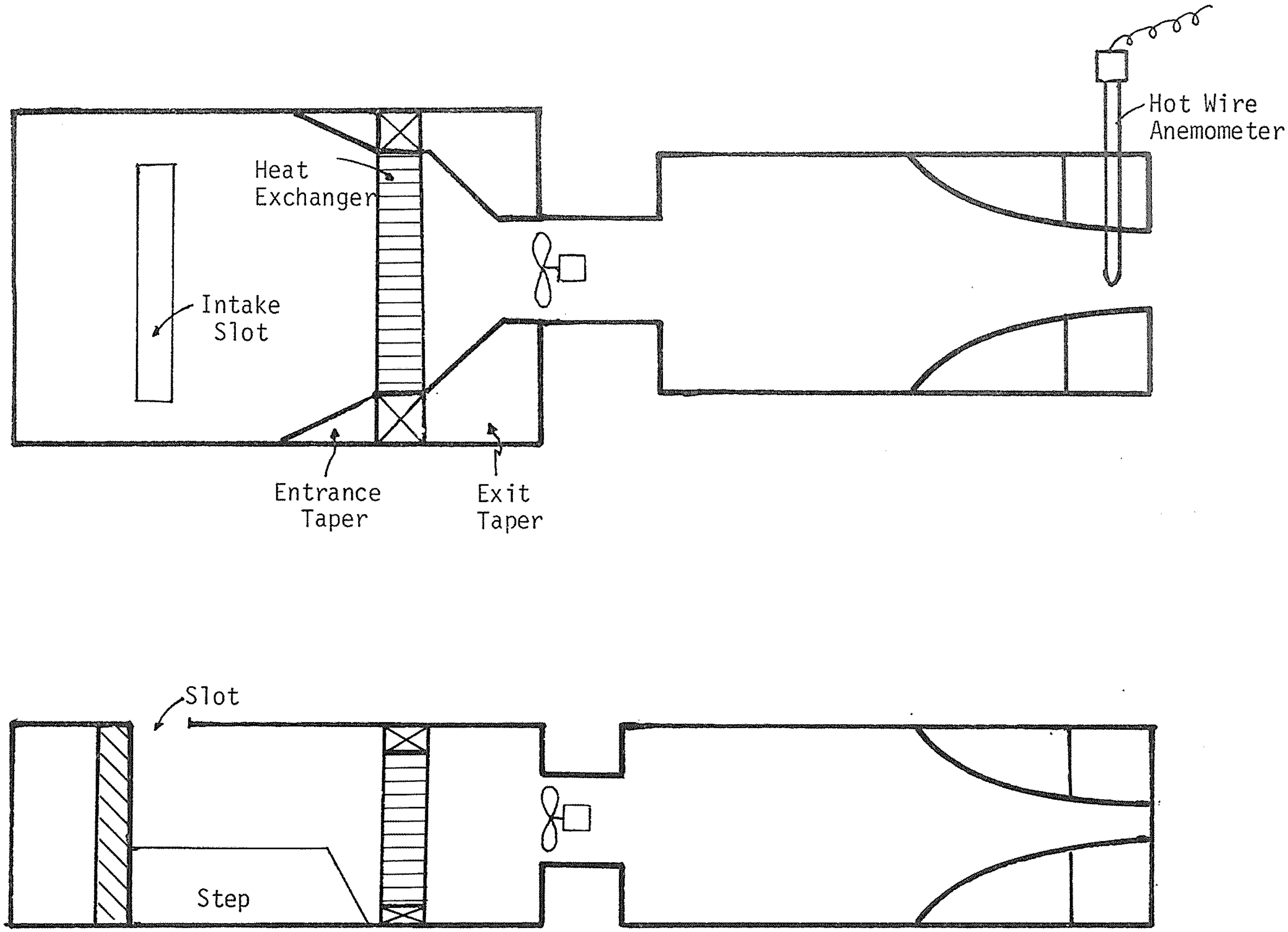


FIGURE 6. AIRFLOW TEST LAYOUT

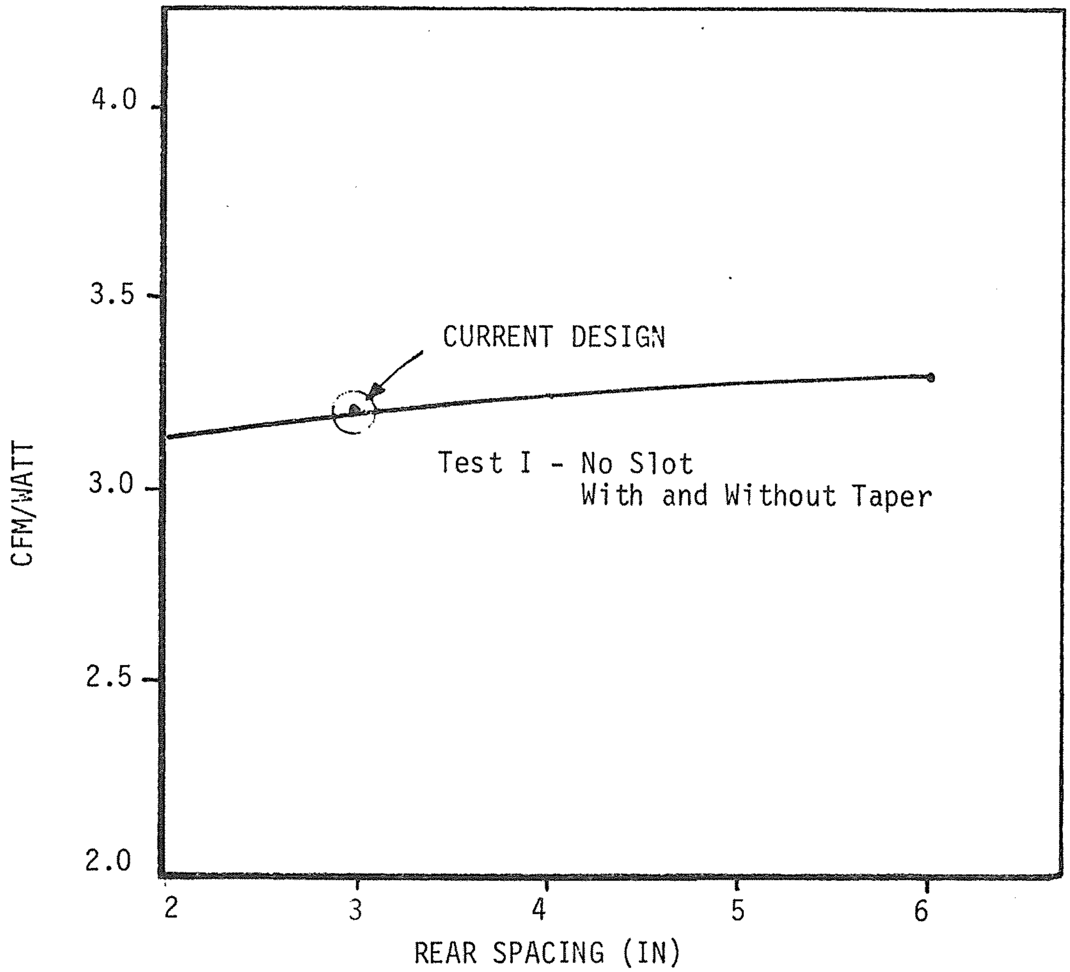


FIGURE 7. EFFECT OF SPACING BETWEEN FAN AND EVAPORATOR ON SYSTEM AIRFLOW

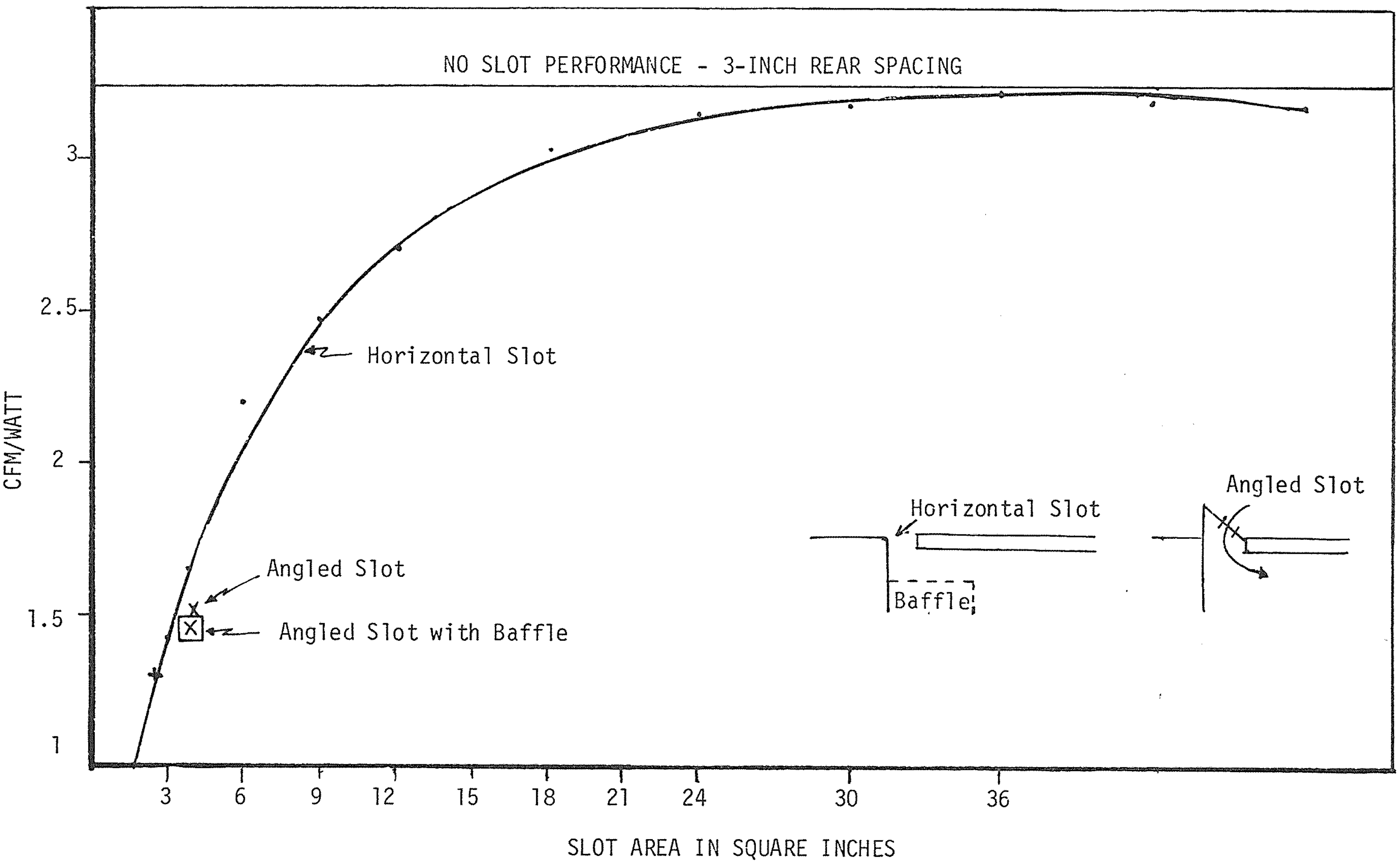
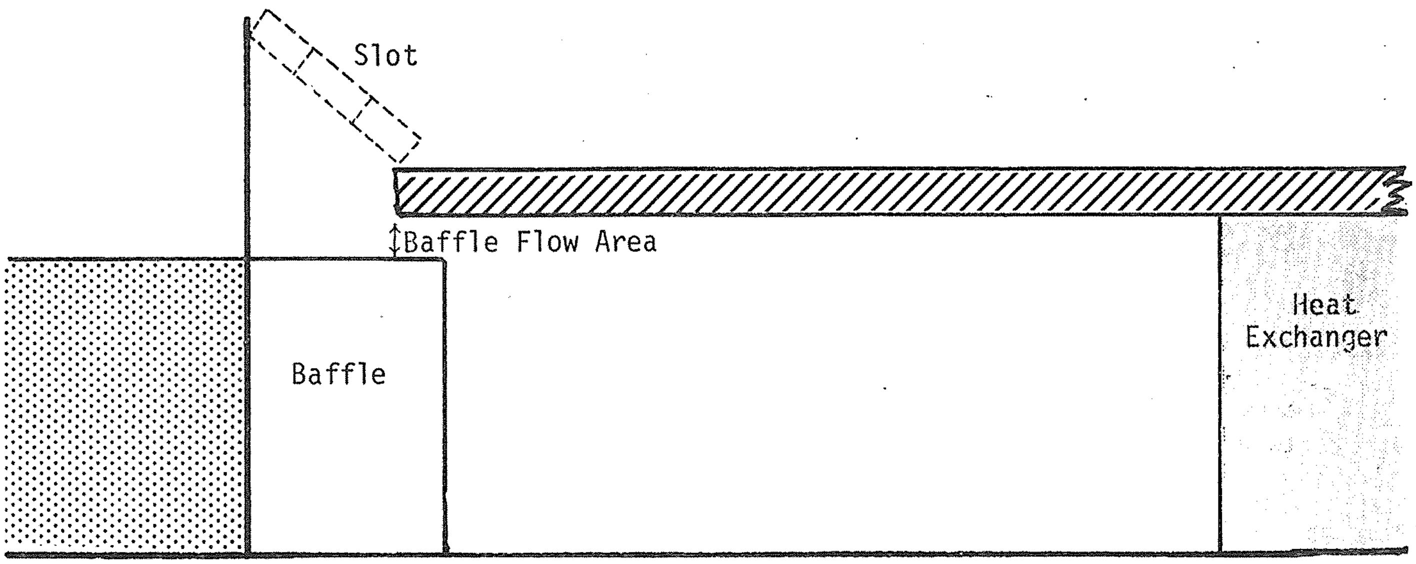


FIGURE 8. CFM/WATT VERSUS SLOT AREA



Configuration	Slot Area (Sq. Inch)	Baffle Area (Sq. Inch)	CFM/Watt
Open Intake, no baffle	48	96	3.25
(4) 7/32" x 4 1/2" slots and baffle	3.9	6	1.45
Open Intake baffle	48	6	2.00
Slot, no baffle	3.9	96	1.50
Large slot, baffle	18.0	6	2.17
Large slot, no baffle	18.0	96	3.10

