JOINT COMMENTS OF

THE ASSOCIATION OF HOME APPLIANCE MANUFACTURERS,
THE NATURAL RESOURCES DEFENSE COUNCIL,
THE AMERICAN COUNCIL FOR AN ENERGY EFFICIENT ECONOMY,
THE NEW YORK STATE ENERGY OFFICE,
THE CALIFORNIA ENERGY COMMISSION,
PACIFIC GAS AND ELECTRIC, AND
SOUTHERN CALIFORNIA EDISON

RELATING TO ENERGY CONSERVATION STANDARDS FOR REFRIGERATOR/FREEZERS

Docket No: EE-RM-93-801

November 15, 1994

JOINT COMMENTS OF THE ASSOCIATION OF HOME APPLIANCE MANUFACTURERS, THE NATURAL RESOURCES DEFENSE COUNCIL, THE AMERICAN COUNCIL FOR AN ENERGY EFFICIENT ECONOMY, THE NEW YORK STATE ENERGY OFFICE, THE CALIFORNIA ENERGY COMMISSION, PACIFIC GAS AND ELECTRIC, AND SOUTHERN CALIFORNIA EDISON

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TABLE OF CONTENTS

2

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I. OVERVIEW

A. Introduction

The above captioned parties are pleased to present to the Department of Energy the results of two years of intensive effort to negotiate a common recommendation for an energy conservation standard that meets the requirements of the National Appliance Energy Conservation Act for refrigerator/freezers. These efforts have successfully culminated in a complete agreement, as described in this document, which will save 20 billion Kwh/year or 0.23 Quads/year of primary energy by 2010 while preserving the quality and functionality of this basic American product and protecting the economic vitality and competitiveness of a critical U.S. industry.

Attachment 1 is the detailed description of the standards agreement in proposed regulatory language. This agreement is fully consistent with the requirements of NAECA. The standards proposal represents the maximum energy savings which are technically feasible and economically justified by standards in 1998.

In these comments, the parties, and their role as the primary stakeholders on appliance energy conservation issues, will be identified. The negotiation process will be described and some of the energy and environmental benefits of the agreed-to standards will be quantified.

In order to explain the rationale for the standard levels agreed to by the parties, including the standard levels applicable to future products which do not use HCFCs in foam insulation, we first examine the potential standards levels preliminarily identified by DOE's contractor, Lawrence Berkeley Laboratory, as possibly reflecting the maximum energy efficiency technically feasible in a 1998 time frame and industry's engineering, economic, utility and marketing critique. Both LBL and industry's analyses already are in the public record, primarily in oral and written responses to the Advance Notice of Proposed Rulemaking.

Next, we examine potential standard levels which address the issues of technical feasibility and economic justification that cause the LBL "max tech" level to fail the requirements of NAECA. These levels take into account some of the issues raised both by industry and by other parties in their respective critiques of the so-called "max tech" levels and account for the significant technical, economic, and consumer concerns of higher trial standard levels. These trial standards, which are similar to levels of efficiency discussed in the negotiation process, are shown below to fail the test of economic justification. Industry believes that these levels also failed the technical feasibility and "safe harbor" criteria. Then, we describe a standard level based on an approach to technology and risk that the parties believe meets the requirements in NAECA to balance the seven factors of economic justification. From industry's viewpoint, this level can provide reasonable consumer paybacks and acceptable manufacturers impact. Finally, the justification for the standards levels adopted is provided, including a discussion of the energy penalties associated with the phaseout of the production of HCFC 141b, and the relief provided in the regulations, and the special considerations for freezer and compact products.

B. The Parties to the Agreement

The Association of Home Appliance Manufacturers (AHAM) represents the manufacturers of virtually all (over 99%) refrigerator/freezers produced and/or sold in the United States. In particular, the companies active in the negotiations and who support the agreement are: Amana Refrigeration, Inc, Frigidaire Company, General Electric Appliances, Marvel Industries, Maytag Company, Sanyo Company, Sub-Zero Corporation, U-Line Corporation, W.C. Wood Company and Whirlpool Corporation.

The American Council for an Energy Efficient Economy (ACEEE) is a non-profit organization dedicated to advancing energy efficiency as a means of promoting both economic prosperity and environmental protection. ACEEE was very involved in the development and passage of NAECA as well as subsequent appliance standards rulemakings.

Natural Resources Defense Council (NRDC) is a national environmental organization with over 170,000 members and contributors. NRDC has promoted energy efficiency at the state, regional, national, and international level for over 20 years, and has participated in DOE appliance efficiency rulemakings since 1980 and state appliance efficiency proceedings since 1975. The National Appliance Energy Conservation Act of 1987 follows from an agreement negotiated between NRDC and the major appliance manufacturer trade associations.

The New York State Energy Office (NYSEO) was created in 1976 in the wake of the nation's first energy crisis to help guide New York to a more sustainable energy future by developing sound energy policies and promoting energy efficiency. The Energy Office started promulgating state appliance efficiency standards in 1978. The Office has participated in the development of national appliance efficiency standards since their inception.

The California Energy Commission (CEC) is California's energy planning agency, responsible for licensing power plants, establishing efficiency standards for buildings and appliances, and encouraging the development of more efficient and renewable energy resources. Since 1977 the CEC has adopted standards for appliances sold in California. The CEC supported NAECA, despite the state preemption, because it provided economic and environmental benefits to the nation, as well as to California. The move toward national standards has also fostered collaborations such as this effort. CEC states that the resulting refrigerator proposal is consistent with California's policy to reduce the cost of energy services to consumers by supporting the adoption of national standards that are both technically feasible and cost-effective to consumers.

Pacific Gas and Electric Company (PG&E) is the nation's largest investor-owned power utility. PG&E serves central and northern California. PG&E has been heavily involved in energy conservation activities since the mid-seventies. Because of the energy its customers have saved from these activities, PG&E has been able to postpone and cancel plans for power plants.

Southern California Edison is the nation's second largest electric utility, based on the number of customers. The 107-year old investor-owned utility serves more than 4.1 million customers in Southern and Central California. Its 50,000 square mile service area has a population of nearly 11 million. It has been active in energy conservation measures for many years.

Since the 1970's these parties have been the primary parties in DOE and state appliance standards, research and development, utility incentive and demand side management activities. They represent a broad spectrum of interests and points of view.

C. Rationale For Negotiations

The parties entered into informal discussions regarding 1998 refrigerator/freezer standards because of the experienced disadvantages of a normal NAECA rulemaking and their expectations regarding the advantages of a different approach. Rulemaking procedures, even informal rulemakings as are conducted under NAECA, tend to cause participants to take relatively rigid, adversarial, and ideological positions. The free exchange of information and the ability to enter into constructive dialogue are limited. In contrast, negotiations offer the opportunity for open, candid, and collegial in-depth discussion and exchanges of ideas and data. Innovative regulatory approaches, as reflected in this agreement, can be developed.

In addition, previous NAECA rulemakings had the disadvantage of providing only the legally required minimum lead time notice of the regulatory requirements. Planning, research and development, testing and investment in costly advanced technology require long notice and lead time. (For these products, NAECA requires a three year lead-in between the final rule publication and its effective date and there is a mandatory 5-year lock-in between the effective date of the 1993 standards and a revised, 1998 standard.) Industry benefits from the certainty associated with knowing the level of efficiency investment that will be required farther in advance: these benefits are likely to translate into minimizing cost increases to consumers and the maintenance of high quality, long-lived products.

This agreement may provide an additional year or more actual notice for manufacturers who will expend hundreds of millions of dollars to comply with these standards. In light of the January 1, 1996, phaseout of CFCs used as refrigerant and in insulation and the January 1, 2003 phaseout of HCFC-141b, the initial foam blowing agent substitute for CFC-11, the tooling and design changes anticipated for the 1998 standards will be far more costly, complex, and challenging than those needed for the 1993 standards. Therefore, a final rule identified three and a half to four years before the implementation of the 1998 standards will give manufacturers much needed extra time necessary to comply with this and related EPA stratospheric ozone protection regulations.

Finally, this standards agreement reduces the risk that technical errors are inadvertently built into the energy standards equations or product class definitions. Serious errors of this type in the refrigerator/freezer standard which would have gone into effect in 1993 threatened to destroy the viability of entire compact and freezer categories until DOE issued a technical correction. Some non-industry parties believe that this "technical correction" was procedurally irregular and did not afford the public the opportunity of comment.

From the non-industry parties' standpoint, negotiations provide them with unprecedented access to critical engineering, product and economic information which allowed the parties to make more informed judgments as to the economic feasibility of different levels of energy efficiency, as well as the manufacturer concerns in achieving energy gains through one approach rather than another. Negotiations also allow them to invest their resources in energy conservation activities other than prolonged rulemakings. It can also result in less delay in issuing standards and thus earlier effectiveness dates. From a state viewpoint, earlier decisions offer more certainty in the planning and resource evaluations that provide the foundation for state energy policies.

From the federal government viewpoint, encouragement and support of this process are beneficial and substantially support its obligations and goals under NAECA. Considering DOE's enormous statutory mandate, informal agreements by a wide range of stakeholders which comport with the standards-setting criteria in NAECA have major public policy benefits.

D. The Negotiations Process

The negotiations process commenced in July 1992 and agreement was reached in August 1994. Over a dozen meetings and conference calls were held between the parties, and industry, for example, met internally on an additional 20 occasions. An estimated 15,000 - 20,000 hours of effort went into the development of the standards.

The reason that effort was so resource -- and time -- intensive is because extensive discussions were held and proposals were empirically based, relying on data and analysis developed by the parties and LBL. LBL analysis, which is in the public record, was critical to the success of the effort and is much appreciated by all the parties, even where it was not agreed to by all the parties. Before standards were discussed, industry and LBL spent many months reviewing and revising the ERA refrigerator engineering model, gathering technical and economic data, and drafting and critiquing detailed engineering analyses. The products of this work were placed before all parties as the basis for standards discussion. All standards considered and the final standards are based primarily on these analyses, applying the relevant criteria in NAECA. The negotiations and the agreement, however, specifically relate only to the refrigerator/freezer rulemaking and create no substantive precedents for other DOE appliance standards actions.

II. CONSIDERATION OF LBL'S MAX TECH ASSESSMENT

As a starting point for consideration and debate by the parties, a "max tech" analysis of the five auto-defrost product classes was developed by LBL. This analysis was developed from industry and non-industry data input to the ERA computer simulation and predicted potential energy reductions of 45% to 49% below the 1993 NAECA standards on these products (See Attachment 2).

The industry and non-Industry parties have different responses to this LBL analysis. But all parties agree that this LBL "max tech" level would not meet the criteria for economic justification under NAECA, and that DOE must therefore reject this level as a standard. Industry also believes these supposed "max tech" levels were not technically feasible and are not, therefore, "max tech".

A. Industry Response

Industry provided directly to LBL and in the ANPR process a detailed response and critique of the reality and viability of these so-called "max tech" levels. The analysis drew from a statement of principles consistent with NAECA. These principles are:

1) New standards must result in a reasonable payback to the consumer.

2) New standards must be able to be implemented in a technically and economically manageable timeframe.

3) New standards must not dictate the use of technologies that reduce utility, quality or reliability of the product.

4) New standards must be sustainable and capable of manufacturing in a large volume, mass production environment.

While there were dozens of issues associated with the assumptions and conclusions in the LBL analysis, the major portion of the industry critique focused on:

a) ERA accuracy
b) Design option feasibility/marketing utility
c) Impact of increased wall thickness
d) Variance and uncertainty in LBL assumptions, including cost assumptions
e) Additional obsolete design options
f) Unrealistic cost assumptions
g) Other significant problems in max tech analysis

a) *ERA Accuracy*

The first and most basic issue is the accuracy of the ERA computer model used to project energy savings. The ERA is not only used to estimate energy savings of a particular refrigerator/freezer design, but data output from ERA is used to project several different types of cost analysis. Therefore, the accuracy of the ERA model affects the viability of energy savings attributable to individual and groups of options, and when used with the costs generated for those individual options, ultimately determines the credibility of projected life cycle costs, years for consumer payback and cost of conserved energy estimates. Understanding the accuracy of the ERA model is critical to understanding limits on and drawbacks of the accuracy of all subsequent economic and technical analyses which rely on this basic analysis.

Industry's assessment of the accuracy of the ERA model was divided into two phases. The first phase was to use current technology and currently available products to determine the accuracy of the ERA estimates versus actual energy data from refrigerator/freezers. The second phase of this assessment was to determine how the ERA model handles non-conventional technologies, e.g., those technologies not currently in production. Phase 2 was executed through a consultancy with the University of Illinois.

Manufacturing members constructed 100 ERA input files on products ranging from compact refrigerator-freezers and freezers to full-size automatic defrost refrigerator-freezers. Several major errors were discovered in the ERA model through this exercise. The fact that the ERA model was incapable of modeling freezers or compact refrigerator-freezers with a single door was also identified. Several revisions to the ERA model took place during the last two years to accommodate modeling these types of products. The accuracy of the ERA model on the remaining refrigerator-freezer files was (in terms of standard uncertainty) $\pm 19\%$ (see

Attachment 3). While this accuracy level makes the ERA useful to examine engineering assessments of potential energy savings options, it is not a sufficient tool to determine multi-million dollar rulemaking impacts. Practical engineering judgements also must be heavily weighted even if they do not result in easily manipulated quantifiable outputs.

Phase 2 of the ERA analysis was performed by Dr. Clark Bullard at the University of Illinois' Air Conditioning and Refrigeration Center. This phase of the analysis on the ERA model focused on non-conventional technologies, that is, technologies that have yet to be built into full-size refrigerator-freezers and tested or are not yet currently in production. Dr. Bullard's final report noted that many of these design options being modeled in the ERA model had errors between 50-75% compared to actual results taken from these technologies being researched in some manufacturers' laboratories (see Attachment 4).

The ERA model is a design model and not a simulation model, i.e., it can estimate energy impacts of design inputs, but cannot predict the impact of those inputs on other design aspects. It does not take into account that energy driven design options may not keep food from spoiling. Furthermore, even if all inputs and equations were known with perfect certainty, the model would only be capable of representing performance at a single operating condition. Refrigerators must accommodate voltage, usage, and ambient variations under many different operating conditions. The accuracy of the output of the ERA model also is dependent upon the qualifications of the personnel using it. Recognizing the limitations of the ERA model is critical to understanding the variation inherent in the life cycle cost analysis that utilizes ERA's and LBL's estimates.

b) *Design Feasibility/Marketing Utility*

A list of 30 various design options and sub-options (41 total options) identified by LBL was reviewed by AHAM manufacturing members in the fall of 1992. This list of design options was inclusive of energy savings options already implemented by manufacturers as well as many theoretical technologies in various stages of feasibility assessment. Based on industry input, seven options were qualified as having 0% probability of implementation. Twelve additional options were identified as having a probability of implementation between 0 and 50%. The remaining 11 options were identified as having feasibility of 50 to 100% (see Attachment 5). LBL staff also recognized the marginal feasibility of many of these options which were dropped from additional consideration. Comments provided to the DOE at the January 6-7, 1994 ANPR hearings, as well as the written comments supplied to DOE on February 1, 1994, contain detailed analyses and support data on the design feasibility, marketing utility and economic impacts of these proposed options.

Among the areas of significant industry challenge to LBL's max tech findings were the marketing utility of design options, particularly marketing and salability impacts of increases in wall and door thicknesses, realistic projection of available compressor energy efficiency ratios (EER) for the 1998 timeframe, realistic application and costing of vacuum panels to refrigerator/freezers, costing and availability of high efficiency motors in the 1998 timeframe and ERA computer simulation run time percentage necessary to project whether refrigerator/ freezers can maintain food quality.

A major area of dispute with the "max tech" assessment was the overly simplistic analysis performed by LBL in developing Life Cycle Cost curves. These curves are developed from the application of assumed costs of design options applied to ERA computer simulation outputs. This assessment ignores CFC and HCFC phaseout, marketability, reliability, and the uncertainty associated with achieving a projected energy savings level. The analysis also used a 19-year product life for refrigerators which is inaccurate. National Family

Opinion (NFO) survey data indicate the product life is 15 years. An adjustment of product life in an LCC analysis affects total and minimum life cycle costs. LBL also used discount rates which are the subject of disagreement between the parties.

c) *Impact of Increased Wall Thickness*

One of the design options which has significant impact in a standards analysis is the application of additional foam in walls, resulting in thicker walls with decreased inner volume or increased external dimensions. The LBL max tech and other analyses rely heavily on increased wall thickness utilizing this option for 30% of the total energy savings. It was recognized that, alternatively, using vacuum panels raises serious technical feasibility and cost issues.

The major issues involved in the debate on increasing the insulation thickness in a refrigerator wall can be summarized as follows:

- Increasing the wall thickness has been identified as the option providing the greatest energy savings.

- Manufacturers have stated that an increase in external dimensions on refrigerator-freezers of as little as a half inch can eliminate as much as 20-30% of a marketplace available for that particular product. This is reflected in the attached marketing utility assessment (see Attachment 6). The LBL analysis does not consider marketability of a product or option.

- If the external dimensions are maintained and the wall thickness increase is made to the inside of a cabinet, allowing it to fit through doorways and into kitchens, three things will happen.

    - The smaller volume cabinet sells at a lower price with less margin;
    - The lower volume cabinet has to meet a more restrictive energy standard (a factor not taken into account by the LBL analysis); and
    - This design sacrifices important utility of the product in violation of the mandates of NAECA.

An expensive alternative to thicker walls is *vacuum panels*. Vacuum panel technologies have progressed since the last refrigerator rulemaking. The appliance industry probably will introduce limited vacuum panel design over the next five to ten years. Issues of concern are *manufacturability, availability, reliability* and *in-product performance*. It is still too early in the development of this technology to apply it as a reliable design option in the production of a 1998 compliant product. Several major issues remain unsolved.

- Vacuum panels *must* be used in concert with foam insulation (polyurethane foam is the mechanical support for the cabinet).

- Wire harnesses, drain tubes, shelf anchors, etc., are between the cabinet shell and inner liner making 100% coverage of vacuum panels impossible. 50% to 60% is about maximum and for freezers would even be less.

- Vacuum panels are 6 to 10 times heavier than foam. Panels in doors may compromise UL tipover requirements. The shipping weight of a typical cabinet with vacuum panels would increase by about 50 pounds.

- Polyurethane foam averages about 15 cents per board foot. Powder filled panels are $2.50 to $3.50 per board foot and fiber filled panels, $5.00 to $7.50 per board foot. An average refrigerator-freezer has about 114 board feet of surface area of which approximately 35 board feet would be vacuum panels.

- Worldwide production capabilities for all types of vacuum panels is between 3 to 5 million board feet per year. Full implementation of vacuum panels in the US alone would require over 400 million board feet of panels.

- Product life performance characteristics (15 to 20 years) are being improved but industry concerns continue to work towards a vacuum panel product that maintains reliability over the life of the refrigerator.

d) *Variance & Uncertainty in LBL Assumptions on Design Option Efficiency & Costs*

Another issue related to the LBL "max tech" analysis was that of the variance and uncertainty inherent in the data being used. Industry formulated a number of different approaches for quantifying the uncertainty and variance inherent in estimated energy savings and estimated costs for individual design options. The basis for quantifying uncertainty and variance in the analysis lies not only in the estimates of what energy savings and cost are reasonable in the 1998 timeframe, but it also lies in the different economies of scales available to the companies in the refrigerator/freezer industry. The impact of design options and associated costs affect these companies' products differently, as well as the cost associated with those options to each company (see Attachment 7).

An example from one of the uncertainty analyses (see Attachment 8) notes the variance in unit cost impacts on top-mount non-dispenser automatic-defrost refrigerators in terms of 1 sigma or 2 sigma variance in the expected energy savings from a specific group of options. In this example, in the 30% below 1993 level, a variance in manufacturing unit costs impact runs from approximately $65 up to $145, dependent upon the delivered energy efficiency of the options considered at the 30% level.

e) *Additional obsolete design options*

Research performed by *Consumer Reports* and other organizations providing confidential submittals to the DOE showed the voltage controller did not provide any energy savings on current production models. In fact, several of the current production models tested with this device increased in energy consumption.

Fluid control valves were thought to have energy savings properties when applied to refrigeration systems. However, these valves have only been successfully utilized on rotary compressor refrigeration systems (at this time, there are no plans to manufacture small, HFC-134a rotary compressors in the worldwide appliance market because of lubrication incompatibilities inherent in that type of design). These valves, when applied to reciprocating type compressors, maintain the refrigerant under high pressures outside the refrigerator during off cycles.

Research performed by Oak Ridge National Laboratory (ORNL) under a Cooperative Research And Development Agreement (CRADA) with the Appliance Research Consortium (ARC) indicated that because of higher compressor start up pressures, larger compressor motors were needed. These larger motors negated any energy savings inherent in the use of this valve. LBL agreed that these two options should be eliminated from any further consideration.

f) *Unrealistic Cost Assumptions*

On the economic side of the engineering analysis, many of the costs obtained for components projected to save energy in 1998 era appliances are considered by industry as being overly optimistic for most, if not all, manufacturers. The cost estimates obtained by LBL on one of the high efficiency evaporator motors projected to save energy on future appliances was requested for an unrealistically large purchase quantity. No single refrigerator/freezer manufacturer purchases any individual motor in that quantity. Economies of scale need to be considered when assuming component costs for all manufacturers from the largest to the smallest.

g) *Other Significant Problems in Max Tech Analysis*

The LBL analysis and other analyses supporting standards levels fail to deal with the lack of an economic justification because of the empirical difficulties in cost passthrough (the amount of increased unit costs that would be able to be absorbed in the marketplace by the consumer), buy down effect (the tendency of consumers to buy smaller, lower margin products as a result of increased costs on existing product), and the resultant change in industry value projected from loss of sales from higher product costs and tremendous capital expenditures necessary to manufacture products compliant with the 1998 refrigerator-freezer NAECA.

Much about the economic analysis of the major appliance industry is controversial and not agreed to by all the parties. In fact, while significant data series exist for shipments, reliable retail price information is very difficult to come by, and is not tracked by any major service. Likewise, cost information is carefully guarded among individual companies and not released publicly for many reasons, among the chief of which relates to antitrust considerations. It is unclear how many companies keep historical records relating to cost. Most businesses work from conventional balance sheets and profit and loss statements, where the costs of individual products and product lines are rarely part of the summation. Even where reliable cost information might be available, it is unlikely to be in a readily usable form.