FIGURE 9. SLOT AND BAFFLE COMBINATION TEST

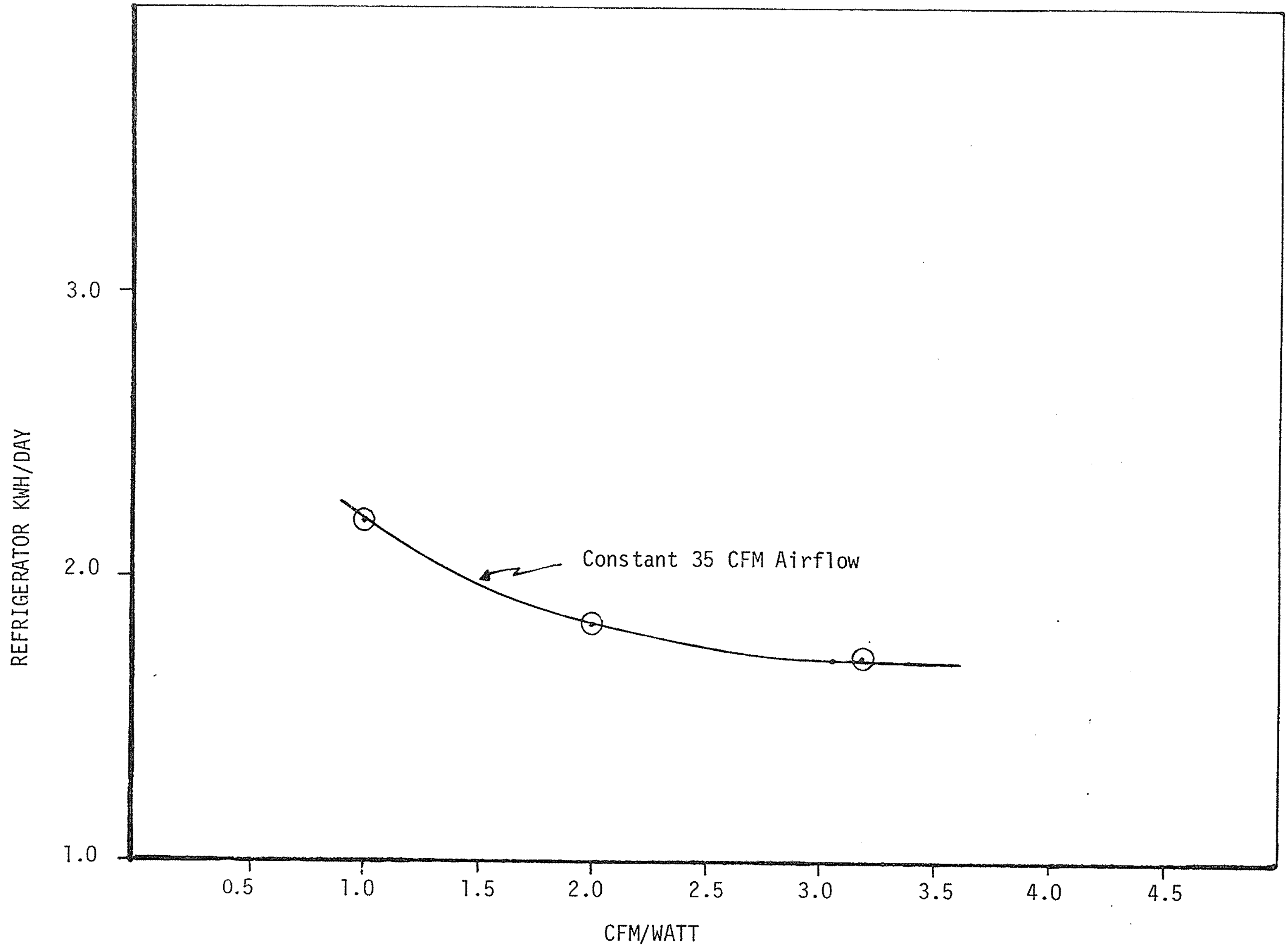


FIGURE 10. EFFECT OF FAN CFM/WATT ON CABINET ENERGY CONSUMPTION

The system curve for the 18 square inch slot area (and no baffle) is shown in Figure 11 along with the relevant fan curves. The intersection of the system curve and fan curve yields the operating points which are:

Configuration	CFM	Watts	CFM/Watt
Std. propeller, std. motor	34.5	11.3	3.05
Reverse curve centrifugal	37.0	12.0	3.08
Tube axial-integral	33.0	8.8	3.75
Std. propeller, permanent split	34.5	6.0	5.75

The last configuration was developed by adapting the standard propeller to a special permanent split capacitor motor. It was tested at one point and extrapolated to the design point.

The air system CFM/Watt values translate to energy cost savings in the following manner.

TABLE 14

Summary of Fan-Motor Improvements
(without a baffle)

Configuration	CFM/Watt	KWH/Day	Annual Energy Cost Savings ($)
Std. propeller, std. motor	3.05	1.72	Baseline
Reverse curve centrifugal, shaded pole	3.08	1.72	0
Tube-axial integral	3.75	1.70	0
Std. propeller, permanent split	5.75	1.59	1.90

For an 18 square inch intake area (no baffle) and an unfrosted heat exchanger, the benefits of the fan and motor improvements are relatively minor.

1.4.3 Findings

- The air inlet should be designed to provide a minimum of 18 sq. inches of intake area.
- The baffle or step should be removed.
- Evaporator coil inlet and outlet air flow tapers (Figure 6) are not effective.
- Improved fan/motors will not provide much savings if an 18" intake area is provided.

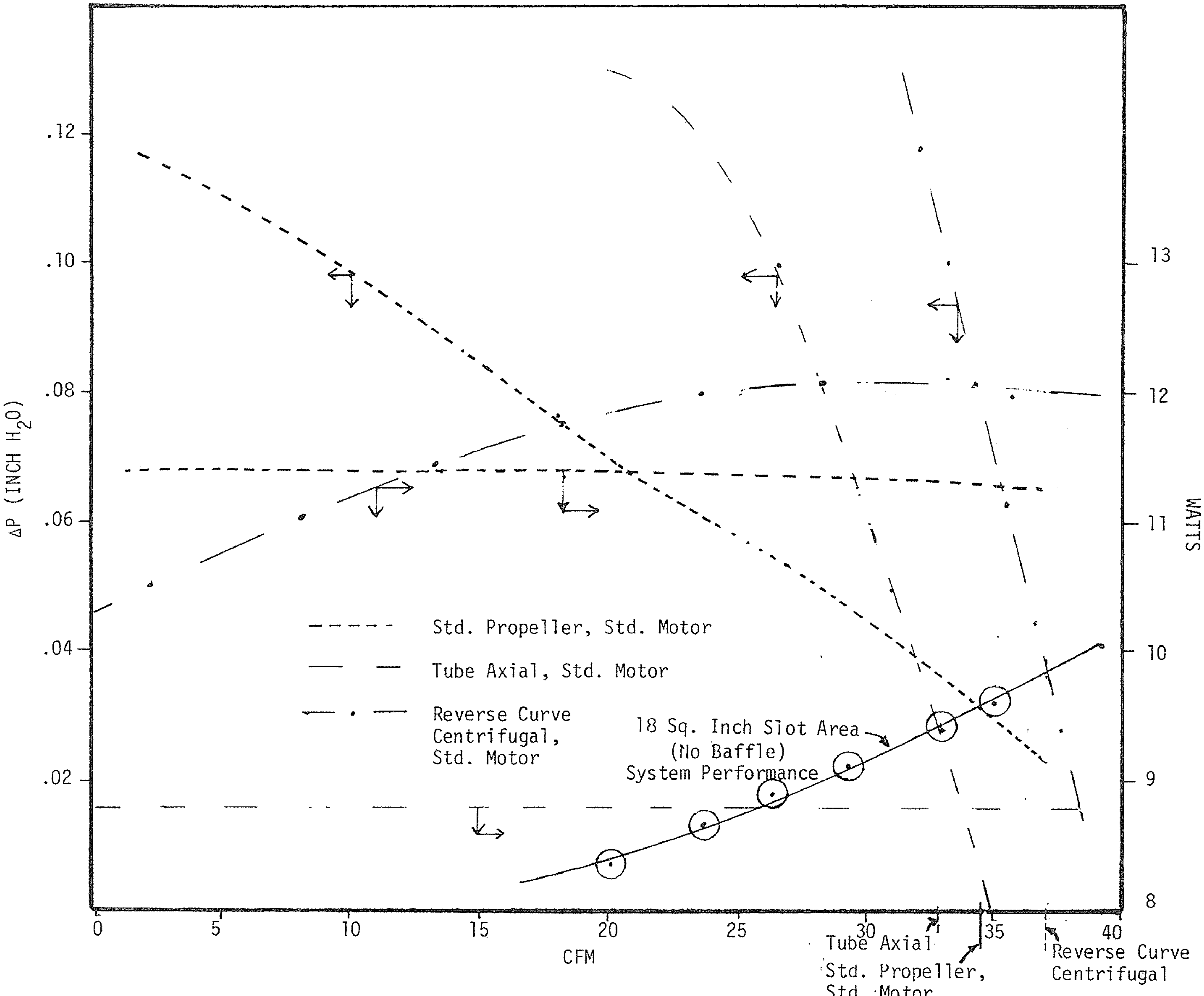


FIGURE 11. FAN/SYSTEM COMPARISONS

1.5 Bench Test-Computer Model Verification

1.5.1 Purpose

Validate the refrigeration component computer model that will be used to size prototype components.

1.5.2 Approach

An apparatus for evaluating the performance of refrigeration components and validating the computer analysis was developed, and a schematic is shown in Figure 12. The apparatus is a calorimeter-type test for the evaporator refrigeration capacity and a controlled ambient condition for the compressor and condenser units. Twenty-six thermocouples located on the key system elements are monitored with a Kaye digital multi-point recorded. The test procedure for evaluating unit performance is described below.

After the components have been assembled and the system leak checked and evacuated, the compressor is started and the refrigerant is filled through the suction side until the evaporator operateds flooded and two-to-five degrees of subcooling from the condenser are achieved. Operation of the calorimeter apparatus at a test point is relatively simple. The desired heat input is set, and after the system comes to complete equilibrium, prêssures and temperatures are monitored.

The bench tests were undertaken to validate the computer model. The computer model consists of submodels of:

- Compressor - mass flow rate and power as a function of evaporator and condenser temperatures
- Condenser - 3 part condenser; subcooling, condensing and super heat sections in a cross-flow (forced air) or natural convection heat exchange.
- Interchanger - counterflow heat exchanger
- Capillary - constant enthalpy
- Evaporator - 2 part evaporator: evaporating and superheat portions

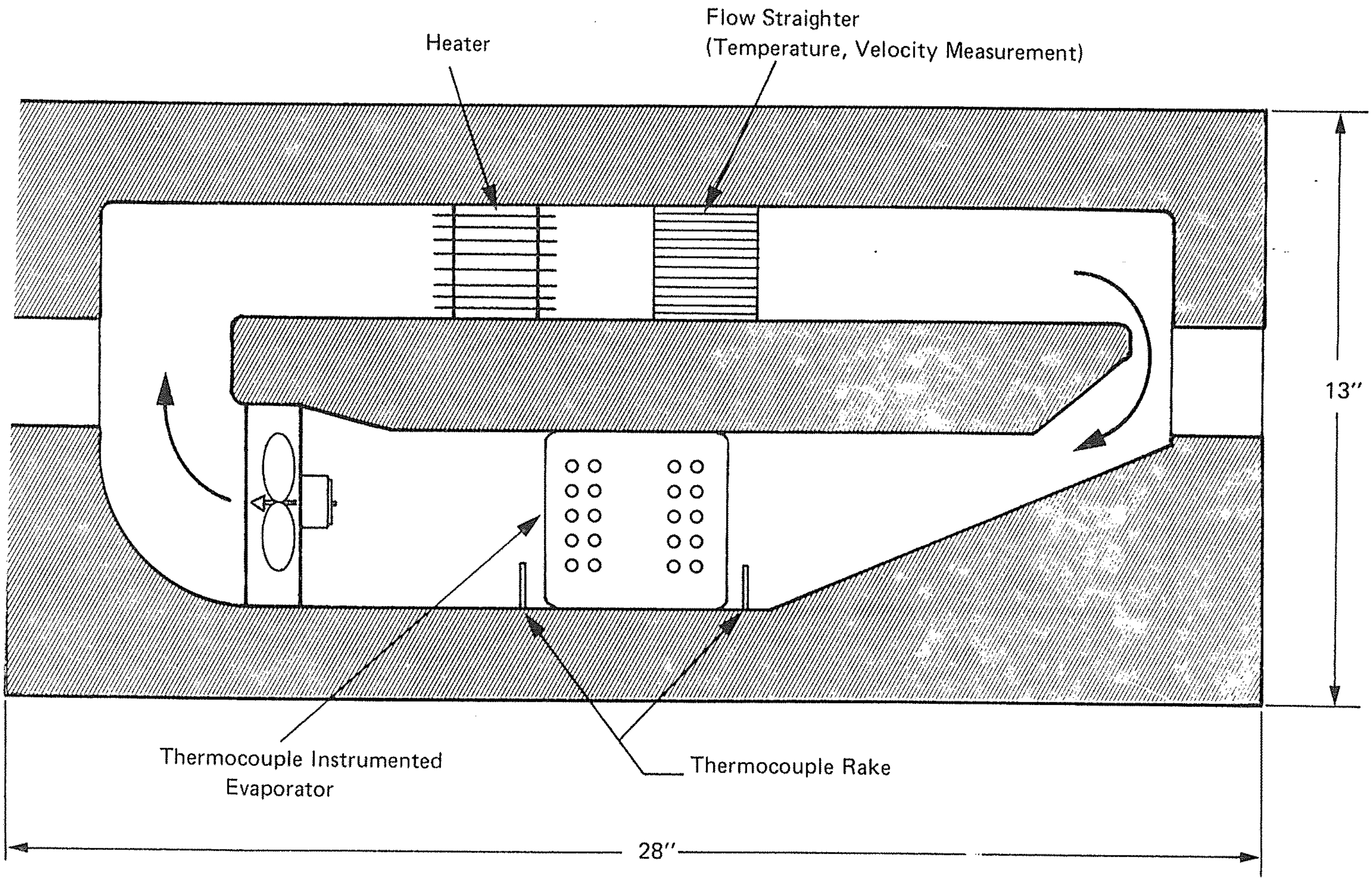


FIGURE 12 BENCH TEST – EVAPORATOR SECTION

1.5.3 Findings

The predicted results and measured experimental data are shown in Table 14.

Manufacturer's compressor performance data were found to be an unreliable source. A calorimeter test on the compressor used in the Phase I prototype performed at Amana (8-23-78) resulted in:

- Mass flow rate: 16% higher than manufacturer data at design point (-10°F, 130°F).

- Power input: 2% higher than rated value at design point (-10°F, 130°F).

This variation was accommodated in the model by varying the compressor performance by a fixed percentage increase (or decrease).

Non ideal component performance such as:

1. Frequency of unwelded condenser wire

2. Increase air temperature over the condenser due to compressor heat

3. Uneven air flow in evaporator

was suspected to cause some error but the individual effects were not evaluated.

Table 15

Validation of Computer Simulation
(Bench Test Data)
(Computer Prediction)

Test Number	Temperatures °F: Condenser, Air: Room	Inlet to Evaporator	Condenser, Refrigerant: Inlet	Mid	Outlet	Evaporator, Refrigerant: Mid	Outlet	Compressor Watts	Refrigeration Btu/hr.	Percentage Agreement
5	90	-6.6	107.5	105	104.5	-17.8	-16	173	537.6	
		-6.8	107.5	103.4	101.8	-18.3	-16.5	163.7	548.1	+2%
3	90	0.3	110	108	107	-13	-11	190	622	
		0.09	110	107	91.8	-13.4	-11.4	184	640	+2.8%
1	90	5.5	111	109	107.5	-9	-7.5	195	684	
		5.3	111	107	101.5	-9.5	-7.9	199.8	703.2	+2.8%
11	90	11.2	112.5	110	106	-5.0	-2.5	208	747.5	
		11.09	112.5	109	99	-5.3	-2.8	212	779.8	+4.0%
13	90	16.3	115.5	112	109	-1.0	+1.0	220	813.4	
		16.2	115.5	110.3	101	-1.6	+ .42	222	844.6	+ 3.8%
1	70	-5.5	88	85	83.5	-17.5	15.5	175	575.8	
		-5.6	88	85.8	83.5	-18.9	-16.9	161	633.5	+10%
2	70	+1.0	89	86	83	-13.5	-11.5	188	680	
		+0.9	89	87.8	83	-14.3	-12.3	179.8	722.6	+6%
20	70	8.0	92	89.5	71.5	-10.5	+4	190	778.4	
		7.9	92	91.9	73.7	- 9.5	+5.0	199	826.5	+6%
22	70	14.3	98	91	71.5	- 8.3	+14	194	801.9	+13%
		14.2	98	92.5	80.5	- 4.9	+17.4	212	908.8	

1.6 Thermostatic Expansion Valve Analysis and Tests

1.6.1 Purpose

Predict the energy savings possible through improved refrigeration controls and demonstrate the level achievable with a thermostatic expansion valve.

1.6.2 Approach

The capillary tube is the standard expansion device, it consists of a length soldered to the return (suction) line and a short additional length leading to the evaporator. Our analysis of the standard capillary tube indicates that most of the pressure drop takes place in the unattached length of capillary tube between the interchanger and evaporator (see Figure 13). The thermostatic expansion valve or other expansion device could be located in place of the capillary tube at this point.

Bench Test

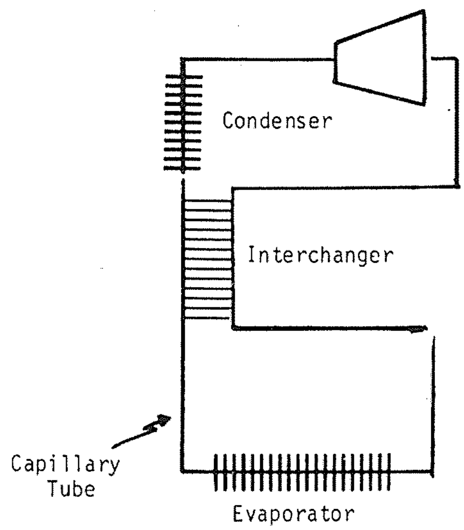
The capillary tube-interchanger performance in a 90°F and 70°F room is shown in Figures 14 and 15. The energy efficiency ratio (cooling capacity ÷ input watts) and the cooling capacity (in Btu/hr.) are given as a function of the average inlet air temperature to the evaporator. The inlet air temperature is the coupling between the cabinet refrigeration requirement and the refrigeration unit. A rise in inlet air temperature corresponds to an increase in heat load. An automatic defrost unit generally has inlet air temperatures between +10°F and -10°F during each compressor cycle. During the initial pull down and during any door openings, the inlet air temperature rises above this level.

High EER Strategy* - Simulation

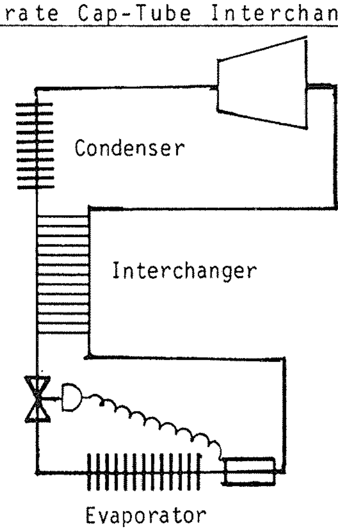
The control strategy used to predict the highest EER for each air temperature was to simulate lowered evaporator temperature until the exiting refrigerant was within several degrees of the entering air. This was the condition of maximum refrigerant superheat, and it corresponds to maximum EER. The results of this evaluation are shown in Figures 16 and 17. A 6% gain in refrigeration unit efficiency in the 90°F test was predicted. The most dramatic effect of this theoretical efficiency control strategy is seen at high loads, where high efficiency and increased refrigeration capacity are achieved. This control approach also increases the efficiency and capacity in a 70°F room.

The benefit of the High EER Control Strategy is not highly significant over the range of air temperatures experienced in the present 90°F test. The major potential benefit of this approach

*The term "high EER strategy" refers to the computer simulation of the system with the expansion device delivering maximum superheat corresponding to maximum EER.