The industry analyzed itself using publicly available data. Essentially it charted the Producer Price Index for refrigerator-freezers versus the Consumer Price Index for the years from 1982 to 1993. Over that period of time, the CPI had risen 46%, while the PPI for refrigerators had increased only 8%. Dividing the two indices into each other, shows that the PPI had effectually declined 26% over time. It was industry's contention that this analysis gave a good indication that new pass-throughs of costs bearing historical mark-ups were highly unlikely and could not be anticipated when judged against recent events.

Furthermore, the US domestic market for refrigerator-freezers is both mature and highly competitive. To date, the manufacturers of these products have remained in business through intensive capital investment, technological innovation, heightened manufacturing efficiencies -- all leading to a general increase in productivity. To meet the demand of this very tough marketplace, companies have, by and large, made use of the most accessible innovations available to them. Additional productivity investments will be costlier, in all likelihood, and have a higher risk of never justifying themselves technologically or financially.

Another area of concern not addressed by LBL was that of the impact of an HCFC phaseout. This impact is mainly focused on the use of HCFC-141b in foam insulation. HCFC-141b is scheduled for phaseout January 1, 2003. At the present time, all non-chlorinated blowing agents being considered to replace HCFC-141b have a minimum 10-12% energy penalty on refrigerator/freezer performance.

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Based on these analyses, industry believed that LBL's identification of max tech was incorrect and that standard levels in that range are not technically feasible or economically justified.

B. Non-Industry Response

Non-Industry participants agreed with the industry assertions that certain of the LBL efficiency measures, such as the voltage controller and fluid control valves are not realistic energy-efficiency options, and that the savings from these measures should be deleted from the analysis. They also agreed that there is a significant degree of uncertainty associated with the costs and performance of some of the measures included in the LBL analysis. At the LBL "max tech" level, this uncertainty is not symmetric: it would be much more difficult to replace the energy savings from an under-performing measure with some new technology compared to the likely benefits of implementing the same or greater energy efficiency at cost equal to or lower than the LBL assumptions. Industry's Attachment 8 suggests that at somewhere around 30% energy savings, the uncertainty in cost associated with the compliance increases significantly compared with the more modest levels of prospective standards.

The LBL "max tech" level is based on increasing both wall and door thickness by one inch--a two inch increase in side-to-side dimensions of the refrigerator. This would be a significant impact on some products particularly since there is little energy left on the table for manufacturers to compensate for this measure should they find it necessary not to produce a thicker wall products, (for example, in the cases of products whose dimensions currently are as large as possible to fit through doors in existing buildings.) Non-industry participants accepted that if manufacturers should find it necessary to maintain a product's current profile and avoid using increased insulation thicknesses, then too few alternatives are left to compensate for the increased energy impact.

Given the large number of uncertainties, and the fact that an unfavorable resolution of any one of these would make the standard non-cost effective or infeasible, this level should be rejected.

III. THIRTY-FIVE PERCENT TRIAL STANDARD LEVEL

A standard level considered by the parties for the five automatic-defrost products was based on top mount non-dispenser refrigerator/freezers. This standard level took into account much of the assessment of the LBL max tech and resulted in standards approximately 35% below the 1993 standard for these five product classes.

At this standard level, there are several key issues. Technological feasibility issues focus on wall thickness as well as available compressor EER's in the 1998 timeframe, data on non-performing technologies used to support this standard and the real-world energy impacts due to an HCFC-141b phaseout. The basis for economic justification of many of the feasible design options relates to the accuracy of ERA output assumptions and methodology in determining cost assumptions used in life cycle cost analyses.

Industry assessments now focused on:

a) Wall thickness impacts on marketability
b) Projected available compressor EER for 1998
c) HCFC Phaseout impacts.

a) *Wall Thickness Impacts*

The 35% standard level depends in large part on increased wall thickness. At this point, wall thickness increases represented over 40% of the energy savings. An industry analysis of the marketability of larger size refrigerator/freezers showed the percentage of available market versus refrigerator width and height (see Attachment 6). A one half inch increase to a wall thickness dimension results in a one inch thickness increase in the entire cabinet. According to the results provided in the refrigerator/freezer marketing survey, a one inch increase in cabinet width could eliminate anywhere between 8 and 24% of the market being served by that product. In other words, those products would no longer fit that particular kitchen opening (or entry door width) in those households forcing consumers to go to smaller, lower margin products with less available storage space.

Refrigerator/freezers are sold by available internal volume as rated according to FTC regulations. The most costly capital tool outlay expended by refrigerator/freezer manufacturers is that of the creation of cabinetry for household refrigerator/freezers. Maximizing the available internal volume to fit the existing marketplace is the most critical capital assessment. The results produced in the refrigerator/freezer marketing survey (see Attachment 6) were obtained from summaries provided by the refrigerator/freezer manufacturers building automatic-defrost units in the U.S. and is representative of many millions of dollars of individual company surveys and research studies performed over the last 10 - 15 years.

In addition, marketing individuals from each refrigerator-freezer company met in confidential sessions with non-industry representatives and DOE to discuss the real world impacts of marginal size increases on refrigerator-freezers and inequitable impacts on individual manufacturers with particular market niches. Non-industry participants concluded that even one-half inch increases in insulation would not be a feasible design measure for *all* models of refrigerators -- that is, some percentage of products, typically much less than half but a larger number than could be completely ignored -- would be impacted in this way. Thus, some products would have to be designed to achieve equivalent energy savings to those achieved by wall insulation using other energy efficiency measures. At the 35% level, this alternate design process could not be achieved with cost-effective design options. Without the use of vacuum panels, such products would be unlikely to be cost-effective compared to the base case of 1993 standards.

b) *Projected Available EER*

Another area of concern is that of available EER compressors in the 1998 timeframe. LBL's original assumption that a 5.8 EER compressor would be available in 1998 is highly disputed by product and compressor manufacturers. LBL reassessed compressor supplier projections for available EER in compressors in the 1998 timeframe and reduced their estimated available EER from the 1998 timeframe to 5.6. Costs associated with the higher EER compressors in the 1998 timeframe were also revised by LBL. The 5.6 estimate was based on the compressor size currently in use in typical refrigerators. To the extent that this standard level is achieved by efficiency measures that reduce heat loads, smaller and less efficient compressors would be required.

c) *HCFC Phase-Out*

Hundreds of chemicals have been assessed in an attempt to implement the most energy efficient, reliable and environmentally benign substitutes for CFC-11 and 12. The current energy conservation regulations are the reason why an HCFC-141b foam is being used in household refrigerators as opposed to a non-chlorinated blowing agent. HCFC-141b is approximately 2-3% less efficient than CFC-11, however, all other non-chlorinated substitutes available to replace HCFC-141b are expected to be a minimum 10% less energy efficient. It was for this reason, as detailed below, that the parties agreed that a separate tier of standards be developed for HCFC-free products. A 10% relief from 1998 standards (approximately 7% relative to 1993 NAECA standards) is proposed for this second tier of non-HCFC products. This number is based on current worldwide research results comparing the best HCFC-141b foam formulations against all available alternatives for HCFC blowing agents.

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Based on these analyses, the 35% standards levels were not justified under NAECA. There were serious questions about its achievability by all product classes and manufacturers, its potential for adverse impact on product utility, and its inordinate cost impacts on consumers and manufacturers. Uncertainties about a number of aspects of the analysis created a strong risk that the burdens of the standard level would outweigh the benefits.

In particular, there were serious uncertainties as to the level of economic impact on manufacturers and on consumers. Industry participants were concerned that insulation increases would result in a lessening in product utility, and, under unfavorable scenarios of the availability of HCFC replacements, the possibility for adverse life-cycle costs for the standard as a whole.

Analysis performed by AHAM suggested a higher degree of uncertainty concerning the cost of meeting the standards at levels of energy savings greater than 25% to 30%. Some of the analysis of cash flow to the industry suggested the potential for serious adverse impacts a the 35% to 40% savings level that were not nearly so serious at the 25% to 30% savings level. These uncertainties are compounded by the issue of HCFC replacement. Since this standard will be in effect after the 2003 phase-out of HCFC-141b, the level selected must be technically justified and economically feasible in a non-HCFC world. Current analysis suggest that this could only occur at the 35% level with the use of vacuum panels. Vacuum panels are not a cost effective measure according to the LBL analysis, and industry participants argued that LBL's cost estimates per vacuum panels were optimistic and that there are serious technical obstacles to full implementation by industry. (Non-industry parties believe that LBL's cost estimates were too pessimistic)

An additional element of uncertainty is the extent to which increased wall insulation can be used in refrigerators that the public would buy. Data from some manufacturers suggested a very low potential for increasing insulation thickness, and all manufacturers' presentations suggested that at least some of the market would be unable to use this efficiency measure.

For freezers and compact refrigerators, the so-called max tech levels raise questions regarding the ability of industry to produce models meeting these efficiency levels that would sell in sufficient numbers to maintain the industry's viability. Unlike refrigerators, which are present in virtually every American household, these products are largely discretionary purchases. Increases in costs to the consumer are more likely to result in reduced overall sales values. Consumer preferences for products with thinner walls would be more likely to lead to reductions in sales as well.

Balancing the benefits and burdens, non-industry participants believe that the balance was relatively close. The relatively high levels of uncertainty in the analysis coupled with the relatively large adverse impacts should these uncertainties be resolved in the unfavorable direction, lead to a conclusion by non-industry participants that this level lacks economic justification. Industry also had serious technical concerns, as described above.

IV. THE TWENTY-ONE/TWENTY-SEVEN PERCENT STANDARD LEVEL

This section discusses a proposal that was on the table as part of the negotiation process. Following the procedures of NAECA, a level less stringent than the recommended standards level may not need to be rejected explicitly. But, to further illuminate the process by which a consensus was reached that the recommended level best fits the requirements of NAECA, it is useful to discuss why a lower standards level was considered to be less economically justified.

The parties reviewed a standard level which used ERA analysis and criteria for standards which were focused on conservatively achievable technology, three years or less payback, and modest financial impacts on manufacturers. This analysis relied on six design options having the highest probability of implementation with prescribed energy savings (see Attachment 9).

The standard level projected for the top freezer, automatic defrost models analyzed in this effort was approximately 21% below the 1993 standard level for an HCFC-free product design. Equivalent HCFC product were extrapolated to a 27% below 1993 standard level.

This standard level was critically evaluated as not providing the maximum energy conservation which could be justified under the law for the following principle reasons.

a) This proposal was based on a 3-year payback level, disallowing design options with a longer payback period which non-industry parties believe could be economically justified.
b) The design options chosen had the highest probability of successful implementation at conservatively estimated energy savings levels. Less conservative analysis indicates that other design options are reasonably available and that energy savings projections may be unduly conservative.
c) This analysis was developed with the purpose of selecting design options that provided an equal economic impact on all manufacturers. The non-industry parties believe that the law does not require this precise result, although manufacturers' impact must be considered.

One of the insights from this exercise was that individual options affected different companies in different ways. When all seven options were applied to the four companies' ERA input files, the resultant life cycle cost curves varied by about five percent, a considerable amount given the identical options were applied to each analysis.

V. THE PROPOSED STANDARDS

A. Refrigerator-Freezers

The proposed standards (see Attachment 1) are based on a negotiated approach to identifying the maximum level of efficiency that is technologically feasible and economically justified. A negotiated approach may provide slightly different results from those achieved by conventional rulemaking because this NAECA criterion can be satisfied in a more flexible way, providing greater overall energy savings for a given level of impacts.