Separate Cap-Tube Interchanger



Thermostatic Expansion Valve

Fig. 13. EXPANSION CONTROL DEVICES

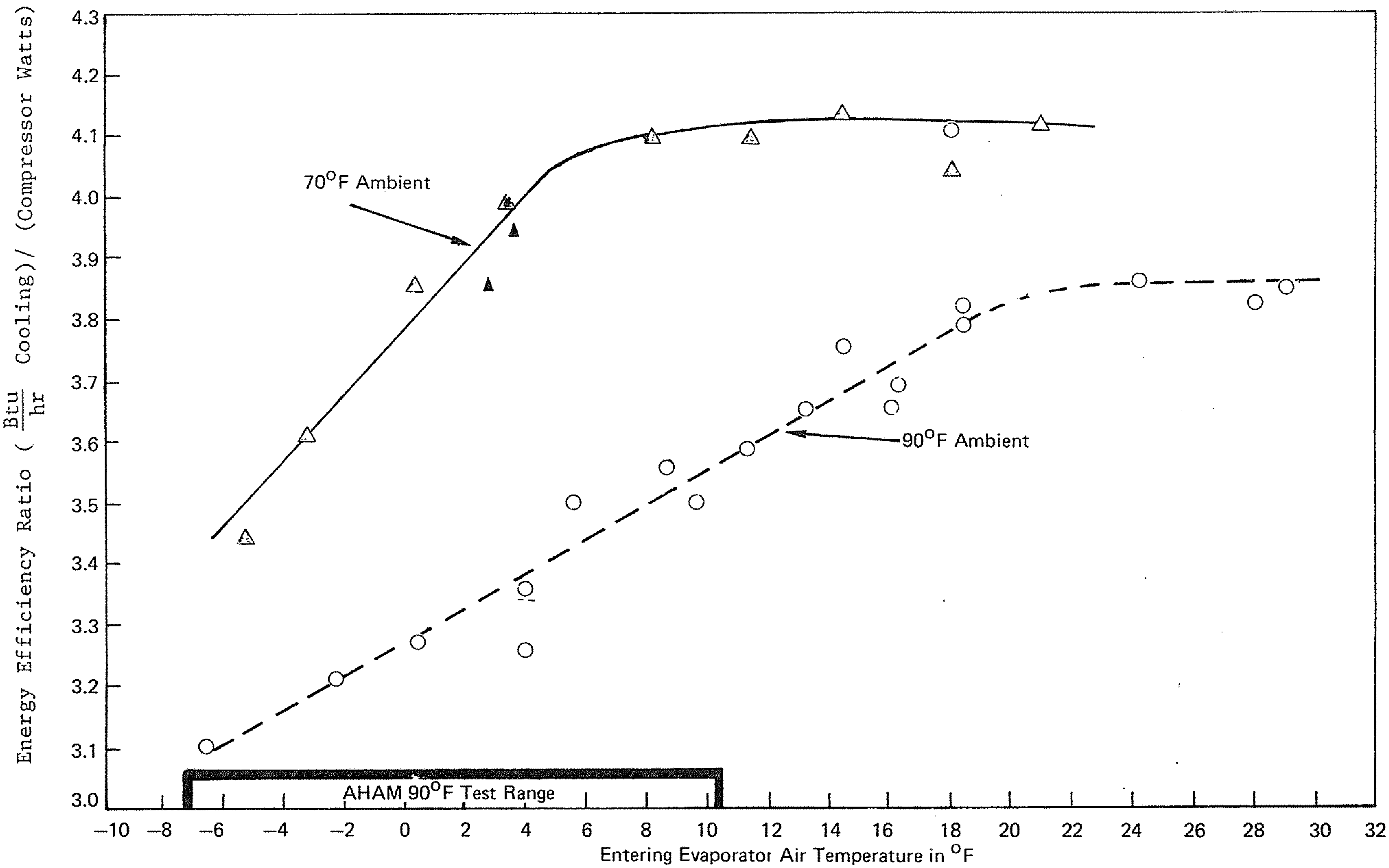


FIGURE 14 PERFORMANCE OF A STANDARD CAPILLARY UNIT

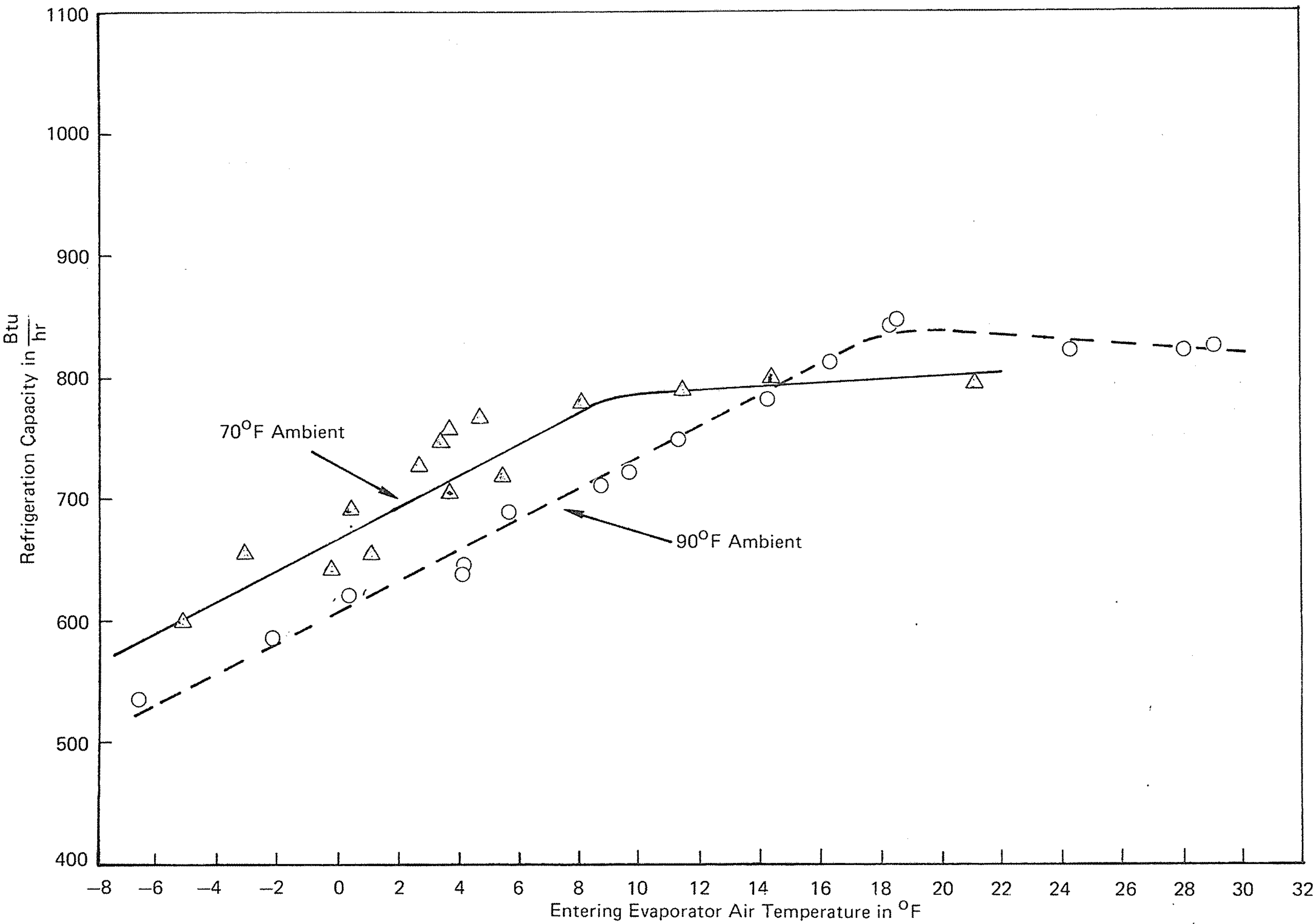


FIGURE 15 PERFORMANCE OF A STANDARD CAPILLARY UNIT

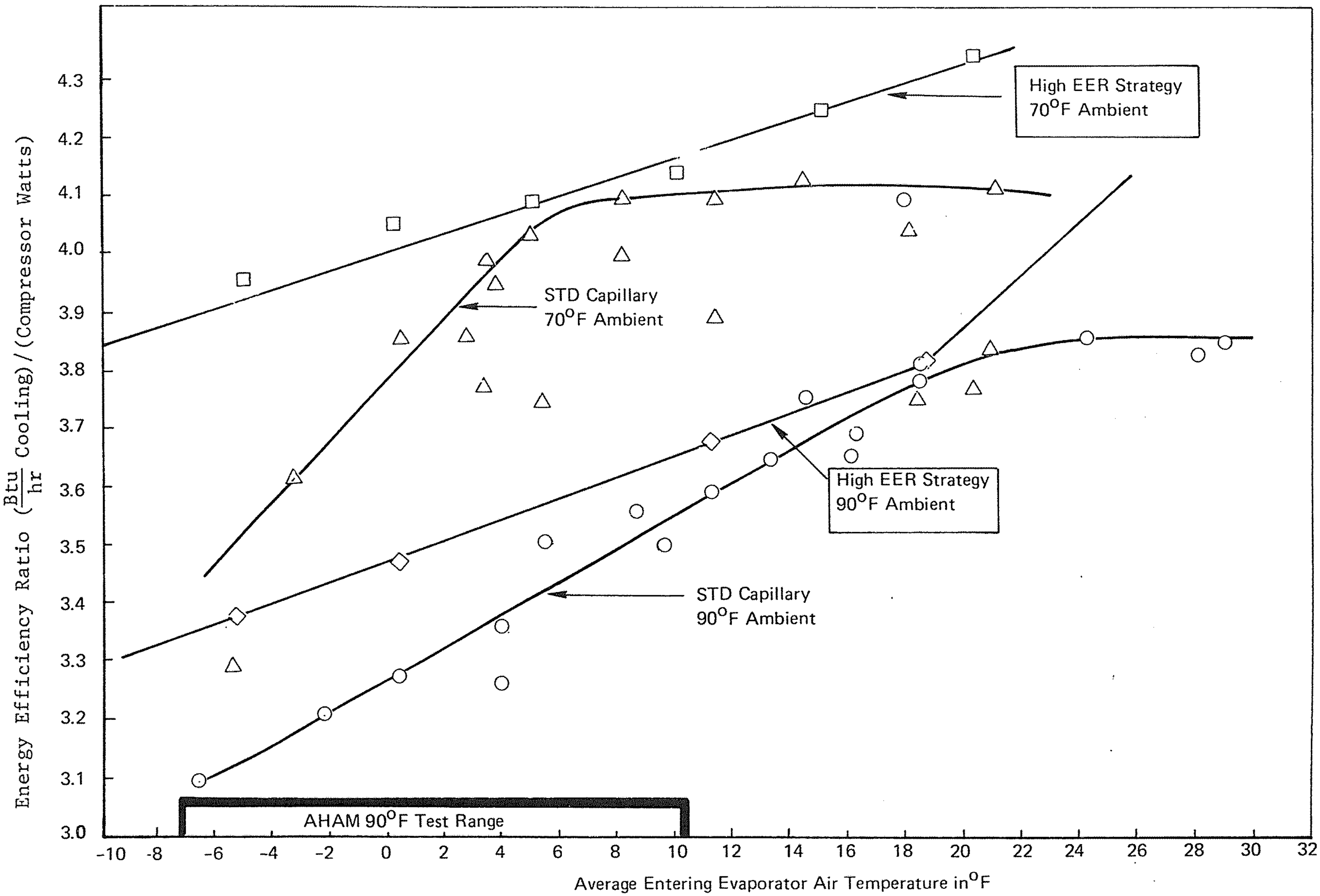


FIGURE 16 HIGH EER CONTROL STRATEGY (PREDICTION)

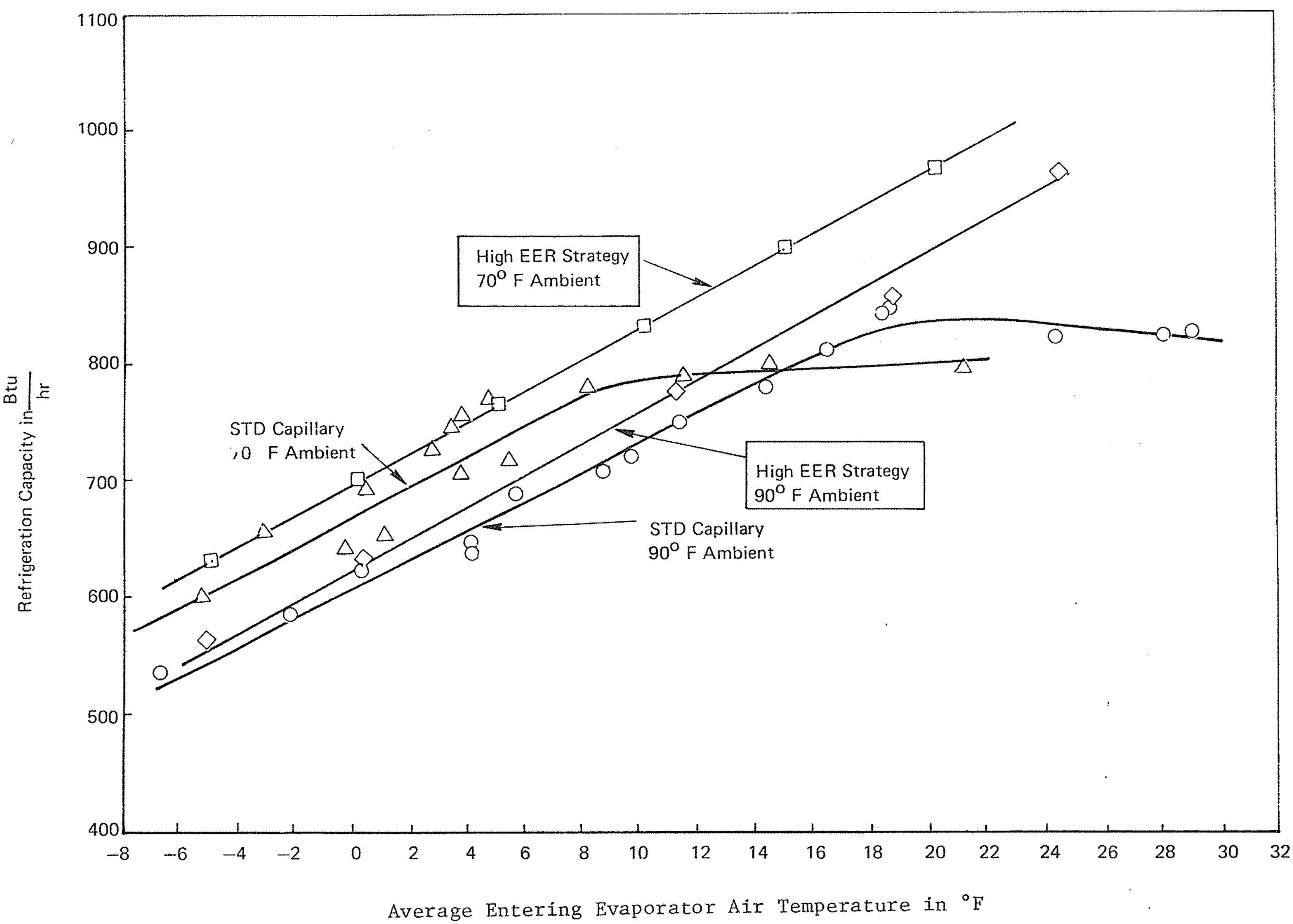


FIGURE 17 HIGH EER CONTROL STRATEGY (PREDICTION)

under today's tests is the increased capacity at higher entering air temperatures. This would increase the unit capacity under the dynamic (door opering) gulf States test where higher air temperatures are experienced. With standard capillaries, the potential unit capacity is not achieved and often times manufacturers are forced to increase heat exchanger and compressor sizes to compensate. Under extreme conditions the capacity advantage over the capillary is 16% in a 90°F room, and 20% in a 70°F room.

The predicted advantage of this control strategy is due to reducing the evaporator temperature until the superheat temperature leaving the evaporator equals the entering air temperature. This can be nearly accomplished by using a Thermostatic Control Valve.

Thermostatic Expansion Valve (TEV) Tests

A series of tests were performed with the same apparatus as used in the previous tests with the exception of a TEV* used in place of a capillary tube. A comparison of the STD capillary tube and TEV test results are shown in Figure 18. The main benefit of the TEV occurs at the higher inlet air temperatures and when a constant charge system is operated at different room temperatures. At the average entering air temperatures experienced in a closed door test (-6 to +10°F) a standard (STD) capillary performs close to the TEV at 90°F and at 70°F. At higher entering air temperatures, the TEV excels in EER by 5% in the 90°F room and nearly 12% in the 70°F room.

The TEV performance (Figure 18) follows the predicted High EER Strategy (Figure 16) closely (within 3%) throughout the range of operation in a 90°F room. The TEV performance in the 70°F room departed from the predicted values:

- 8% lower than predicted at low air temperature
- 5% higher than predicted at higher air temperature

The TEV behavior in the 70°F room test is not totally consistant with the predicted High EER Strategy and we attribute this difference to the constraint of a constant refrigeration charge (an effect the computer model is not designed to manage).

1.6.3 Findings

The TEV is unlikely to show a measurable improvement in unit energy consumption as when the average entering air temperature is about 0°F (see Table 15). The TEV is likely to show an improvement over standard capillaries for high usage operation, when the entering air temperature is closer to 20°F which would occur in a door opening and food load usage test.

*We acknowledge the assistance provided by Sporlan Valve Company in this phase of the program.

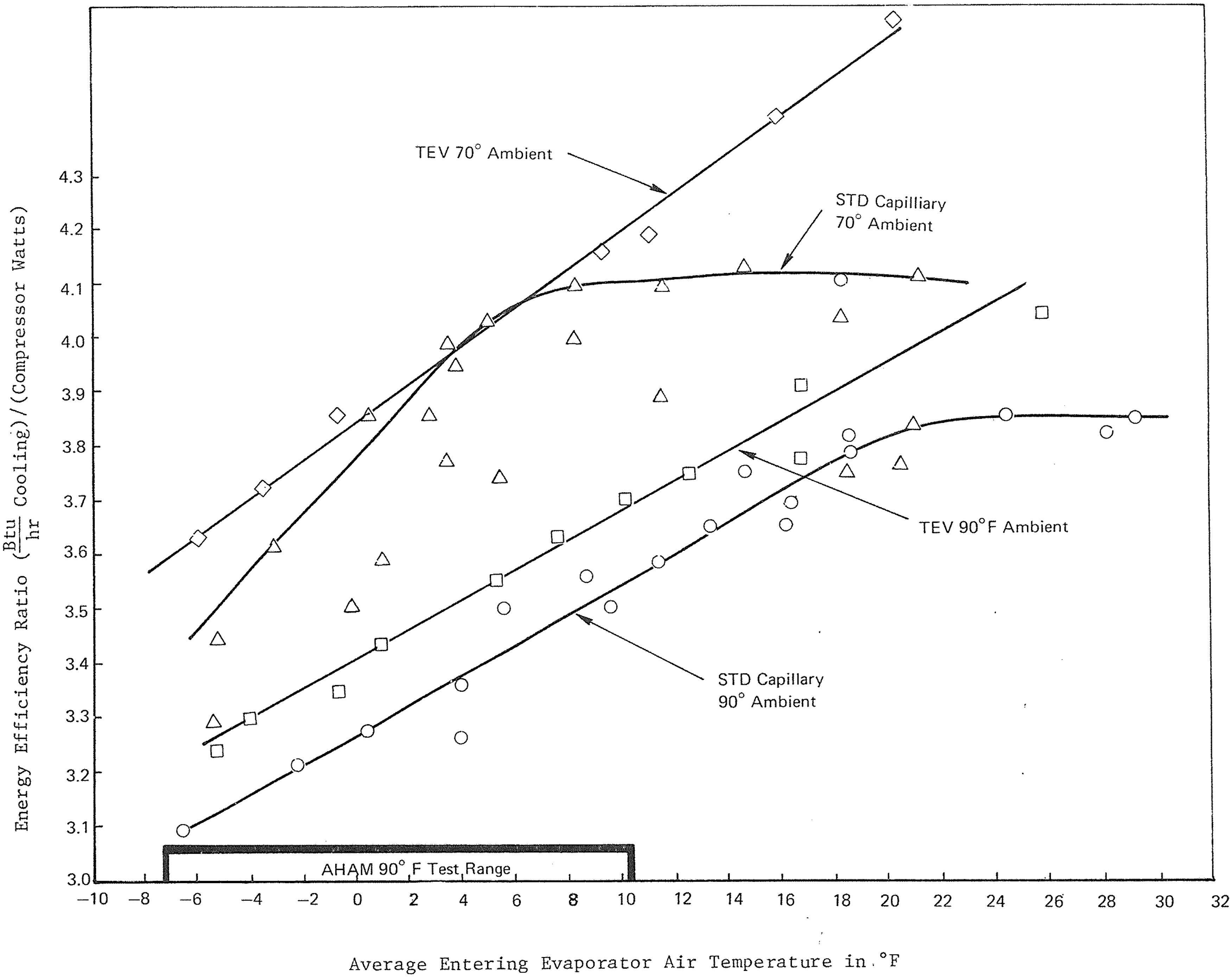


FIGURE 18 COMPARISON OF STD CAPILLIARY AND THERMOSTATIC EXPANSION VALVE TEST RESULTS

TABLE 16

ENERGY SAVING BENEFIT OF TEV

FOR VARIOUS OPERATING CONDITIONS

Room Air Temperature (°F)	Entering Air Temperature (°F)	Capacity	EER	KWH/Day	% Benefit of TEV
90	0 STD	610	3.27	1.51	
	TEV	630	3.40	1.45	4
	20 STD	830	3.85	1.24	
	TEV	890	4.03	1.17	5
70	0 STD	670	3.85	.87	
	TEV	700	3.76	.87	0
	20 STD	300	4.1	0.79	
	TEV	935	4.56	.70	11

The above calculations are based on the following relations:

$$\frac{\text{kwh}}{\text{day}} = .024 \left(\frac{176.7}{Q_{ref} - 57.51}\right) \left(\text{Compressor watts} + 16.8\right) \text{ at } 90°\text{F ambient}$$

$$\frac{\text{kwh}}{\text{day}} = .024 \left(\frac{117.2}{Q_{ref} - 57.51}\right) \left(\text{Compressor watts} + 16.8\right) \text{ at } 70°\text{F ambient}$$

where:

- 176.7 $\frac{\text{Btu}}{\text{hr}}$ is the estimated steady cabinet heat load for the prototype (See Table 11) in a 90°F room.
- 117.2 $\frac{\text{Btu}}{\text{hr}}$ is the estimated steady cabinet heat load in a 70°F room.
- The fan and heater power is 16.8 watts.

Based on the small energy savings of the TEV under standard 90°F closed door test we do not recommend incorporating the device in the prototype, unless:

1. The reduced capacity under high load tests of the prototype with capillary tube indicate that substantial benefit will accrue from increased capacity at high entering air temperature.

2. Impaired performance is observed during cyclic operation of the prototype as the TEV will inhibit unwanted refrigerant flow between high side and low side during off periods.

1.7 Design Guidance Test and Analysis

1.7.1 Purpose

Test a preprototype version of the new evaporator-condenser-fan design, analyze performance and reconcile differences between predicted results and measured performance.

1.7.2 Approach

A preprototype refrigeration unit was fabricated and installed in a baseline ESRFC3-16 production cabinet by Amana.

The unit had the following features:

1. The freezer evaporator fan motor was isolated from the compartment and in close thermal contact with the outer shell.

2. A standard forced convection freezer evaporator was used in combination with a natural convection fresh food compartment self-defrosting evaporator.

3. The hot wall condenser was replaced with a static back mounted condenser.

4. The defrost-to-defrost interval was increased beyond 12 hours of compressor run time, as moist air in the fresh food compartment was no longer admitted to the freezer evaporator.

TABLE 17

Comparison of Unit Performance

Test Date	Dec. 19, 1977 Design Guidance	July 12, 1978 Std. ESRFC3-16 Refrigeration Unit
Average evaporator temperature °F	-10.0	-14°F
Freezer temperature °F	3.5	5
Fresh food temperature °F	36.0	38
Percent run time	45.8	54.3
Energy consumption in Kwh per day	2.09	3.16

Energy savings from evaporator-condenser design = 33.8%

The installed refrigeration capacity can be derived from the measured run time and cabinet heat flow as follows:

- knowing -

  227 Btu/hr - calorimeter value for cabinet with fan on

  0.458 - measured fraction run time

  123.3 Btu/hr - known internal heat: 102.3 fan and heaters
  21.0 gasket

- then -

$$Q_{installed} = \frac{227}{.458} + 123.3 = 618.3 \ \frac{Btu}{hr}$$

A wind tunnel calorimeter test of the actual evaporator and fan (50 cfm) was performed and the overall heat transfer coefficient measured (UA = 70 Btu/hr. °F) for an air mass flow rate x heat capacity (CA) of 57 $\frac{Btu}{hr°F}$ (corresponding to 50 CFM).

Computer simulation of the refrigeration unit performance with the wind tunnel test values predicted a -8.8°F evaporator and 638 Btu/hr capacity, suggesting that the full evaporator potential is not realized when installed. It was hypothesized that due to an air path blockage a fraction of the air flow and heat exchanger surface are not used. A parametric computer analysis was performed using reduced air flow rates and evaporator sizes (UA), a summary is given in Table 17. The installed evaporator-air flow performance corresponds closely to 80% of the (predicted)wind tunnel ideal flow value.

TABLE 18

Effect of Air Delivery to Evaporator

% of Wind Tunnel Value	Freezer Evaporator Size UA	CA (Both in Btu/hr °F)	T_{evap} °F	Q_{total} Btu/hr Expected Value 618.3 Btu/hr
100	70	57	-8.8	639
90	63	51.3	-9.3	628
80	56	45.6	-10.6	611

1.7.3 Findings

a. The refrigeration unit design forced/free convection evaporator combination and static back mounted condenser concept offer 33% energy savings and certainly should be incorporated in the Phase I prototype.

b. The installed unit performance correlates with the performance of 80% utilization of the full evaporator potential suggesting a redesign of the air flow passage to achieve improved flow (performed in Section 1.4).

1.8 Cabinet Calorimeter Test

1.8.1 Purpose

Determine cabinet heat flow so that flange heat leak can be estimated.

1.8.2 Approach

A constant heat source is applied to each cabinet compartment to maintain 110°F $\pm$ 0.2°F in the fresh food compartment and 50°F $\pm$ 0.2°F in the freezer. A cabinet overall conductance is calculated and the cabinet heat leak under 90°F room 5 and 38°F cabinet temperatures is then estimated using an adjustment factor to account for the change in insulation conductivity with average temperature.

1.8.3 Findings

The temperature corrected cabinet flow calorimeter values for the baseline unit yield flange heat flows within 8% of the values calculated from independent flange heat flow tests as reported in Section 1.1.2.

The temperature corrected flange heat transfer rate for the prototype cabinet of 40.18 $\pm$ 3.6 Btu/hr (Table 18) compares with the predicted value of 41.9 Btu/hr (for the double gasket flange) given in Table 9.

Table 19

Summary -- Calorimeter Test Results

Model	Room Temperature	Heat Flow in Btu/hr: Calorimeter Raw Data	Calorimeter Adjusted	Calculated Values: Walls and Wedge	Calculated Values: Flange
ESRF--16-C (Test #1)	68.5°F	306.8	268.3	202.82	65.58
ESRF--16-C (Test #2)	92.0°F	319.6	262.2	202.82	59.38
ADL-16 (Test #1)	91.0°F	207.5	171.4	134.82	36.58
ADL-16 (Test #2)	55.0°F	194.0	178.6	134.82	43.78

2. PHASE I PROTOTYPE

With the completion of the individual design option tasks and analysis, a clear path was indicated for the development of an energy saving refrigerator/freezer prototype containing the following features:

- optimized insulation design
- static backmounted condenser (instead of hot wall type)
- standard forced convection freezer evaporator
- free convection fresh food evaporator in series with the freezer evaporator - for off cycle self-defrost
- new fan/air flow path design
- reduced defrost frequency
- double gaskets

Amana fabricated the prototype cabinet, using a new plastic liner design for the unit and tested the unit. An energy consumption level of 1.78 kwh/day was measured--well below the target level of 2.0 set at the outset of Task 3. The prototype also performed well under the heavy load usage tests.

2.1 Phase I Prototype Design

2.1.1 Purpose

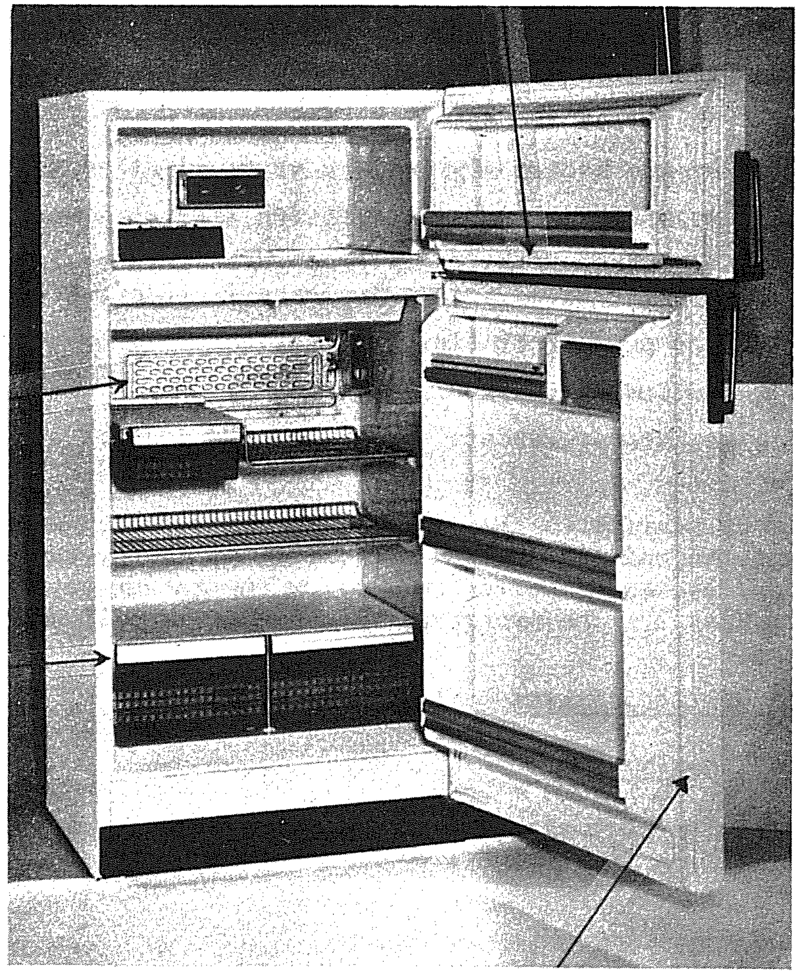
Size the refrigeration unit to the optimized insulation design developed in Section 1.3.

2.1.2 Approach

The cabinet specifications and a picture of the unit are shown in Figure 19.

The standard 5° Freezer, 38°F Fresh Food compartment temperatures and a 90° room temperature were employed in the refrigeration unit simulation to evaluate the desired unit size. A series of parametric studies on each of the components was undertaken and a promising design identified. The predicted performance under the Gulf States Test conditions was then made.

Predicting performance in the Gulf States Test is difficult. It is a high humidity test with scheduled door openings. The highly transient thermal characteristics of this test and the uncertain latent heat loads make steady state predictions awkward. The approach taken for evaluating the likely unit performance under the Gulf States Test was to estimate a door opening load and correlate this load with the measured ballast temperature.



INSULATION THICKNESS (INCHES)							HEAT FLOW (BTU/HR)					
Freezer				Fresh Food								
Side	Front	Back	Top	Side	Front	Back	Freezer	Mullion	Fresh Food	Gasket	Wedge	Total
3.00	2.50	3.00	3.00	2.25	2.50	2.25	43.08	13.28	68.39	41.9	10.0	176.7

FIGURE 19 **PROTOTYPE CABINET SPECIFICATIONS**

Compared to the 70°F closed door test, the Gulf States Test represents a 20% increase in the freezer load and a 250% increase in the fresh food load for the prototype cabinet, making the fresh food compartment the stressed section. Therefore, acceptable performance of the unit is the measured average compartment (water ballast) temperature in the fresh food compartment.

Gulf States Test

The Gulf States Test consists of the following set of conditions:

TABLE 20

Conditions of Gulf States Test

Duration	Door Openings	Conditions: Room Temp.	Conditions: Room Relative Humidity
16 hrs.	120 refrig. 30 frz.	90°F	65%
8 hrs.	none	70°F	85%

Under equilibrium conditions the air off of the evaporator will be at saturated cabinet air temperature, which is the unknown variable (T_{FF}). The heat removed from room air to reach T_{FF} is given below.

TABLE 21

Air Enthalpy Change (BTU/lbm)
(Initial State: 90°F and 65% RH)

Final State T_{FF} in °F Temperature			
8°F	45°F	55°F	65°F
29.7	26.0	20.5	13.6

These data can be curve fitted to yield:

$$\text{air enthalpy change} = \Delta h_{air} = 52.35 - .599\ T_{FF} \text{ in btu/lbm}$$

where T_{FF} is the average fresh food compartment (or ballast) temperature in °F. Therefore the compartment heat load due to the door openings is:

$$\text{fresh food door opening heat load} = \frac{120(12)}{14.3} \times \Delta h_{air} \times \frac{1}{16\text{hr}} = 329.5 - 3.751\ T_{FF}$$

Add the wall heat flow of:

$$\text{wall heat flow} = 136 \left(\frac{90-T_{FF}}{52}\right) = 235.4 - 2.61\ T_{FF}$$

and the total heat flow is:

$$\text{total heat flow} = 564.9 - 6.361\ T_{FF}\ \text{btu/hr}$$

The Design Guidance Model was tested in a Gulf States test and the results of that test were used to check this approach.

The refrigeration capacity with a -10.°F evaporator as maintained in the Design Guidance Test (Section 1.7) is:

$$\dot{Q}_{refrigeration} = 4.58\ (T_{FF} + 10)\ \text{Btu/hr}$$

Equating the heat load with the $Q_{refrigeration}$ capacity

$$564.88 - 6.366\ T_{FF} = 4.58\ (T_{FF} + 10)$$

and solving for T_{FF} , the predicted ballast temperature is:

$$T_{FF} = 47.4°F$$

the measured value of the ballast temperature was:

$$T_{measured} = 49°F$$

This method gives a predicted mean temperature that is within the expected water ballast temperature.

Predicted Prototype Performance in the Gulf States Test

Repeating the above technique for the prototype design yields:

$$T_{FF} = 45.11°F$$

The predicted water ballast (average cabinet temperature) is below 50°F which Amana considers acceptable for the Gulf States tests.

Estimated Energy Consumption

The predicted 90° AHAM energy consumption is:

$$\text{kwh/day} = .024\left(\frac{\underset{\text{Walls}}{43 + 68.39} + \underset{\text{Mullion}}{13.28} + \underset{\text{Flange}}{15.64} + 13.16 + \underset{\text{Wedge}}{10} + \underset{\text{Evaporator Heater}}{20.46}}{\underset{\text{Refrigeration Capacity}}{688.4} - \underset{\text{Fan}}{51.15} - \underset{\text{Mullion Heat}}{(40.9)\ 1/2} + \underset{\text{Evaporator Heater}}{20.46} - \underset{\text{Flange}}{13.05}}\right)$$

$$\text{X}\left(\underset{\text{Compressor Watts}}{187.4} + \underset{\text{Fan Watts}}{15} + \underset{\text{Defrost}}{\frac{500.25}{47}} - \underset{\text{Heater}}{6}\right) + \underset{\text{Defrost Timer Watts}}{6\ (.024)}$$

$$= 1.593\ \frac{\text{kwh}}{\text{day}}$$

which is below the target level of energy consumption.

2.1.3 Findings

The Phase I prototype is predicted to meet the energy consumption target of 2.0 KWH/day and meet the Gulf States performance goal and should be fabricated and designed.

2.2 Prototype Testing

2.2.1 Purpose

Evaluate the prototype design under heavy load conditions and under standard AHAM energy rating conditions.

2.2.2 Approach

The Phase I prototype was fabricated at Amana Refrigeration, Inc., according to design specifications for the refrigeration unit developed at Arthur D. Little, Inc. A new cabinet liner was developed for this purpose along with a new outer cabinet design. The liner was vacuum molded on a model shop apparatus.

The prototype underwent the following performance tests:

1. 107°F ambient performance tests.
2. A high energy usage Gulf States test.
3. A low ambient 65° test.
4. An AHAM energy test.

These tests were performed at Amana Refrigeration, Inc., following standard test procedures developed for their product line. The high ambient (107°) tests are designed to evaluate the pull-down (cabinet cool down from ambient) characteristics, the steady-state operating conditions under 107° ambient tests and the performance with the freezer door open in the 107° room.