The process of identifying the appropriate standards levels under NAECA attempts to maximize energy savings subject to the constraint that the economic impact of the standards, both on consumers and on manufacturers, is beneficial on balance.

Impacts on manufacturers are different for different product classes. For product classes representing discretionary purchases, such as some compact refrigerators and most freezers, cost increases due to standards may result in much greater reductions in sales compared to the refrigerator-freezers classes, whose purchase is essentially necessary when a new house is constructed or when an existing product fails. Some design options with perceived consumer or marketing disadvantages, such as increasing wall thickness, are more troublesome for these more discretionary classes of products.

The consumer cost-effectiveness of increasing levels of energy efficiency, as well as the impact of these levels on manufacturers, also depends on the scale on which the product is produced. For those products with the highest production volumes, capital cost increases can be amortized over a larger number of units, resulting in fewer impacts. In contrast, for products with smallest sales volumes capital cost increases will be spread over fewer models and will have a larger impact on product cost. These effects will operate differently for different manufacturers, depending on the mix of their sales.

The negotiation process allowed a forum at which levels of standards could be discussed that maximized energy savings for any given level of impact on industry. This approach provides, we believe, greater energy savings and lower industry impacts than the more traditional rulemaking approach.

As a result, the final agreement (Attachment 1) concentrates the largest energy savings on the five automatic defrost categories, with the very largest percentage reduction in the two classes with highest sales volumes. These five classes represent more than two thirds of the total energy consumed by all refrigerator/freezers. These five product classes represent 85% of the total energy savings generated from the standards.

The parties agreed that in the interest of conserving engineering and capital resources while maximizing energy savings, the greatest changes in design should be concentrated on the largest two product classes -- top mount, non dispenser, and side by side with dispensers -- and not other refrigerator/freezers, freezers or compacts. The negotiation process provided a degree of sensitivity and flexibility to fashion these cost effective arrangements not otherwise available in the traditional arms length notice and comment process. That flexibility permitted the participants, for the first time, to address 1) the cumulative economic impact of individual design options, and 2) the varying severity of that cumulative economic impact upon different product classes and differently situated manufacturers. The negotiation process allowed for a cumulative assessment of impact which, in turn, led to adjustments among various product standard levels in order to better balance the economic impact among manufacturers.

There are five product classes that meet the criteria of automatic defrost refrigerator-freezers: Top-mounted Freezer Non-dispenser, Top-mounted Freezer Dispenser (Ice and/or Water), Side-mounted Freezer Non-dispenser, Side-mounted Freezer Dispenser, and Bottom-mounted Freezer. The only aspect these products have in common is that they contain a refrigerated compartment, a freezer compartment, and automatic defrosting.

Dispensers for ice and/or water through the door affect the performance of Top-mount Freezer models the (the dispenser normally is in the fresh food door) and Side-mount Freezer models (the dispenser is normally in the freezer door) in significantly different ways. Because of this difference, the energy consumption of a Side-mount Freezer dispenser can be higher than a Top-mount Freezer dispenser due to a greater amount of heat transferred through a freezer door dispenser.

Similar design options affect the performance of Top-mount, and Bottom-mount Freezer models in different ways. For example, an improvement in gasket design will save the most energy on a Side-mount Freezer model and the least on a Top-mount Freezer model since the former has more linear inches of gasket than the latter. Gasket improvements have a lesser impact on small volume products of any class versus large volume models for the same reason. *Most manufacturers do not build all product classes or all sizes within a product class. This fact emphasizes the need to maximize the total energy savings while considering the resultant economic impacts to each company.*

B. Compact Refrigerator/Freezers

This new set of classes includes all products less than 7.75 cubic feet (FTC/AHAM rated volume) and 36 inches or less in height. The marketplace and industry recognize products meeting these criteria as a separate niche with special engineering and investment constraints. Much smaller privately held family owned single product companies dominate this market. The economies of scale restrictions placed on these companies are much different than those of the full size product manufacturers and because of the niche products these manufacturers produce, external dimensions of these products are even more critical.

a) *LBL Max Tech Assessment of Compact Refrigerator/Freezers*

Because of LBL's need to use the ERA design model to generate these performance and cost figures, the problems inherent in generating economic and engineering numbers for this product class were amplified. Using the most recent version of the ERA model, the accuracy for compacts is still plus or minus 30% which is well beyond acceptable modeling simulations.

Eleven different options were applied to compact models in an attempt to gauge payback periods and life cycle costs for energy efficient options. The options that were selected for "max tech" consideration included compressor efficiency improvements, motor efficiency improvements (in models that utilize motors), anti-sweat heater improvements (in models that utilize an anti-sweat heater), condenser efficiency improvements, evaporator efficiency improvements, wall and door insulation thickness increases,and gasket efficiency improvements. The only options that were identified by industry as having a high probability of feasibility from a design and marketing aspect were those of improved compressor efficiency and improved motor efficiency (in models having motors).

The other five option areas identified by LBL in their max tech assessment for compacts were identified by the manufacturers as having an extremely low design feasibility or marketing utility when applied to their products. Those options when graded on a scale from 0-100 averaged about 20% (not buildable, not saleable) for marketing utility and design feasibility.

As stated earlier, many of the different compact models (while in different product classes) are similar enough in construction and design within each of the small manufacturing firms that they can be assessed as a single group. LBL's "max tech" assessment for compacts was approximately 25% below 1993 NAECA standards (see Attachment 10).

Increased wall and door thickness options were only applied to one class of compacts in LBL's "max tech" assessment. The case against increased wall and door thicknesses, as elaborated earlier in discussions on the impact of increased wall thicknesses on full size product, has an even more severe impact on compact refrigerators. Marketing considerations of compacts do not allow for an increase in wall thickness since most products are designed for niche or undercounter application with no room for expansion of the cabinet size. Any increase in wall thickness would compromise the utility of the product by decreasing the usable interior volume for a product that already has limited applications in the marketplace.

A similar problem applies to insulation increases in top and bottom panels; this space constraint is recognized in the new definition of the compact class as limited to models below 36 inches in height.

These new set of compact classes include all products less than 7.75 cubic feet (FTC/AHAM rated volume) and 36 inches or less in height. The marketplace and industry recognize products meeting these criteria as a separate niche with special engineering and investment constraints. Much smaller, privately-held, family-owned, single-product companies dominate this market. The economies of scale restrictions placed on these companies are much different than those of the full-size product manufacturers and because of the niche products these manufacturers produce, external dimensions of these products are even more critical.

b) *AHAM Life Cycle Cost and Payback Assessment of Compact Refrigerator/Freezer*

The five compact refrigerator/freezer manufacturers supplying data for life cycle cost and payback analysis identified a "max tech" limitation to their products of approximately 15% below 1993 levels. This level did not take into account economic justification (consumer and manufacturer) or safe harbor issues. This assessment took into account the following (Section ________ of NAECA, see Attachment __________):

- High efficiency compressors of 5.5 EER are not realistic for compact refrigerator/freezers. Low capacity compressors available for compact refrigerator/freezers in the 1998 timeframe are expected to have efficiencies of approximately 3.6 EER.

- Most compact refrigerator/freezer manufacturers are small companies with limited research and development funding and capital resources.

- High efficiency foams require high pressure impingement systems that are only economically viable for very large manufacturers. Most compact manufacturers use what is known as an auto froth foaming system (low pressure) that cannot produce high efficiency foam insulation. Non-CFC auto froth formulations are also limited to moderately energy efficient replacements.

- In most cases, compact refrigerator/freezers and freezers do not employ fan motors, mullions, auto-defrost or through-the-door features. As a result, design strategies which relate to these components or technologies are not available for improvement.

- The need for high efficiency components by compact refrigerator/freezer and freezer manufacturers carries a low priority with component suppliers. Motor and compressor manufacturers apply their engineering resources to larger volume manufacturers leaving the low volume niche type compact refrigerators to the tail end of their design cycles. For example, there are compact manufacturers that still have not been provided with sample non-CFC-12 compressors that provide acceptable energy efficiency to household appliance applications.

Compact manufacturers analyzed the 30 design options and sub-options (41 total options) identified by LBL as applicable to their products. These options were quantified and ranked from 1-41, relative to their design feasibility, energy savings and marketing utility. Of the 41 options and sub-options identified, 12 were identified as not applicable to compact refrigerator/freezers, 26 were identified as having less than 50% design feasibility/marketing utility, and only 3 options were identified as having a design feasibility/marketing utility of over 50% (see Attachment 12). Those three options were: improved gaskets, improved compressor efficiency and improved fan motor efficiency (on those products that have fan motors).

c) *Proposed Standards for Compact Refrigerator/freezers and Freezers*

Because of the special design constraints and limited number of options applicable to compact refrigerator/freezers and freezers, it was difficult to develop life cycle cost analyses that reflected the real marketing situation for these products. An LBL assessment using inputs from AHAM compact manufacturers showed that an energy savings level of 2-3% below the 1993 standards would result in a minimum five-year payback for consumers. This assessment did not take into consideration unique marketing restrictions of individual compact refrigerator/freezer and freezer manufacturers.

The engineering, marketing and economic realities of the compact refrigerator/freezer and freezer manufacturers can easily be put into perspective by understanding that the total energy consumption of all compact refrigerator/freezers and freezers in the U.S. is less than 2.6% of the total energy consumed by all refrigerator/freezers and freezers. The inaccuracy of the ERA model in simulating these models has rendered economic assessments stemming from that model as an unfair statement of the real situation that compact refrigerator/freezer and freezer manufacturers face.

The real situation faced by these manufacturers is that there are only three or four options applicable to their products as energy savings options for the 1998 timeframe. Because of a lack of economies of scale available to the large product manufacturers, the impact on these manufacturers is also more severe. In an effort to balance the economic impact on the manufacturers and the time for consumers to realize the payback for improvements in energy efficiency in these products (which had been assessed at about 2-3% below the 1993 levels for a five-year payback) the compact refrigerator/freezer manufacturers agreed to an energy level approximately 5% below the 1993 standards for all eight compact type refrigerator/freezers and freezers. This proposal was also found acceptable by non-industry participants in this negotiation.

C. Household Freezers

The category of household freezers includes three product classes defined as: chest freezers with manual defrost; vertical freezers with manual defrost; and vertical freezers with automatic defrost. As a group, the freezer product classes have technical and marketing constraints unique to their individual markets. These design constraints are amplified by the fact that the 1993 NAECA energy efficiency standards imposed an additional 14% stricter target on household freezers than refrigerator/freezers. Energy efficiency gains on household freezers out pace any other appliance standard in the U.S. Some parties believe that as a direct partial consequence of the 1993 NAECA standards, three companies terminated production of these products.

a) *LBL Max Tech Assessment of Freezer Products*

LBL completed an assessment of freezer products and life cycle cost curves generated from ERA model outputs. As was stated in the case of the compact refrigerator/freezers and freezers, the ERA model accuracy for the three freezer product classes was +33%, which is still well beyond acceptable modeling simulations.

When considered as a group, the ERA modeling accuracy for the three freezer classes has the highest degree of error than the other ERA assessments.

Much of the rationale for the proposed standard levels for compacts and freezers have resulted from lessons learned in the development of proposed standards for the automatic defrost refrigerator/freezers. For example, when the first LBL "max tech" levels were developed, the maximum technically feasible numbers coming from that assessment were in the mid-40% level below the 1993 NAECA standard. After two years of refinement of the numbers, and clarification of the design aspects presented for those "max tech" levels, the LBL "max tech' level ended up in the low 30% level below the 1993 levels (assuming no change in insulation thickness).