The prototype unit was placed in a 107° room with an enclosure simulating the built-in conditions to be experienced in kitchen application. Starting with the cabinet at ambient conditions the unit was operated until steady state performance was achieved. Its characteristics during this pull-down period were monitored to assure continual proper compressor, fan/motor, and defrost performance. Of particular concern during this test was the evaporator fan motor stack temperature and bearing temperature. The evaporator fan motor was located within the insulation and was not cooled by the evaporator discharge air and abnormally high motor temperatures are expected as the only heat transfer is through the wall to the back of the unit.

The thermostatic control is shorted and the unit is permitted to operate in the 107° room until steady temperatures are achieved.

With the thermostat still shorted, the unit is operated with the freezer door open until the compressor overload is actuated and operation is discontinued. The unit is then permitted to restart on its own to assure that the compressor can start under adverse pressure ratio conditions due to the high ambient period.

The Gulf States Test consists of a series of door opening conditions at 90° and 65% relative humidity as discussed in Section 2.1. This test evaluates the thermal and frosting characteristics of the unit under heavy usage. The unit is expected to maintain a freezer temperature below 10°F and a fresh food temperature below 50°F. Two design factors are critically examined by this test and they are:

- the defrost interval and defrosting control necessary to maintain appropriate operation under the high ambient, high humidity door opening tests, and
- the proper refrigeration capacities to maintain a desired balanced unit operation.

A 65°F ambient test was performed primarily to evaluate the control ability of the unit. Often times difficulty in maintaining cabinet temperatures occurs at low ambient and low percent run times.

A 90° AHAM door closed test was performed. Because of the increased time between defrosts a 4 point AHAM test was not performed. Rather a 1 point AHAM test was performed with the thermostat controls adjusted so that a 5°F and 38°F cabinet condition was established in the freezer and fresh food compartments respectively.

2.2.3 Findings

The results of the 107° tests indicated acceptable unit performance for the pull-down, ultimate trip and balance-out conditions (see Table 21). During the test the fan motor bearing reached a temperature of 165.5°F which is considered acceptable, though improved cooling should be considered in any future work.

The unit performance in the Gulf States Test was good by Amana standards. As can be seen in Table 22, the freezer compartment never exceed 5°F, and the refrigerator compartment was over 50° for a very limited period of time. The average percent run time during the Gulf States Test was approximately 55% which is 40% below the 90°F CD average run time of the standard ESRF 16 high efficiency production refrigerator. This indicates that reduced energy consumption is anticipated for the actual field test operation of the unit with door opening and food admission.

Table 22

ADL-16 High-Efficiency Refrigerator/Freezer Electrical/AHAM Performance

107° Room Test

Peak Conditions

Power Input	321 Watts
Winding Temperature	255° F
Condenser Temperature	149.5° F
Shell Temperature	205.5° F

Stabilized Conditions

Power Input	200 Watts
Winding Temperature	241.7° F
Condenser Temperature	136.0° F
Number of Overloads	0
Shell Temperature	191.0° F

Ultimate Trip Conditions

Power Input	365 Watts
Winding Temperature	248.6° F
Condenser Temperature	159.0° F
Shell Temperature	197.5° F

Energy Consumption

AHAM Energy Test

One Point	1.72 $\frac{\text{kwh}}{\text{day}}$ @ 33.7% R.T.

5°F Freezer
38°F Fresh Food

TABLE 23

ADL-16 HIGH - EFFICIENCY REFRIGERATOR/FREEZER

Dynamic Usage Test Results

TEST DAY	1	2	3	4	1	2	3	4	5	6
Refrigerator: Start Temperature	38.7	37.5	37.5	39.0	40.7	40.7	40.8	41.0	40.4	39.8
Maximum Temperature	50.3	48.3	47.5	48.2	50.2	50.3	49.7	49.8	49.7	48.2
Recovery Temperature *	37.5	37.5	39.0	38.8	40.7	40.8	41.0	40.4	39.8	38.8
24 Hour Average	43.8	43.5	43.7	44.0	46.1	46.4	46.8	46.5	45.6	44.5
Hours Over 50°	1	0	0	0	3	3	0	0	0	0
Freezer: Start Temperature	0	-6.0	-6.0	-6.0	+4.0	+3.5	+3.0	+2.5	+3.0	+2.0
Maximum Temperature	+2.5	0	-6.0	+4.0	+5.0	+3.5	+3.5	+5.0	+3.0	+2.0
Recovery Temperature *	-6.0	-6.0	-6.0	+2.0	+3.5	+3.0	+2.5	+3.0	+2.0	+2.0
24 Hour Average	-6.9	-6.2	-7.3	-1.6	+0.6	+1.2	+0.8	+1.5	+0.8	-0.4
Hours Over 10°	0	0	0	0	0	0	0	0	0	0
Hours Over 5°	0	0	0	0	0	0	0	0	0	0
Percent Run Time	63.6	57.8	56.5	49.7	53.1	51.1	52.1	49.7	52.2	60.5
Unit Cycles	18	21.0	18	28	26	28	27	31	23	14
Evaporator Motor Temperature Peak	158.5	149.5	145	154.5	152.5	130	128.5	128.5	130.0	151.5
Defrost cycles	0	1	0	1	0	0	0	0	0	0
Defrost Frequency	—	37 Hr	—	42 Hr.	—	—	—	—	—	—
Number of Door Openings										
Refrigerator	120	120	120	120	120	120	120	120	120	120
Freezer	30	30	30	30	30	30	30	30	30	30

*Temperature after door opening portion

The unit maintained the desired 5 and 38 cabinet temperatures in the 63°F ambient test, a necessary condition for reliable field performance. The unit performance in the AHAM tests resulted in a test energy consumption level of 1.72 kilowatt hours per day at a 33.7% on time. This compares with the predicted results given in Section 2.1.2 of 1.6 kwh per day and a 30% run time.

The prototype design has achieved the target efficiency levels set during Task 2 and as such we consider the design and development phase complete.

Arthur D. Little manufacturing staff analyzed the major design changes between the baseline (ESRFC3-16) unit and the prototype, from a manufacturing viewpoint. Unit costs are extremely difficult to estimate as costs depend on the existing (capital) equipment. Certain manufacturers will fabricate all of the major components requiring only added materials, while other manufacturers may have to purchase some of the major parts. We have elected to reflect costs with purchase components as this may better reflect the initial product introduction cost (low volume). We have assumed that the manufacturer has foam in place machines, and plastic liner equipment and that only tooling for the new cabinet is required. The following manufacturing per unit costs have been estimated:

	$/Unit
Materials	1.71
Labor	.56
Purchased Parts	17.02
Tooling (3 year amortization)	5.00
Ex factory cost	$24.29
Estimated Retail Price (2.5 mark-up)	$60.73

APPENDIX
HEAT FLOW THROUGH THE DOOR AREA WEDGE

The wedge conduction is approximated by the conduction through a sector of a circle as shown in Figure A.1. The surfaces at the top and bottom of the wedge are treated as adiabatic surfaces. The lower surface is adiabatic because the conduction across and through the flange metal is evaluated separately and is added to the wedge conduction. The upper surface is considered adiabatic. The validity of this assumption is uncertain though the error induced is judged not to be significant.

Let

$$\zeta = \frac{\pi}{\alpha}\theta \qquad \eta = \frac{\pi}{\alpha}\ln\frac{a}{r} \tag{1}$$

The cylindrical heat transfer equation is transformed to the simple equation:

$$\frac{d^2v}{d\zeta^2} = \frac{\alpha^2 v}{\alpha\zeta^2} \tag{2}$$

where

v = temperature.

The boundary conditions for the adiabatic surfaces are:

$$\frac{dv}{\alpha r} = 0 \quad \text{at } r = a,\ b \tag{3}$$

and for the side of the wedge:

$$v = v_0 \quad \text{at } \theta = 0 \tag{4}$$

$$v = v_1 \quad \text{at } \theta = \alpha \tag{5}$$

Transforming these to the new coordinates, we get:

$$\frac{\alpha v}{\alpha\eta} = 0 \text{ at } \eta = 0,\ \frac{\pi}{\alpha}\ \ln\frac{a}{b} \tag{6}$$

and

$$v = v_0 \quad \text{at } \zeta = 0 \tag{7}$$

$$v = v_1 \quad \text{at } \zeta = \pi \tag{8}$$

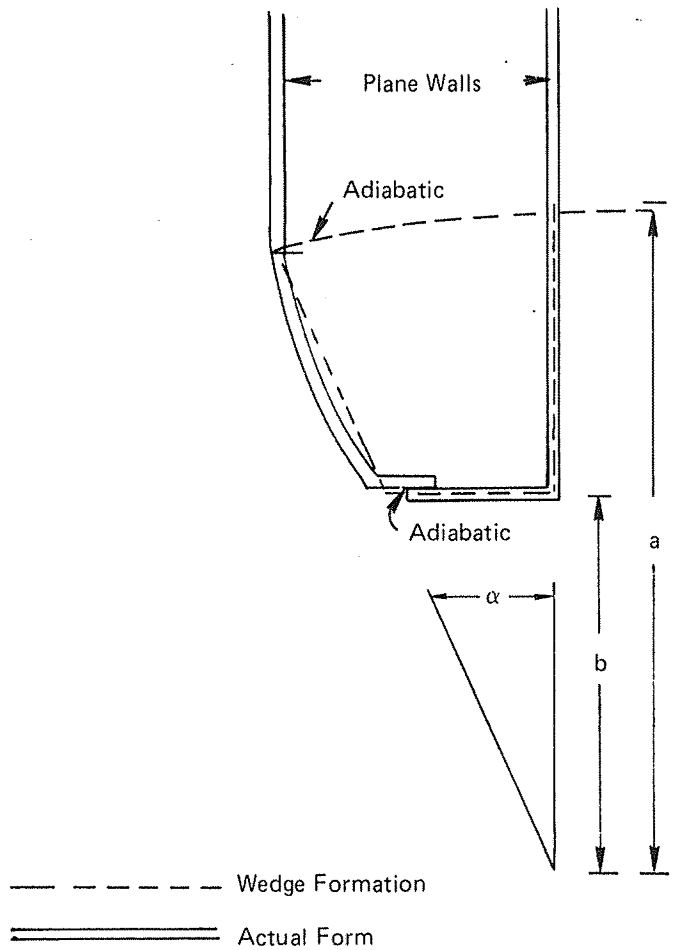


FIGURE A.1 WEDGE CONDUCTION

The transformed heat flow Equation (2) is in the form of the conduction equation for rectilinear coordinates and can be pictured in ζ, η space as shown in Figure A.

The heat transfer for the door wedge can then be simply written as:

$$\frac{q_{wedge}}{\text{Unit length}} = \frac{k \, \ell n \, a/b}{\alpha} (v_1 - v_0)$$

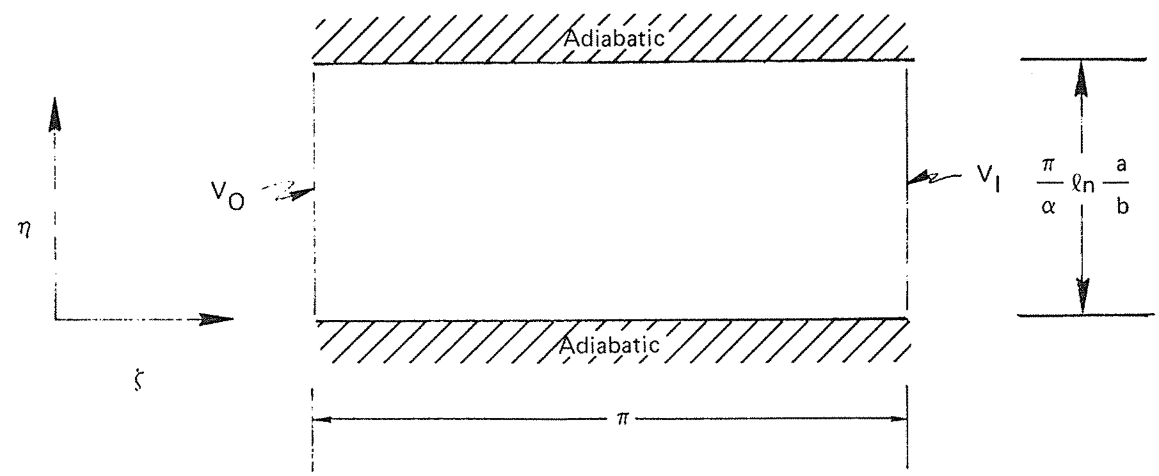


FIGURE A.2 DOOR WEDGE HEAT FLOW IN TRANSFORMED COORDINATED (η, ξ SPACE)

DEVELOPMENT OF A HIGH EFFICIENCY, AUTOMATIC DEFROSTING REFRIGERATOR/FREEZER

TASK 4 REPORT

DEMONSTRATION PLAN

Arthur D Little, Inc.

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PROPOSED DEMONSTRATION PROGRAM
FOR A
HIGH EFFICIENCY REFRIGERATOR/FREEZER

1. INTRODUCTION

Federal and State energy efficiency programs have targeted industry-wide efficiency improvements for the refrigerators to be offered in the 1980 marketplace. These targets were based on government-sponsored studies and industry-submitted statements at public hearings. For the typical 18-cubic-foot, automatic defrost combination refrigerator-freezer, the tentative Federal level is 3.1 kilowatt-hour per day consumption.* During Phase I of this program, Arthur D. Little and Amana Refrigeration, Inc. evaluated a number of design modifications to reduce energy consumption below this level. A number of promising concepts were selected, and a prototype unit was designed and built. Laboratory tests following the Federal test procedures have shown a 36% reduction in energy usage when compared to the Federal level (based on the maximum technologically feasible energy efficiency level, Federal Register 42, No. 178). This represents a 407 kilowatt-hour savings per year per unit, and meets the target set for the prototype at the outset of the study.

This project was initiated for the purpose of accelerating the introduction of a high efficiency refrigerator-freezer into the marketplace. The Phase I project results stimulated Amana to seriously consider the early commercialization of a high efficiency unit and though Phase I successfully achieved high efficiencies in the laboratory, additional work was identified to help move the design further along the commercialization path. Field testing of the unit under actual usage and market testing would provide this essential information for product development. Figure 1 highlights the normal progression to full scale production and marketing for an all-new refrigerator-freezer design. We concluded that with DOE support the Amana-ADL team could leap several steps to a field-market test saving about 3 years of normal development time.

This plan outlines a project to evaluate the energy savings, marketability, reliability, and consumer acceptance of the high efficiency unit for the purpose of accelerating commercialization. Specifically we recommend a publicly-supported demonstration project designed to assess:

1) The comparative energy savings of the high efficiency refrigerator-freezer against a baseline production model in actual kitchen use.

2) The maintenance and service requirements of the high-efficiency refrigerator-freezer over a reasonable time

*This represents the Maximum Technologically Feasible Energy Efficiency Level for an 18.0 cubic ft. unit with a 4.3 cubic ft. freezer.

under field conditions.

3) A sampling of consumer response to the new product (usability and marketability).

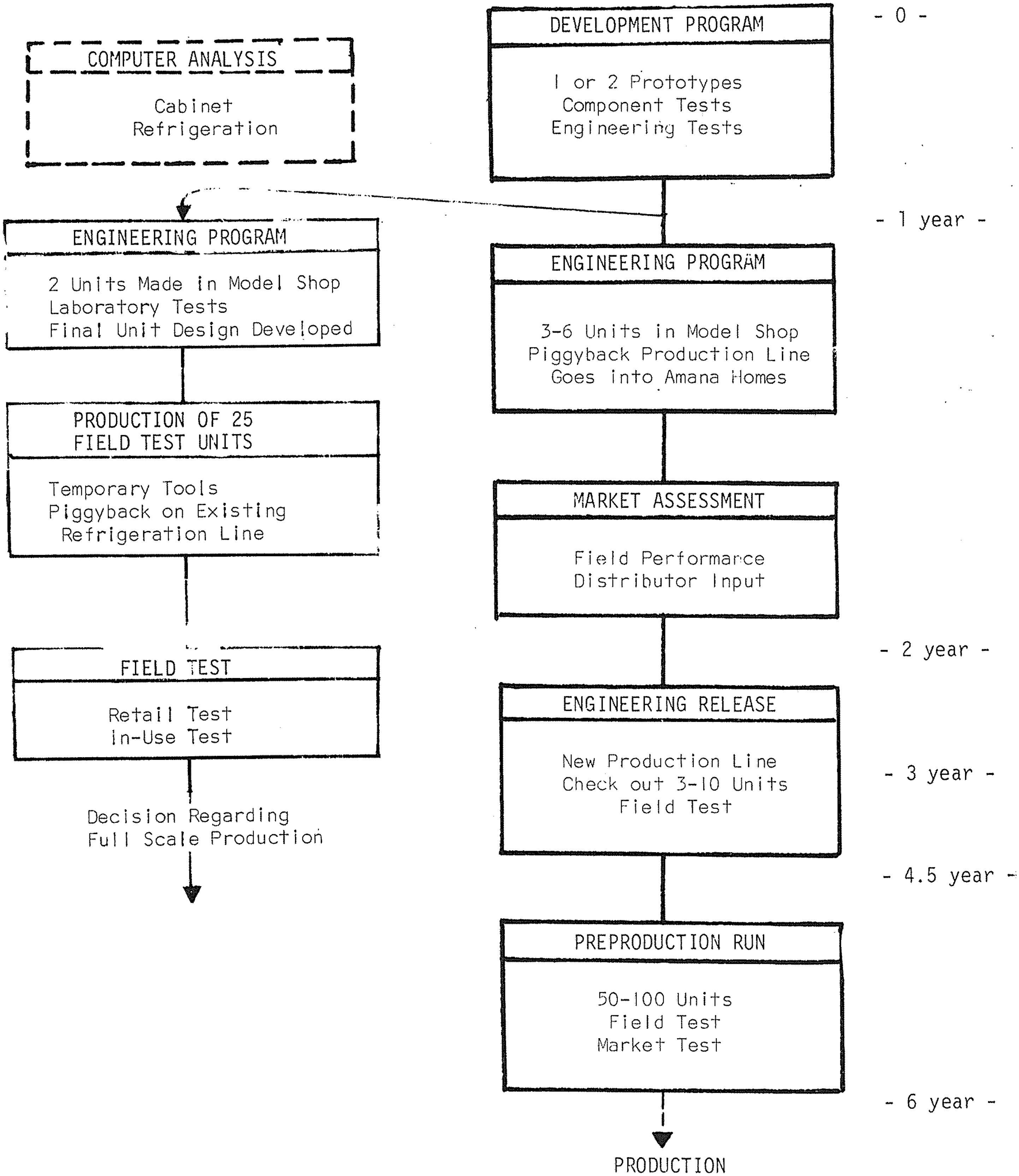


FIGURE PHASE II APPROACH RELATION TO NORMAL PRODUCT DEVELOPMENT

2. SCOPE OF WORK

Based on the experience of Phase I and the goals outlined in the foregoing section, we recommend a field test project, with tasks outlined below, be initiated. The project should be performed (as in Phase I) by an established appliance manufacture and engineering-management company team. The team relations provides the benefits of an advanced product design and analysis in a practical manufacturing framework. Expedient computer aided design analysis should be employed to establish optimum designs guided by many years of manufacturing and marketing experience.

A. SCOPE OF WORK

Task II.1: Specifications and Manufacturing Facility

Design, fabricate, and subject to a complete set of engineering tests two high-efficiency refrigerator-freezer units. Specify pilot production tooling and provide specifications to tooling vendors for at least the following items:

- plastic liner mold, including doors
- gaskets
- shelves and crisper

Task II.2: Manufacturing, Testing, and Demonstration

Manufacture approximately 25 energy-saving units. Perform quality control inspection equivalent to standard practice, and test a random sample of units according to the standard DOE test procedure (90°F closed door test). Instrument each unit for field tests. Test a unit in the laboratory to monitor durability and degradation.

Purchase 25 baseline units for a field test comparison with the 25 high efficiency refrigerator-freezer units. Develop a method for selecting representative families, place units in households, check out instrumentation, and monitor the energy consumption of the baseline and high-efficiency refrigerator-freezer units during one year of use and assess the energy savings of the high efficiency units.

Wherever possible, these units should be placed through a retail marketing effort.

After one year, assess:

- energy consumption;
- maintenance and service requirements;
- user and market acceptance.

Task II.3: Monthly Reports and Project Management, and Final Summary Report

Prepare and submit monthly reports and oral reports to the ORNL-TM concerning program progress. Prepare and submit a draft final summary report containing an executive summary and Task Reports 6 and 7. Submit a final summary report reflecting resolution of comments from the ORNL-TM on the draft.

Task II.4: Long-Term Surveillance

Evaluate the long-term performance of the demonstration units at a reduced, but adequate, level of surveillance to document the important characteristics, such as annual savings, problems, reliability, maintenance, and user acceptance of appliances. The duration of this task is not specified and will be negotiated annually as long as appropriate.

B. MANUFACTURER SUBCONTRACT (An Amplication of the Manufacturer Portion of the Scope of Work

Task II.1: Specifications and Manufacturing Facility

Fabricate and test two high-efficiency refrigerator-freezer units and develop specifications for pilot production tooling.

Task II.2: Manufacture, Testing, and Demonstration

Manufacture approximately 25 units (exact number will depend upon tooling nd facility constraints), perform quality control tests and sample test for energy consumption.

Deliver the units to the designated retail outlet, manage the retailing of the high efficiency units and their delivery to designated home test sites.

Provide any required service and maintenance of the units during the test period.

3. SCHEDULE

The program schedule for Phase II, including deliverables, is shown in Figure 2.

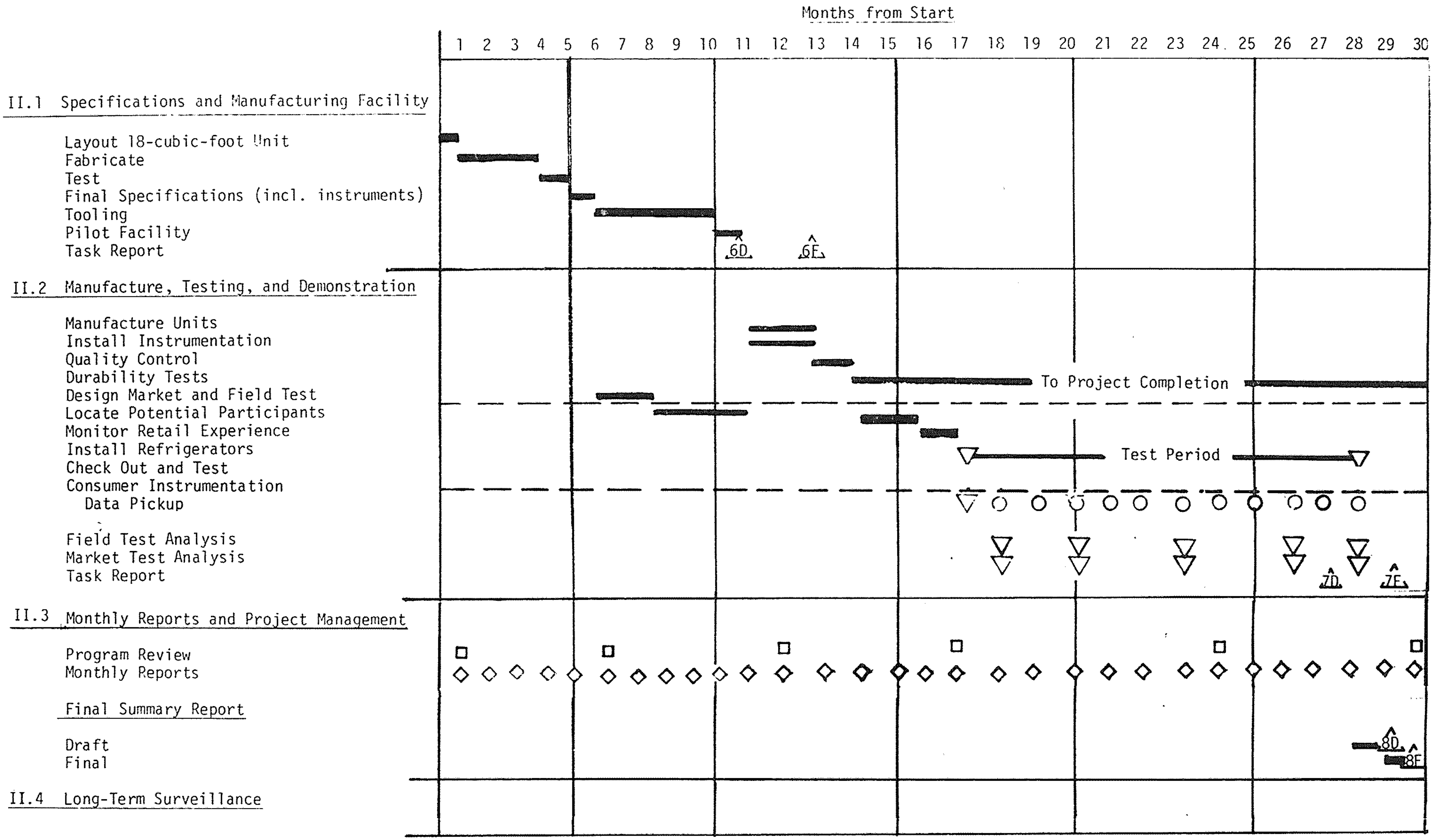


FIGURE 2 PHASE II DEMONSTRATION PROGRAM SCHEDULE

4. GENERAL APPROACH

To achieve the goal of demonstrating the energy savings, maintenance and service requirements, and consumer acceptance of the high-efficiency refrigerator-freezer, 50 (25 baseline and 25 high efficiency units) field test units will be monitored during marketing and kitchen use. Features of this field test program are discussed below.

One should undergo standard engineering tests at the manufacturers facility while the other unit should be equipped with the instrumentation package and tested at the engineering-management company. The results of both tests should be used to define the field test model specifications.

Pilot production tooling and a pilot production facility will be developed and 25 prototype high efficiency units will be manufactured. These units, along with 25 baseline units, will be placed in a selected city. The units will be instrumented to monitor energy consumption. We believe that 25 units will provide a sufficiently large sample so that comparative energy savings can be evaluated without stipulating the usage patterns.* Field test results relating usage to energy savings will be reconciled with predicted usage performance developed from the laboratory tests of the pre-pilot production unit, if necessary.

The units will be placed through a market test by offering the units through a normal retail outlet. Customers interested in the unit will be informed about the program and asked to participate. As many of the higher efficiency units as possible will be placed in this manner. The remaining high efficiency units and the baseline units will be placed through a telephone survey of present refrigerator owners in the area. This customer contact will also be useful in assessing general attitudes about energy-efficient refrigerators and kitchen space constraints (the high-efficiency unit is bigger than a standard unit). The test market program will provide valuable information on the factors controlling market acceptance. The evaluation of the kitchen space constraint, and market acceptance factors are considered by Amana to be key elements in the evaluation of the unit.

*See Section VI for a detailed discussion of sample size.

5. TECHNICAL DISCUSSION

A. TOOLING SPECIFICATIONS AND MANUFACTURING FACILITY

Necessary tooling should be designed and a pilot manufacturing facility planned to produce 25 high-efficiency 18-cubic-foot units for the field test demonstration. New tooling will be needed to manufacture the following parts:

- plastic liner
- crisper pans
- crisper top
- refrigerator liner top
- plastic center mullion
- fan housing scroll
- compressor support
- door panel
- outer case weldment

Desgin specifications for the 18-cubic-foot unit cabinet and refrigeration unit should be developed and evaporator and condenser designs to optimize the heat exchanger size versus unit kwh/day energy consumption evaluated. A computer model suppo-ted by laboratory tests of the present 16-cubic-foot Phase I prototype should be used to develop these design specifications.

B. PRE-PRODUCTION BUILD-UP AND TESTING

Prior to the pilot production run of 25 units designed as an extension of the Phase I 16-cubic-foot prototype into the 18-cubic-foot size, two prototypes should be fabricated for development testing. The 18-cubic-foot units to be used in Phase II are not expected to differ substantially in design from the Phase I 16-cubic-foot prototype, as shown in Table 1.

One prototype will be subjected to standard manufacturers' testing which should include the following tests:

1) 107°F room pull down;
2) 65°F room low ambient;
3) Gulf States (door opening and high humidity);
4) AHAM energy test (closed door 90°F)

The other prototype and a baseline unit should undergo field instrumentation checkout, refrigeration unit performance checkout, and durability testing. The following elements should be examined on the prototype for input to the computer program:

- Fan and airflow performance
- Cold plate heat transfer, frosting characteristics
- Forced air heat exchanger performance
- Condenser performance

TABLE 1

PRELIMINARY ESTIMATES OF PREDICTED PERFORMANCE

Phase I - 16-Cubic-Ft Model
Phase II - 18-Cubic-Ft Model

Model	Overall Dimensions Height	Overall Dimensions Width	Overall Dimensions Depth	90°F Room Cabinet Heat Flow	90°F Room Performance %RT*	90°F Room Performance kwh/day
16-cu-ft	65.0	32.0	27.0	193 Btu/hr	33	1.8
18-cu-ft	65.0	32.0	29.8	203 Btu/hr	35	2.0

*RT - Run Time

Field test unit design specifications should be made on the basis of both sets of tests and subsequent computer analysis.