In the case of freezer products, the first shot, LBL "max tech' levels are approximately 40% below the 1993 NAECA standards (see Attachment 13). As opposed to the analyses of the automatic defrost "max tech" levels, the difference between the manufacturers' assessment and LBL's assessment of the three freezer classes is much greater, with the manufacturers' "max tech" level at the mid-20% level below 1993 NAECA standards (see Attachment 14).

The number of energy saving options applicable to household freezers is almost as limited as those for compact refrigerator/freezers. The options applied by LBL in its "max tech" analysis included increased wall and door thicknesses higher EER compressors, improved gaskets and enhanced performance of evaporator and condenser coils. In the automatic defrost vertical freezer product class, adaptive defrost and more efficient motors are applied. These latter options are not used on manual defrost models.

The impact of increased wall thickness on household freezers is a concern as it is for household refrigerator/ freezers. One basic problem is getting the unit through doorways, down hallways and through stairwells. Another problem is that since the freezer market is declining, introduction of designs which are unacceptable to some consumers is even more troublesome. As stated in the argument for the five automatic defrost categories, increased wall and door thicknesses are not options for increased energy performance for household freezers. One freezer manufacturer presented information at the ANPR hearing regarding how it had been forced to reduce its wall thickness by 1/2 inch because of the negative-marketability of the product (the company also stated that energy efficiency gains were less than half that as projected by the ERA model).

Another argument that is carried forward from automatic defrost and compact products is that of available EER for compressors in the 1998 timeframe. With improvements that have been made in foam insulation and gaskets on these products, the size compressor needed to maintain food quality is smaller than it had been in previous years. These smaller compressors do not have EERs as high as the ones stated in the LBL "max tech" analysis. The consensus of the freezer manufacturers and compressor suppliers indicated that an improvement of approximately 7% in EER can be expected between 1994 and the 1998 timeframe.

b) *AHAM Life Cycle Cost Analysis of Freezer Products*

Many technical issues special to freezer product classes have been overlooked in LBL's assessment of "max tech" on these products. For instance, the CFC replacement issue has been especially difficult to resolve on freezer products. HFC-134a, the preferred refrigerant replacement, has an additional 3-4% energy penalty inherent in its performance at temperatures necessary for household freezer products as compared to refrigerator freezers.

Along the same lines, the most common replacement for CFC-ll in the blowing agent for foam insulation is HCFC-141b. Since this chemical is basically in a liquid phase while exposed to temperatures produced in household freezers, the liquid thermal conductivity is especially important in its performance as an energy efficient CFC-11 replacement. As applied to household freezers, however, this particular CFC-11 replacement carries an approximate 5 - 6% energy penalty when applied to household freezers. These two aspects were not taken into account in the LBL "max tech" assessment.

Taking into account the options presented by LBL in its "max tech' assessment, the AHAM LCC analysis agreed, in concept, to all the options stated with the exception of increased foam in walls and doors for the reasons stated in previous sections. The AHAM analysis also agreed with the energy reductions inherent in options improving the efficiency of evaporator coils as well as improved gasket numbers provided in the LBL life cycle cost assessment.

There were differences in the AHAM and LBL analyses in the absolute energy reductions projected for improved EER compressors, the amount of energy that can be saved through improved condenser heat transfer surfaces and the application of adaptive defrost to the vertical freezers with auto defrost.

c) *Proposed Standard for Freezer Products*

Freezers are an optional commodity in a typical U.S. household. They are basically sold in the replacement market, and due to the price sensitivity of this market, there is less opportunity to pass through costs of energy improvements to the consumers. Thus, if regulatory induced costs cannot be passed on, the product line becomes relatively unprofitable.

Because of the simplicity of freezer design, there are fewer applicable design options than there are for automatic defrost refrigerator/freezers. Additionally, since a larger percent of the energy used by freezer products has been reduced through regulation over the last ten years, there is less of an opportunity left to reduce energy consumption further on these products.

In cooperation with industry, DOE, Arthur D. Little, and LBL implemented revisions to the ERA computer simulation to accommodate modeling of household freezers. However, significant errors still exist in the ERA's ability to do this modeling. For example, freezers typically require compressor run times of around

65% while ERA modeling shows run times of around 39%. Freezers require smaller and inherently less efficient compressors to produce a realistic design simulation capable of preserving food in the long term.

With respect to increases in wall thickness, freezer manufacturers showed that the application of this design option to their products does not produce the energy savings estimated from the ERA output and has the marketability problems discussed above.

After carefully reviewing the feasibility and energy efficiencies inherent in options proposed by LBL, while incorporating inputs from refrigerator manufacturers and compressor manufacturers, industry and non-industry participants in this exercise came to the agreement on standards levels for freezer products. Industry agreed to base the standard on most of the design options identified by LBL while non-industry participants agreed to base the standards on the more conservative industry energy savings estimates.

D. Manual/Partial Defrost Products

There are only a few models with a small market niche in this declining product category. The percentage of U.S. sales in these product classes is 1.7% and falling. Data and analysis on elementary engineering and economic issues are difficult to obtain. However, non-industry participants felt that it is important to recommend a relatively stringent US standard on this product class because of the potential impact on similar products produced in or for less-developed countries. They believe that it is likely that other countries then will adopt similar standards. Because of the limited availability of data and small market, a standard proposal related to the top freezer, automatic defrost proposal was developed. Industry and non-industry representatives agreed to an energy standard 10% lower than the top freezer, automatic defrost for manual/partial defrost refrigerator-freezers.

The energy consumption differential between automatic defrost and non-automatic defrost units has been declining over time, and is expected to decline further as adaptive defrost options become incorporated into the automatic defrosting systems. The standards proposal is based on the judgment of all the participants that a 10% energy consumption difference for a given adjusted volume accounts for the relatively irreducible minimum change in energy consumption relating to a manufacturer's decision not to use automatic defrost.

E. Non-HCFC Products

The treatment of HCFC's becomes a significant issue in the design of these standards because of the schedule for implementing energy standards and HCFC standards. Under the Environmental Protection Agency's regulations, HCFC-141b will become unavailable after December 31, 2002. These refrigerator standards will go into effect in 1998, and would remain in effect at least through the December 31, 2002 HCFC-141b termination date (if DOE chooses to set a 2003 standard) or beyond that if DOE does not act. DOE could set standards at a level appropriate considering non-HCFC chemicals in its (optional) next rulemaking, but is prohibited under NAECA from making the standards less stringent.

Therefore, the structure of the energy standards could present a serious problem for the refrigerator industry unless dealt with now with the best available data in this standards selling process.

Current data from Europe, Japan, and the United States support approximately a 10% energy penalty in the shift from HCFC-141b to likely Hydroflourocarbon and Hydrocarbon substitutes. These energy penalties may be the ones that industry faces in its attempt to design, with an appropriate lead time, products that can be sold in 2003.

The parties recognize that in coming years more data will be developed on the energy penalty issue. New technologies may be available to reduce or eliminate the energy penalty, but it is impossible to forecast with certainty their manufacturability by 2003. The parties have proposed to address these issues by developing new product classes for refrigerator/freezers that do not use HCFC-141b in foam insulation. These product classes parallel the conventional product classes set forth in the agreement and Attachment 15. Any non-HCFC, non-ozone depleting foam blowing agent which is approved by EPA under the Clean Air Act's Safe Alternatives Program would qualify a product for these classes. Blends or mixtures which use less than 10% HCFC's qualify.

These non-HCFC classes would permit 10% greater energy use than the comparable HCFC-using classes to provide industry with a known, feasible way of meeting the standards before 2003. The less stringent standard expires 6 years after the effective date of the primary standard (absent another DOE rulemaking). Thereafter, it is anticipated that alternative design options will be available.

The separate tier of standards is triggered or would go into effect under the following circumstances and would apply to units produced:

1) 18 months prior to the total phaseout by EPA of HCFC-141b in January 1, 2003, to wit, July 1, 2001;

2) 18 months prior to any earlier phaseout date or restriction on use of HCFCs in refrigerator/freezer foam set by EPA; or

3) After the granting of a petition by DOE which petition demonstrates that HCFC-141b is in such short supply or economically infeasible to use due to, for example, chemical supplier announcements or other actions affecting supply or use.

After the 1998 effective date of the basic standards and before the effective date of the non-HCFC standard as stated in (1) - (3) above, each manufacturer may annually produce non-HCFC units subject to the alternative standard for up to 5% of its total production or for 10,000 units, whichever is less. This allowance for the non-HCFC standard to apply to a small number of units allows manufacturers the ability for field testing with real consumers under actual commercial conditions, which will be necessary in the case of the advanced technology which will be required to meet the 1998 standards.

The non-HCFC standard would terminate and non-HCFC products would be subject to the basic standard 6 years after the basic standard's effective date unless DOE acts to renew or revise the non-HCFC standard levels in a subsequent rulemaking.

DOE would monitor and require reports on compliance with the field testing exception.

VI. COMPLIANCE WITH NAECA REQUIREMENTS

The agreed-to standards comport fully with the standards criteria in NAECA and have been set to achieve the maximum improvement in energy efficiency which is technologically feasible and economically justified as required by Section 325(O)(2) of NAECA.

By working down from purported max-tech levels and evaluating more stringent levels, such as the 35% level discussed above, the parties settled on the first and most stringent standard level which is clearly technically feasible for all product classes and is economically justified, taking into account the diverse considerations and societal, consumer, and manufacturer interests set forth in Section 325(O)(2)(B).

A. Economic Impact of the Standard on Manufacturers and on Consumers

The agreed upon standards have a favorable impact on consumers, assuming that the markups over direct product costs (labor, materials and factory overhead) remain at or below the industry averages determined by LBL. If costs cannot be fully passed on, then the consumer economics are more favorable than those presented in the LBL analysis but the industry economics are less favorable.

The overall effects of the proposed standards on manufacturers depend heavily on assumptions concerning the manufacturers' ability to pass through both direct production costs and mark-ups sufficient to cover required investments and related costs. Many industry participants believe that it will not be possible to pass on even the full direct product costs as indicated by the fact that the producer price index for refrigerators has significantly lagged behind general increases in consumer process over the past seven years (see Attachment 16). The relationship between these two indices implies that the manufacturers have not been able to pass through their full cost increases in the past, a period with much smaller cost increases than anticipated by the proposed standards.

The refrigerator industry has analyzed the potential effects of the proposed standards on manufacturers using a Government Regulation Impact Model (GRIM), previously submitted to DOE and now in the public domain, developed as an alternative to previous analytical approaches. The GRIM assesses the change in value from the affected portion of the refrigerator industry through a straightforward estimation of the future costs, revenues and investments required to meet any proposed set of standards. All discussions of economic impacts exclude any impacts related to the costs associated with CFC or HCFC phase-outs.

Using the GRIM approach, under an optimistic set of assumptions, the financial impact of the proposed standards could be favorable for the manufacturers of automatic defrost refrigerators. If the manufacturers can pass through their *incremental* direct costs at their current *average* mark-up, then their industry value would remain essentially constant or increase slightly at any of the proposed standard levels (see Attachment 17). If manufacturers can pass through only their direct costs with no mark-up, a situation considered still optimistic by many manufacturers, then the manufacturers are at risk to lose approximately 25% of their total industry value under the proposed standard level (see Attachment 18). At the original "max tech" standard level, the refrigerator manufacturers lose over 100% of the total industry value (or over $800 million) if they can only pass through direct product costs.

The volumes associated with freezer production cause freezer manufacturers to be adversely affected under all reasonable assumptions. Even at the current average mark-up of 30% over direct costs to manufacturers price implicit in the LBL analysis, freezer manufacturers lose industry value even at the proposed standard level (see Attachment 19).

The limited investment requirement by compact refrigerator manufactures reduces greatly the likelihood of adverse impacts from the proposed standard level.

B. The Savings and Operating Costs Compared to Price Increases

At any of the parametric choices of discount rates that have been analyzed by DOE, the savings in operating costs are significantly larger than the increases in price.

C. The Total Projected Amount of Energy Savings

The energy savings provided by this agreed upon standard are very significant: participant estimates are a savings of 20 billion Kwh or .23 quads of primary energy annually by 2010.

D. Any Lessening of the Utility or Performance of the Covered Products

These standards were chosen at a level that provides for no significant lessening of utility or performance.

E. The Impact of any Lessening of Competition

None of the parties to these agreement believe that the standards will lead to a likelihood of reduced competition.

F. The Need for National Energy Conservation

As noted, these standards produce very significant energy savings, both on the absolute level and compare to the results of other DOE rulemakings.

G. Other Factors the Secretary Considers Relevant

These standards allow for the Environmental Protection Agency to make appropriate decisions on the use of HCFCs without interacting negatively with the energy efficiency standards. They allow for a significant reduction in the emission of global greenhouse gases due to the large energy savings.

The standards also have been carefully set to avoid the proscription in Section 325(O)(4) against presenting a standard which would result in the unavailability in the United States in any covered product type or class of performance characteristics (including reliability), features, sizes, capacities, and volume that are substantially the same as those now available. This has been accomplished in several ways. The stringency of the standards, including the slope of the line in the energy equation, have been set to avoid eliminating features, size, and levels of reliability on which consumers have depended for decades. The separate product classes and standards for compacts and freezers will ensure the viability of these products but also provides significant energy savings. Finally, the non-HCFC standards avoids the possibility that some or all manufacturers would have to drastically reduce product offerings because of their inability initially to meet the 1998 standards without HCFC-141b.

The standards agreement also uses and is compatible with the authority in Section 325(q) which authorizes separate standards and classes for products which have a capacity or other performance-related feature which justifies a higher or lower standard, taking into account utility to consumers. The new product classes for compacts, freezers, and partial/manual defrost take this principle into account as do the non-HCFC product classes.

The agreement also represents a recognition of the difficulty manufacturers have in fully passing through costs of regulation. The agreed to standards attempt to cushion manufacturers in the event that they are required to absorb most of the regulatory costs. It is recognized that the five full-line manufacturers also must absorb the significant costs of multiple NAECA standards in addition to manufacturer estimates of $_____________ investment and $__________/unit to meet these standards.

Finally, the fact that stakeholders have made this recommendation should weigh heavily with DOE, indicating an consensus on economic justification.

VII. CONCLUSION

These standards will result in significant electricity savings and will eventually reduce the amount of primary energy use and pollutant emissions by refrigerator/freezers when the current fleet of products has been totally retired. Attachment 20 describes in more detail the energy savings and related environmental benefits.

PROPOSED REVISED 10 CFR §430.32(a)

Existing Section 430.32(a) of the DOE energy conservation standards would be revised as follows.

§430.32 Energy conservation standards and effective dates.

(a) *Refrigerators/freezers.*

(1) *Exclusions.* These standards do not apply to refrigerators and refrigerator-freezers with total refrigerated volume exceeding 39 cubic feet or freezers with total refrigerated volume exceeding 30 cubic feet.

(2) *1990 and 1993 energy conservation standards.* The energy conservation standards for products manufactured on or after January 1, 1990 but before January 1, 1993, and for products manufactured on or after January 1, 1993 but before [effective date of standards] are as follows:

Product Class	Energy standards equations (Kwh/yr) Effective dates	
	January 1, 1990	January 1, 1993
1. Refrigerators and Refrigerator-Freezers with manual defrost	(16.3AV+316)	(13.5AV+299)
2. Refrigerator-Freezer -- partial automatic defrost	(21.8AV+429)	(10.4AV+398)
3. Refrigerator-Freezers -- automatic defrost with: Top-mounted freezer without through-the-door ice service[1]	(23.5AV+471)	(16.0AV+355)
4. Refrigerator-Freezers -- automatic defrost with: Side-mounted freezer without through-the-door ice service	(27.7AV+488)	(11.8AV+501)
5. Refrigerator-Freezers -- automatic defrost with: Bottom-mounted freezer without through-the-door ice service	(27.7AV+488)	(16.5AV+367)
6. Refrigerator-Freezers -- automatic defrost with: Top-mounted freezer with through-the-door ice service	(26.4AV+535)	(17.6AV+391)
7. Refrigerator-Freezers -- automatic defrost with: Side-mounted freezer with through-the-door ice service	(30.9AV+547)	(16.3AV+527)
8. Upright Freezers with: Manual defrost	(10.9AV+422)	(10.3AV+264)
9. Upright Freezers with: Automatic defrost	(16.0AV+623)	(14.9AV+391)
10. Chest Freezers and all other Freezers	(14.8AV+223)	(11.0AV+160)

[1]Including all refrigerators with automatic defrost

AV = Total adjusted volume, expressed in Ft^3, as determined in Appendices A1 and B1 of Subpart B of this Part.

(3) *1998 Energy Conservation Standards.*

(A) The energy conservation standards for products manufactured on or after _____________, 1998 are as follows subject to the provisions in (B) below:

Product Class	HCFC-Containing Product	HCFC-Free Product[1]
i. Automatic Defrost Refrigerator-Freezers (excludes compact refrigerator-freezers)		
1. Top-mounted freezer without through-the-door ice service	9.80AV+276.0	10.78AV+303.6
2. Top-mounted freezer with through-the-door ice service	10.20AV+356.0	11.22AV+391.6
3. Side-mounted freezer without through-the-door ice service	4.91AV+507.5	5.40AV+558.3
4. Side-mounted freezer with through-the-door ice service	10.10AV+406.0	11.11AV+446.6
5. Bottom-mounted freezer without through-the-door ice service	4.60AV+459.0	5.06AV+504.9
ii. Compact Refrigerator/Freezers (AHAM/FTC volume less than 7.75 cubic feet and less than 36 inches in height)		
1. Manual defrost refrigerator-freezer	10.70AV+299.0	11.77AV+328.9
2. Partial automatic defrost refrigerator-freezer	7.00AV+398.0	7.70AV+437.8
3. Top-mounted freezer automatic defrost refrigerator-freezer	12.70AV+355.0	13.97AV+390.5
4. Side-mounted freezer automatic defrost refrigerator-freezer	7.60AV+501.0	8.36AV+551.1
5. Bottom-mounted freezer automatic defrost refrigerator-freezer	13.10AV+367.0	14.41AV+403.7
6. Upright freezer automatic defrost	11.40AV+391.0	12.54AV+430.1
7. Upright freezer manual defrost	9.78AV+250.8	10.76AV+275.9
8. Chest freezer manual defrost	10.45AV+152.0	11.50AV+167.2
iii. Freezers (excludes compact freezers)		
1. Upright automatic defrost	12.43AV+326.1	13.67AV+358.7
2. Upright manual defrost	7.55AV+258.3	8.31AV+284.1
3. Chest freezer manual defrost	9.88AV+143.7	10.87AV+158.1
iv. Manual and partial defrost refrigerator-freezers (excludes compact refrigerator/freezers)		
1. Manual defrost	8.82AV+248.4	9.70AV+273.2
2. Partial automatic defrost	8.82AV+248.4	9.70AV+273.2

[1]See subsection B(i).

(B) *Application of HCFC-free energy conservation standard.*

(i) The HCFC-free energy conservation standard applies to products which contain 10% or less by mass hydrochlorofluorocarbon in the blowing agent portion of the foam insulation.

(ii) The HCFC-free energy conservation standard applies to products described in subsection (i) above which are:

a. manufactured on or after July 1, 2001; or

b. manufactured on or after the date 18 months prior to any rule promulgated by the United States Environmental Protection Agency which revises the December 31, 2002 phaseout date for the use of HCFC-141b in refrigerator/freezers; or

c. manufactured on or after the date on which, after notice and an opportunity for public comment, the Department shall grant a petition by any party for a change in the effective date if the petition demonstrates that continued use of HCFC-141b has or will by a certain date become infeasible due to government, supplier or other actions relative to supply, availability or cost (including federal or state taxes).

(iii) Effective [effective date of standards] and prior to the effective dates set in subsection (ii) above, each manufacturer may annually manufacture product eligible under subsection (i) subject to the non-HCFC energy conservation standards in quantities of up to 5% of its total annual production for that year or 10,000 units, whichever is less.

(iv) Manufacturers manufacturing products under the subsection (iii) shall report this production by serial number to the Department annually as an addendum to the annual report or within 30 days thereafter. Such manufacturers shall maintain sufficient records so as to be able to verify their compliance with the production limits under this subsection.

(v) The HCFC-free energy conservation standard shall terminate [six years from basic standards effective date] unless the Department determines to extend or revise the standard following the procedures and criteria under Section 327 of the Act after public notice and the opportunity for comment.

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Attachment 2
1/5

Life Cycle Costs and Payback Periods of Top-Mount Auto Defrost Refrigerator-Freezer (no TTD features)
ERA Simulation: Draft Version