Sample kwh meters which will be used in the field demonstration should be used in the laboratory program to verify accuracy and reliability prior to production. Conceivable modes of failure should be examined during this check out testing and corrective action taken for any failures identified.

C. PLACEMENT OF REFRIGERATOR FIELD TEST UNITS

1. Placement Schemes

Alternative schemes are possible for the field placement of the prototype and baseline refrigerator-freezer models. The chosen distribution pattern should optimize project payoff for all aspects of the demonstration. In particular, the following should be addressed: energy consumption and efficiency, reliability, performance, safety, costs, and other marketability aspects determined to be important in promoting use of the units. Three basic distribution schemes have been identified:

a) Distribution by survey: In this scheme, units would be placed in homes identified through a survey of Amana warranty cards as being appropriate to the test and meeting the geographical and usage criteria established. The advantage of this site selection process is that the sample can be closely controlled and any resulting variability can be minimized. The disadvantage of this scheme is that no real information concerning marketability and customer acceptance of this product in the consumer marketplace can be ascertained. Therefore, this procedure should be used only if it is decided that costs and market data should be sacrificed for precise usage-by-class performance data.

b) Distribution by retail sales: In this procedure, retail sales at selected dealers would be used to distribute the entire sample of 25 baseline and 25 prototype units. A prototype unit and a baseline unit could be displayed, side by side for instance, in a retail appliance store and actual sales techniques would be used to fill the field test quota. The advantage of this procedure is that the true market potential of the prototype unit in comparison to the baseline unit could be ascertained and certain sales and marketing techniques could be applied to determine customer preference and concerns. The effect of prime differential on prototype sales could also be determined. However, this scheme poses some problems. The test program planned requires that the field test units be placed in the field as soon as possible and all at about the same time. Waiting for the required number of acceptable customers to "purchase" the field test units could be intolerable. Also, appropriate sales training, advertisement, and sales incentive (commission) must be provided to have a realistic and meaningful sales test.

c) <u>Combined survey/sales placement scheme</u>: In this third alternative, both the prototype and baseline units would be placed on display in a retail store and "sold". Purchasers would not pay for the units at that time and would be told that the final arrangements will be made in 4 weeks, at which time the "purchasers" would be told about the program and asked to particiate. Any remaining prototypes or baseline units would be placed in homes with the desired characteristics for testing energy consumption. A population of Amana customers identified through warranty data again would be utilized for the survey-identified test group.

The following arrangements should be established with the participating homeowners:

- They release ADL, Amana and the participating dealer from product liability for the 50 units during the duration of the contract.
- The unit they "purchased" (selected) would be removed after one year and they would have the opportunity to purchase an equivalent unit at a reduced cost.
- They will permit a meter reader into the house regularly.
- They will permit the installation of a recording instrument for approximately 1-6 months.

- They will permit exchange of an equivalent refrigerator unit after six months if we elect to switch high efficiency and baseline units among the participants in order to improve the comparison as discussed in Section VI.
- They will not tamper with elements in the machine compartment.

2. Number and Location of Test Sites

The sample size must be large enough to be reasonably representative of the user population. We feel that a sample whose mean has a 95% probability of estimating the population mean to better than ±10% is a suitable sample size. Using an assumed distribution of energy consumption for a large population located in a single city, we calculated the probable standard deviation and concluded that 25 samples would provide a mean estimate within ±7% of the actual population mean. Supporting data and further analysis of the sample size and its likely impact on the quality of the field test data is given in Section VI.

As the available number of prototypes (25) is only sufficient to be placed in a single city, the site selection is difficult. A population-weighted degree day location would be in a central state; however, we also desire some units to be located in a Gulf state to evaluate the high humidity performance. The final selection of the site location should consider the size and history of the local market to include a retail population large enough to place the 50 units in a 1 or 2 week period.

D. FIELD TEST PERFORMANCE AND INSTRUMENTATION

Compact kilowatt-hour meters should be installed in all units and the energy consumption monitored over a one-year period. If the usage variation is as expected, then the mean of each sample size will be used to compare unit performance. In the event that substantial energy consumption variation is measured among the two sample populations of 25, and meaningful comparisons between the two populations of units cannot be made, then a task for reducing the statistical variation could be undertaken at the request of UCC-ORNL involving additional field instrumentation.

This task might involve the following: six selected units would be instrumented to measure and record energy consumption, several temperatures, and number and duration of door openings with a continuous recording system. After the first month of test the three units representing the median and the two extremes in energy consumption for the baseline and the prototype refrigerator populations would be instrumented with these six recording units. These units would be shifted as necessary to maintain a monitor of the extremes and a mean.

The additional information gained would reduce the statistical variation of the population by:

- allowing for correction of wide usage variations by measuring usage
- indicating unit malfunctions.

The data acquisition system used in the laboratory test would also be used for analyzing the field test data tapes from these six units.

6. FIELD TEST SAMPLE SIZE

Three estimates of refrigerator energy use variation were used to determine the sample size necessary to estimate the expected mean power consumption of a future large population of high-efficiency refrigerators. The estimates used were:

- NBS field test data
- Midwest Research field test data
- ADL estimates of unit performance.

The field test data from both sources was neither a random nor a systematic sample (in the statistical sense) representative of a given segment of the refrigerators in the U.S. The data were a collection of various types, brands, and sizes of refrigerators which caused large variations in energy consumption. Since we will use a single size, brand, and type of refrigerator, estimates based on these data will tend to overstate the required sample size. In another attempt to approximate the likely variation of the two samples we estimated the range of power consumption values we expected to find in our field test; then calculated the required sample size.

The following sections outline each analysis and sample size determination. It should be noted that these calculations of sample size are *estimates* and are only as good as our assumption that the standard deviation of our test will be equal to that of the past tests.

NBS Study

NBS instrumented 22 refrigerators in the homes of their staff in 1974 to measure daily energy consumption and other data. The test sample was comprised of various unit types, brands, ages, and sizes. Our analysis of these data produced a required sample size of 145 to arrive at a sample mean energy consumption within $\pm 5\%$ of the "true" mean at the 95% confidence level (i.e., with a sample size of 145 units, we would be 95% certain that the mean energy consumption we estimated in the test would be within 5% of the mean of the population). By limiting the NBS sample to the seven 15-cubic-foot automatic defrost units instrumented (of different brands), our required sample size dropped to 78. The dissimilarities of the refrigerators and test conditions remain significant even in this sub-sample. We therefore believe that the sample size can be much smaller for our test program than that predicted by the NBS data.

Midwest Research Institute Study

In 1975, Midwest Research Institute conducted a program for DOE entitled, "Patterns of Energy Use by Electric Appliances Study". As part of this program, 118 refrigerators in sample households in different regions were instrumented to measure electric power consumption. Again as in the NBS study, the refrigerators monitored were different sizes, ages, types and brands. Therefore, application of these results to our test should also overestimate our required sample size. Analysis of the MRI data gives a required sample size of 37 to arrive at an estimate of mean energy consumption within 5% of the "true" mean at the 95% confidence level.

Analytical Sample Size Determination

An additional evaluation of the sample size could be made by assuming the mean power consumption and standard diviation of the baseline and high-efficiency units.

Choosing an Amana TC-18W refrigerator as the baseline with the mean energy usage of 4.5 kwh/day and assuming that customer usage patterns are such that virtually all of the unit population lies within the energy consumption range of 3-6 kwh/day, then the standard deviation is:

$$\sigma \simeq .5 \text{ kwh/day.}$$

We want a sample size sufficiently large to assure that within 95% confidence the mean, $\bar{x}$ of the field test is within 5% of the mean of a large future population. Therefore, the precision is:

$$e = .05\ (4.5) = .225 \text{ kwh/day,}$$

and the required sample size is estimated to be,

$$N = \frac{Z^2 \sigma^2}{e^2}$$

where:

N = sample size
e = precision
Z = number of standard deviations from the mean required to achieve the confidence level. For a standardized normal distribution and a 95% confidence level: Z = 1.96.
σ = standard deviation.

and therefore:

$$N = \frac{(1.96)^2 \ (.5)^2}{(.225)^2}$$

$$= \ 19.$$

Assume the high-efficiency model has a mean energy consumption of 2.0 kwh/day and that virtually all of the population will lie within a range of 1-3 kwh/day, then the standard deviation is:

$$\sigma \simeq .33 \text{ kwh/day}.$$

Proceeding with a similar analysis as above, the precision is:

$$e = .05 \ (2.0) = .1 \text{ kwh/day}$$

and the required sample size is estimated to be:

$$N = \frac{(1.96)^2 \ (.33)^2}{(1.1)^2}$$

$$= 42.$$

It should be noted that the reason the estimated required sample size is over twice that required for the TC-18W is because the estimated standard deviation for the high-efficiency unit is a much higher percentage of the mean energy consumption, even though it is a smaller absolute number. We see therefore that small absolute differences in power consumption will have a relatively large percentage effect on the high-efficiency units.

The sample size can be reduced if we are willing to sacrifice the precision of our estimate of the mean. If, for instance, our sample size is 25, then:

$$e^2 = \frac{Z^2 \ \sigma^2}{N}$$

$$= \frac{(1.96)^2 (.33)^2}{25}$$

$$= \ .02$$

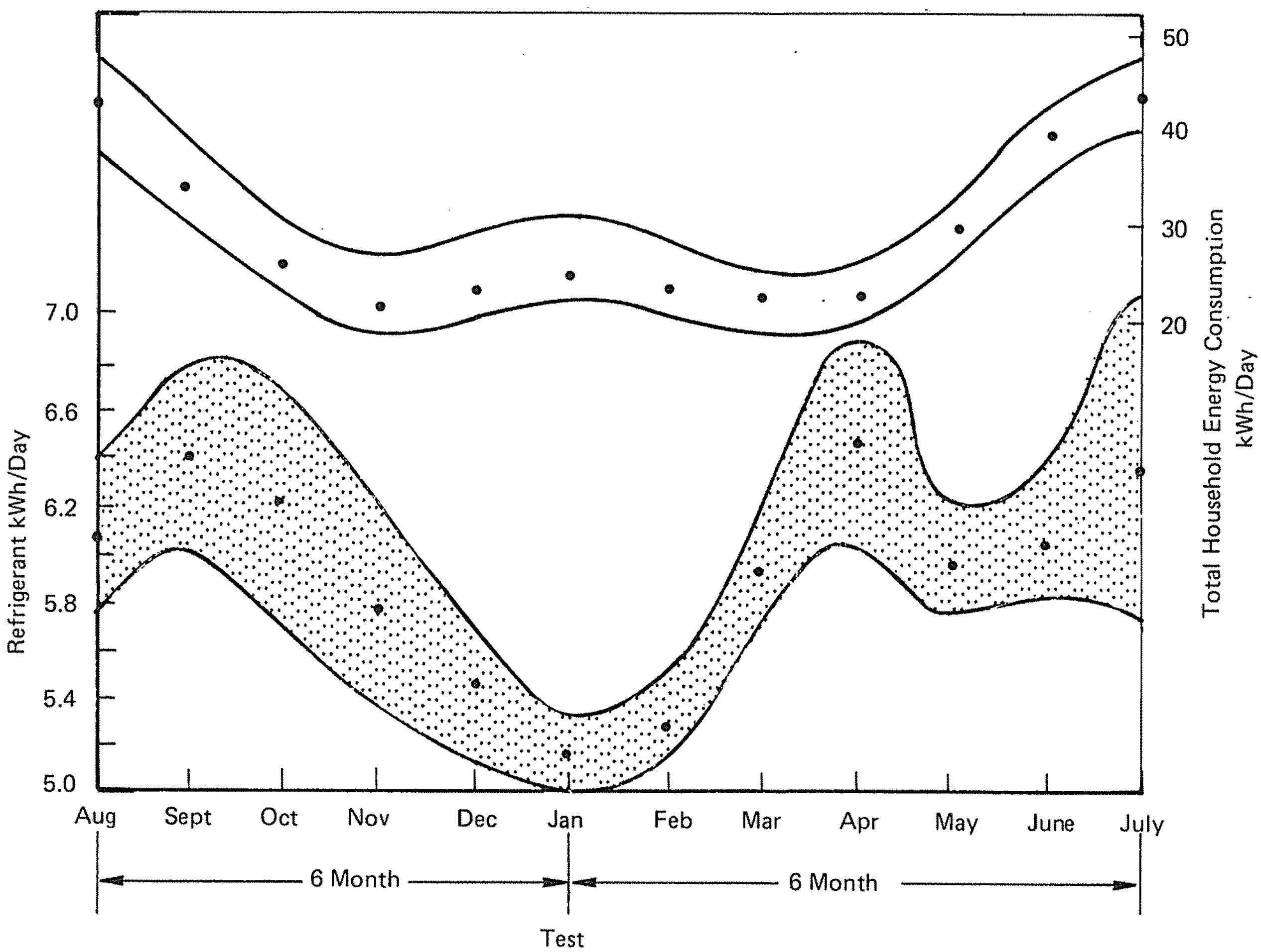
and

$$e = .13 = (2.0)(.065).$$

Therefore, for a sample size of 25, the width of a 95% confidence interval would be only 6.5% of the estimated mean energy consumption. For only a 1.5% loss of precision, we can lower the sample size by almost half.

The precision may be improved by switching the high-efficiency and baseline units between similar users, 6 months through the 1 year test. This should give a back-to-back comparison in each household as well as doubling the effective sample size. Attention to the effect of seasonal energy usage variations is important, and based on the MRI data shown in Figure 4, we tentatively plan to rotate units in January or July depending on the start date of the test.

Switching the refrigerators midway through the test period involves some trade-offs while increasing the breadth of the analyses. The major source of variation introduced by the switch is the seasonal variation within the year. This, however, appears to be minimal if both summer and winter are included in equal proportions within each test period. Perhaps this can be best achieved by equalizing the degree-days for each half of the test. The obvious advantage is that the switching of refrigerators permits a direct comparison within households between the two refrigerators. The additional sources of variation may be measured and their effects accounted for in the data.



REFRIGERATOR ENERGY CONSUMPTION IN THE SOUTH REGION

Figure 3

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