Level	Option	Manufr Cost (1992$)	Incr. Mfr Cost (1992$)	Retail Price (1992$)	Annual Energy Use (kWh)	Annual Energy Cost (1992$)	Cumulative Payback (years)	Lifecycle Costs (1992$) 4%	6%	10%	Cumulative CCE (cents/kWh) 4%	6%	10%
0	Baseline	$267.98	-	$616.35	664.30	$58	NA	$1,384.14	$1,268.64	$1,105.35	NA	NA	NA
1	0 + 5.41 EER compressor	$270.48	$2.50	$622.10	620.50	$55	1.49	$1,339.27	$1,231.38	$1,078.86	1.00	1.18	1.57
2	1 + 5.80 EER compressor	$274.48	$4.00	$631.30	587.65	$52	2.22	$1,310.50	$1,208.32	$1,063.88	1.49	1.75	2.33
3	2 + decr cond motor power	$278.98	$4.50	$641.65	562.10	$49	2.81	$1,291.32	$1,193.58	$1,055.42	1.88	2.22	2.96
4	3 + decr evap motor power	$285.48	$6.50	$656.60	529.25	$47	3.39	$1,268.30	$1,176.28	$1,046.19	2.27	2.67	3.56
5	4 + 1/2" insulation to doors	$289.20	$3.72	$665.15	511.00	$45	3.62	$1,255.76	$1,166.91	$1,041.31	2.42	2.85	3.81
6	5 + 1/2" insulation to walls	$301.09	$11.89	$692.51	470.85	$41	4.47	$1,236.71	$1,154.85	$1,039.11	3.00	3.53	4.71
7	6 + reduce gasket leak 7%	$304.10	$3.01	$699.43	459.90	$40	4.62	$1,230.98	$1,151.02	$1,037.97	3.09	3.64	4.86
8	7 + 1" insulation to doors	$307.23	$3.13	$706.64	448.95	$40	4.76	$1,225.53	$1,147.47	$1,037.11	3.19	3.76	5.01
9	8 + 1" insulation to walls	$316.27	$9.03	$727.41	423.40	$37	5.24	$1,216.77	$1,143.15	$1,039.08	3.51	4.13	5.51
10	9 + increase evaporator area	$319.62	$3.35	$735.12	416.10	$37	5.44	$1,216.04	$1,143.70	$1,041.42	3.64	4.29	5.72
11	10 + incr condenser area	$322.98	$3.36	$742.86	408.80	$36	5.63	$1,215.34	$1,144.27	$1,043.78	3.77	4.44	5.92
12	11 + adaptive defrost	$330.15	$7.17	$759.34	397.85	$35	6.10	$1,219.17	$1,150.00	$1,052.21	4.09	4.81	6.42
13	12 + fluid control valve	$340.09	$9.94	$782.21	383.25	$34	6.71	$1,225.16	$1,158.52	$1,064.32	4.49	5.29	7.05
14	13 + 6.0 EER linear compressor	$349.59	$9.50	$804.06	372.68	$33	7.31	$1,234.79	$1,170.00	$1,078.39	4.90	5.77	7.69
15	14 + voltage controller	$374.91	$25.32	$862.29	356.65	$31	9.08	$1,274.50	$1,212.49	$1,124.83	6.09	7.16	9.56
16	15 + improved expansion valve	$446.29	$71.38	$1026.47	341.65	$30	14.44	$1,421.35	$1,361.94	$1,277.97	9.68	11.39	15.20
17	4 + reduce gasket leak 7%	$288.49	$3.01	$663.52	518.30	$46	3.67	$1,262.57	$1,172.45	$1,045.05	2.46	2.90	3.86
18	14 + fluid control valve	$298.43	$9.94	$686.39	496.40	$44	4.74	$1,260.12	$1,173.81	$1,051.79	3.18	3.74	4.99
19	15 + increase evaporator area	$301.78	$3.35	$694.09	489.10	$43	5.04	$1,259.39	$1,174.35	$1,054.13	3.38	3.98	5.30
20	16 + increase condenser area	$305.14	$3.36	$701.83	481.80	$42	5.32	$1,258.69	$1,174.92	$1,056.49	3.57	4.20	5.60
21	17 + adaptive defrost	$312.31	$7.17	$718.32	467.20	$41	5.88	$1,258.30	$1,177.07	$1,062.23	3.94	4.64	6.18
22	18 + vacuum panels on W&D	$359.06	$46.75	$825.84	383.25	$34	8.47	$1,268.80	$1,202.16	$1,107.96	5.68	6.68	8.91
23	19 + 6.0 EER linear compressor	$368.56	$9.50	$847.69	372.68	$33	9.01	$1,278.43	$1,213.63	$1,122.03	6.04	7.11	9.48
24	20 + voltage controller	$393.88	$25.32	$905.93	356.65	$31	10.70	$1,318.14	$1,256.13	$1,168.46	7.17	8.44	11.25
25	21 + improved expansion valve	$465.27	$71.38	$1070.11	341.65	$30	15.98	$1,464.98	$1,405.58	$1,321.60	10.71	12.60	16.81

Assumptions:

(1) Energy consumptions for the baseline and for each design option were obtained from an ERA simulation of an actual 18.0 cubic foot refrigerator.

(2) Manufacturer cost of the baseline unit was interpolated from the AHAM manufacturer cost vs kWh curve for this product class.
Using a linear interpolation between the two closest points on the AHAM curve to the ERA baseline consumption of 664.3 kWh, the ERA baseline cost is $616.35.

(3) Electricity cost = 0.088 $/kWh (average cost in 1998 obtained from an interpolation of the 1995 and 2000 prices of electricity forecast in DOE's Annual Energy Outlook 1993, inflated to 1992 dollars). The interpolated value (for 1998) is 0.082 $/kWh (1991$). After adjusting for inflation from 1991 to 1992, it becomes 0.085 $/kWh.
The electricity price was then adjusted by an enduse factor for refrigerators of 1.04.

(4) Installation and maintenance costs are not included in the above calculations.

(5) Lifetime =19 years.

(6) Markup factor= 2.3. This represents the average of markup factors for this product class in the 1989 TSD.

(7) Baseline: Compressor EER = 4.98; Evaporator fan motor power = 9.1W; Condenser fan motor power = 12W.
Insulation thicknesses: freezer door and sides are 1.50" and 2.15", fresh food door and sides are 1.50" and 1.70". Foam resistivity is 0.58 m2-degC/W-cm.
Evaporator and condenser areas: 2.22 sq.m and 0.64 sq.m, respectively.

(8) Level 22 assumes that 50% of total wall and door surface area is covered by 1" thick vacuum panels. The delta cost (compared to foam insulation) assumes a variable cost of $1.20 per board foot, which includes materials, installation labor and shipping. A depreciated investment cost of $10/unit was also assumed. Both costs are derived from Waldron, J.M., "Vacuum Panel and Thick Wall Foam Insulation for Refrigerators: Cost Estimates for Manufacturing and Installation", prepared for US EPA Global Change Division, EPA Project No. X818749-01-0, October 1992.

* This represents the cumulative engineering payback period

Life Cycle Costs and Payback Periods of Side-by-Side Refrigerator-Freezer (with TTD features)
ERA Simulation: Draft Version

Level	Option	Manuf'r Cost (1992$)	Incr. Mfr Cost (1992$)	Retail Price (1992$)	Annual Energy Use (kWh)	Annual Energy Cost (1992$)	Cumulative Payback* (years)	Lifecycle Costs (1992$)			Cumulative CCE (cents/kWh)		
								4%	6%	10%	4%	6%	10%
0	Baseline	$597.31	-	$1373.81	801.04	$70	NA	$2,298.80	$2,159.65	$1,962.93	NA	NA	NA
1	0 + 5.56 EER compressor	$599.81	$2.50	$1379.56	757.63	$67	1.51	$2,254.43	$2,122.82	$1,936.76	1.01	1.19	1.58
2	1 + incr evaporator area	$601.94	$2.13	$1384.47	741.85	$65	2.05	$2,241.11	$2,112.24	$1,930.06	1.37	1.61	2.15
3	2 + decr cond motor power	$606.44	$4.50	$1394.82	714.23	$63	2.75	$2,219.56	$2,095.49	$1,920.09	1.84	2.17	2.89
4	3 + 5.80 EER compressor	$610.44	$4.00	$1404.02	690.55	$61	3.11	$2,201.42	$2,081.47	$1,911.88	2.08	2.45	3.27
5	4 + incr condenser area	$613.18	$2.74	$1410.32	674.77	$59	3.29	$2,189.50	$2,072.28	$1,906.57	2.20	2.59	3.46
6	5 + 1" insulation to doors	$620.32	$7.14	$1426.74	639.25	$56	3.72	$2,164.90	$2,053.85	$1,896.87	2.49	2.93	3.91
7	6 + decr heat load for TTD	$623.19	$2.87	$1433.34	623.47	$55	3.81	$2,153.28	$2,044.97	$1,891.86	2.55	3.00	4.01
8	7 + 1" insulation to walls	$650.02	$26.83	$1495.05	540.60	$48	5.29	$2,119.30	$2,025.39	$1,892.63	3.54	4.17	5.56
9	8 + decr evap motor power	$656.52	$6.50	$1510.00	520.87	$46	5.53	$2,111.46	$2,020.98	$1,893.07	3.70	4.36	5.81
10	9 + decr gasket loss 14%	$663.62	$7.10	$1526.33	505.09	$44	5.86	$2,109.57	$2,021.83	$1,897.80	3.92	4.62	6.16
11	10 + adaptive defrost	$668.08	$4.46	$1536.59	497.20	$44	6.09	$2,110.72	$2,024.35	$1,902.25	4.08	4.80	6.40
12	11 + 6.0 EER linear compressor	$677.58	$9.50	$1558.44	481.41	$42	6.57	$2,114.35	$2,030.72	$1,912.50	4.40	5.18	6.91
13	12 + voltage controller	$690.63	$13.05	$1588.46	460.71	$41	7.17	$2,120.45	$2,040.42	$1,927.28	4.80	5.65	7.54
14	13 + fluid control valve	$700.90	$10.27	$1612.08	449.84	$40	7.72	$2,131.53	$2,053.38	$1,942.91	5.17	6.08	8.11
15	14 + improved expansion valve	$772.07	$71.17	$1775.77	433.63	$38	12.44	$2,276.50	$2,201.17	$2,094.68	8.33	9.80	13.08
16	5 + decr heat load for TTD	$616.05	$2.87	$1416.92	662.93	$58	3.55	$2,182.43	$2,067.27	$1,904.47	2.38	2.80	3.73
17	16 + decr evap motor power	$622.55	$6.50	$1431.87	639.25	$56	4.08	$2,170.04	$2,058.99	$1,902.01	2.73	3.22	4.29
18	17 + adaptive defrost	$627.02	$4.46	$1442.14	627.41	$55	4.48	$2,166.63	$2,057.64	$1,903.56	3.00	3.53	4.70
19	18 + decr gasket loss 14%	$634.12	$7.10	$1458.47	611.63	$54	5.08	$2,164.74	$2,058.49	$1,908.29	3.40	4.01	5.34
20	19 + vacuum panels in W & D	$685.44	$51.32	$1576.51	501.14	$44	7.69	$2,155.19	$2,068.14	$1,945.07	5.15	6.06	8.08
21	20 + voltage controller	$698.49	$13.05	$1606.52	479.60	$42	8.23	$2,160.33	$2,077.01	$1,959.24	5.51	6.49	8.65
22	21 + fluid control valve	$708.76	$10.27	$1630.14	465.63	$41	8.69	$2,167.82	$2,086.93	$1,972.58	5.82	6.85	9.14
23	22 + 6.0 EER linear compressor	$718.26	$9.50	$1651.99	452.78	$40	9.09	$2,174.84	$2,096.18	$1,984.99	6.08	7.16	9.55
24	23 + improved expansion valve	$789.43	$71.17	$1815.69	436.57	$38	13.79	$2,319.81	$2,243.97	$2,136.76	9.23	10.87	14.49

Assumptions:

(1) Energy consumptions for the baseline and for each design option were obtained from an ERA simulation of an actual 22.0 cubic foot refrigerator. A correction factor of 1.081 was applied to the ERA values in order to account for the difference between the simulated and the actual baseline usage.

(2) Manufacturer cost of the baseline unit was interpolated from the AHAM manufacturer cost vs kWh curve for this product class.
Using a linear interpolation between the two closest points on the AHAM curve to the ERA baseline consumption of 801.04 kWh, the ERA baseline cost is $597.31.

(3) Electricity cost = 0.088 $/kWh (average cost in 1998 obtained from an interpolation of the 1995 and 2000 prices of electricity forecast in DOE's Annual Energy Outlook 1993, inflated to 1992 dollars). The interpolated value (for 1998) is 0.082 $/kWh (1991$). After adjusting for inflation from 1991 to 1992, it becomes 0.085 $/kWh.
The electricity price was then adjusted by an enduse factor for refrigerators of 1.04.

(4) Installation and maintenance costs are not included in the above calculations.

(5) Lifetime =19 years.

(6) Markup factor= 2.3. This represents the average of markup factors for this product class in the 1989 TSD.

(7) Baseline: Compressor EER = 5.18; Evaporator fan motor power = 8.0W; Condenser fan motor power = 11.6W.
Insulation thicknesses: freezer door and sides are 1.50" and 2.30", fresh food door and sides are 1.50" and 2.02". Foam resistivity is 0.53 m2-degC/W-cm.
Evaporator and condenser areas: 1.55 sq.m and 0.84 sq.m, respectively.

(8) Vacuum panel option assumes that 50% of total wall and door surface area is covered by 1" thick vacuum panels. The delta cost (compared to foam insulation) assumes a variable co
$1.20 per board foot, which includes materials, installation labor and shipping. A depreciated investment cost of $10/unit was also assumed. Both costs are derived from Waldron, J.M., "Vacuum Panel and Thick Wall Foam Insulation for Refrigerators: Cost Estimates for Manufacturing and Installation", prepared for US EPA Global Change Division, EPA Project No. X818749-01-0, October 1992.

* This represents the cumulative engineering payback period.

Attachment 2
3/5

EE-RM-93-801 COMMENT 49

Life Cycle Costs and Payback Periods of Side-by-Side Refrigerator-Freezer (without TTD features)

ERA Simulation: Draft Version

Level	Option	Manuf'r Cost (1992$)	Incr. Mfr Cost (1992$)	Retail Price (1992$)	Annual Energy Use (kWh)	Annual Energy Cost (1992$)	Cumulative Payback (years)	Lifecycle Costs (1992$) 4%	6%	10%	Cumulative CCE (cents/kWh) 4%	6%	10%
0	Baseline	$396.00	-	$910.80	753.41	$66	NA	$1,780.79	$1,649.91	$1,464.89	NA	NA	NA
1	0 + 5.56 EER compressor	$398.50	$2.50	$916.55	705.68	$62	1.37	$1,731.43	$1,608.84	$1,435.54	0.92	1.08	1.44
2	1 + 5.80 EER compressor	$402.50	$4.00	$925.75	678.41	$60	2.27	$1,709.14	$1,591.29	$1,424.68	1.52	1.79	2.38
3	2 + decr evap motor power	$409.00	$6.50	$940.70	640.91	$56	3.02	$1,680.78	$1,569.45	$1,412.05	2.02	2.38	3.18
4	3 + decr condenser mtr power	$413.50	$4.50	$951.05	620.46	$55	3.44	$1,667.51	$1,559.73	$1,407.36	2.30	2.71	3.62
5	4 + adaptive defrost	$419.22	$5.72	$964.21	600.00	$53	3.96	$1,657.05	$1,552.82	$1,405.47	2.65	3.12	4.16
6	5 + enhanced evap ht surface	$421.65	$2.43	$969.80	593.18	$52	4.19	$1,654.77	$1,551.72	$1,406.05	2.80	3.30	4.40
7	6 + add 1" insulation to walls	$464.95	$43.30	$1069.39	490.91	$43	6.87	$1,636.26	$1,550.98	$1,430.42	4.60	5.41	7.22
8	7 + add 1" insulation to doors	$475.91	$10.96	$1094.59	467.05	$41	7.30	$1,633.91	$1,552.78	$1,438.08	4.89	5.75	7.67
9	8 + reduce defrost heat	$480.02	$4.11	$1104.05	460.23	$40	7.50	$1,635.49	$1,555.54	$1,442.52	5.02	5.91	7.88
10	9 + fluid control valve	$491.22	$11.20	$1129.81	443.18	$39	8.03	$1,641.57	$1,564.58	$1,455.74	5.38	6.33	8.44
11	10 + 6.0 EER linear compressor	$500.72	$9.50	$1151.66	429.55	$38	8.46	$1,647.67	$1,573.05	$1,467.56	5.66	6.67	8.89
12	11 + reduce gasket leak 6.6%	$509.26	$8.54	$1171.30	419.32	$37	8.87	$1,655.50	$1,582.66	$1,479.68	5.94	6.99	9.32
13	12 + enhanced cond ht surface	$513.00	$3.74	$1179.90	415.91	$37	9.07	$1,660.17	$1,587.92	$1,485.78	6.07	7.15	9.53
14	13 + voltage controller	$539.13	$26.13	$1240.00	398.02	$35	10.54	$1,699.61	$1,630.47	$1,532.72	7.05	8.30	11.07
15	14 + improved expansion valve	$614.87	$75.74	$1414.20	384.01	$34	15.50	$1,857.64	$1,790.93	$1,696.62	10.38	12.21	16.29
16	6 + fluid control volve	$432.85	$11.20	$995.56	572.73	$50	5.34	$1,656.91	$1,557.41	$1,416.76	3.57	4.20	5.61
17	16 + reduce defrost heat	$436.96	$4.11	$1005.01	562.50	$49	5.61	$1,654.55	$1,556.83	$1,418.70	3.76	4.42	5.90
18	17 + enhanced cond ht surface	$440.70	$3.74	$1013.61	555.68	$49	5.91	$1,655.28	$1,558.75	$1,422.28	3.96	4.66	6.22
19	18 + 6.0 EER linear compressor	$450.20	$9.50	$1035.46	538.64	$47	6.60	$1,657.44	$1,563.88	$1,431.60	4.42	5.20	6.94
20	19 + vacuum panels in W & D	$501.75	$51.55	$1154.03	453.41	$40	9.22	$1,677.59	$1,598.83	$1,487.48	6.17	7.27	9.69
21	20 + reduce gasket leak 6.6%	$510.29	$8.54	$1173.67	443.18	$39	9.64	$1,685.43	$1,608.44	$1,499.60	6.45	7.59	10.13
22	21 + voltage controller	$536.42	$26.13	$1233.77	424.13	$37	11.16	$1,723.52	$1,649.85	$1,545.69	7.47	8.79	11.73
23	22 + improved expansion valve	$612.16	$75.74	$1407.97	410.12	$36	16.47	$1,881.55	$1,810.30	$1,709.59	11.03	12.98	17.31

Assumptions:

(1) Energy consumptions for the baseline and for each design option were obtained from an ERA simulation of an actual 22.0 cubic foot refrigerator. A correction factor of 0.934 was applied to the ERA values in order to account for the difference between the simulated and the actual baseline usage.

(2) Manufacturer cost of the baseline unit was interpolated from the AHAM manufacturer cost vs kWh curve for this product class.
Using a linear interpolation between the two closest points on the AHAM curve to the (adjusted) ERA baseline consumption of 753.65 kWh, the ERA baseline cost is $396

(3) Electricity cost = 0.088 $/kWh (average cost in 1998 obtained from an interpolation of the 1995 and 2000 prices of electricity forecast in DOE's Annual Energy Outlook 1993, inflated to 1992 dollars). The interpolated value (for 1998) is 0.082 $/kWh (1991$). After adjusting for inflation from 1991 to 1992, it becomes 0.085 $/kWh. The electricity price was then adjusted by an enduse factor for refrigerators of 1.04.

(4) Installation and maintenance costs are not included in the above calculations.

(5) Lifetime =19 years.

(6) Markup factor= 2.3. This represents the average of markup factors for this product class in the 1989 TSD.

(7) Baseline: Compressor EER = 5.18; Evaporator fan motor power = 10W; Condenser fan motor power = 10W.
Insulation thicknesses: freezer and fresh food doors 2.00"; freezer sides (avg of side, back, top and bottom) 2.12"; fresh food sides (avg of side, back, top and bottom) 2.05". Foam resistivity is 0.573 m2-degC/W-cm. Evaporator and condenser UA: 34.3 W/degC and 30.3 W/degC, respectively.

(8) Vacuum panel option assumes that 50% of total wall and door surface area is covered by 1" thick vacuum panel. The delta cost (compared to foam insulation) assumes a variable cost of $1.20 per board foot, which includes materials, installation labor and shipping. A depreciated investment cost of $10/unit was also assumed. Both costs are derived from Waldrom, J.M., "Vacuum Panel and Thick Wall Foam Insulation for Refrigerators: Cost Estimates for Manufacturing and Installation", prepared for US EPA Global Change Division, EPA Project No. X818749-01-0, October 1992.

* This represents the cumulative payback period.

Life Cycle Costs and Payback Periods of Top-Mount Auto Defrost Refrigerator-Freezer (with TTD features)
ERA Simulation: Draft Version

Level	Option	Manufr Cost (1992$)	Incr. Mfr Cost (1992$)	Retail Price (1992$)	Annual Energy Use (kWh)	Annual Energy Cost (1992$)	Cumulative Payback* (years)	Lifecycle Costs (1992$) 4%	Lifecycle Costs (1992$) 6%	Lifecycle Costs (1992$) 10%	Cumulative CCE (cents/kWh) 4%	Cumulative CCE (cents/kWh) 6%	Cumulative CCE (cents/kWh) 10%
0	Baseline	$492.83	-	$1133.51	795.70	$70	NA	$2,053.17	$1,914.82	$1,719.23	NA	NA	NA
1	0 + reduce load for TTD	$496.65	$3.82	$1142.29	759.89	$67	2.79	$2,020.56	$1,888.44	$1,701.65	1.87	2.20	2.93
2	1 + 5.80 EER compressor	$500.65	$4.00	$1151.49	724.09	$64	2.85	$1,988.38	$1,862.48	$1,684.50	1.91	2.25	3.00
3	2 + decr cond motor power	$505.15	$4.50	$1161.84	696.24	$61	3.24	$1,966.54	$1,845.48	$1,674.35	2.17	2.55	3.40
4	3 + decr evap motor power	$511.65	$6.50	$1176.79	660.43	$58	3.64	$1,940.10	$1,825.27	$1,662.94	2.44	2.87	3.82
5	4 + 1" insulation to doors	$523.93	$12.29	$1205.05	624.62	$55	4.75	$1,926.98	$1,818.38	$1,664.84	3.18	3.75	5.00
6	5 + fluid control valve	$533.67	$9.73	$1227.44	600.75	$53	5.48	$1,921.78	$1,817.33	$1,669.66	3.67	4.32	5.76
7	6 + 1" insulation to walls	$581.63	$47.96	$1337.74	509.25	$45	8.10	$1,926.32	$1,837.78	$1,712.60	5.43	6.39	8.52
8	7 + 6.00 EER linear compressor	$591.13	$9.50	$1359.59	493.33	$43	8.50	$1,929.78	$1,844.00	$1,722.74	5.69	6.70	8.94
9	8 + reduce gasket leak	$596.41	$5.28	$1371.73	485.38	$43	8.72	$1,932.73	$1,848.33	$1,729.03	5.84	6.88	9.18
10	9 + decrease anti-sweat heater	$607.80	$11.39	$1397.94	469.46	$41	9.21	$1,940.53	$1,858.91	$1,743.51	6.17	7.26	9.69
11	10 + adaptive defrost	$615.53	$7.73	$1415.71	461.51	$41	9.60	$1,949.12	$1,868.87	$1,755.43	6.43	7.57	10.09
12	11 + increase condenser area	$624.50	$8.97	$1436.35	449.57	$40	9.94	$1,955.96	$1,877.79	$1,767.29	6.66	7.84	10.46
13	12 + voltage controller	$650.02	$25.52	$1495.04	430.23	$38	11.24	$1,992.30	$1,917.50	$1,811.74	7.53	8.87	11.83
14	13 + increase evaporator area	$665.84	$15.82	$1531.42	425.70	$37	12.22	$2,023.44	$1,949.42	$1,844.79	8.19	9.64	12.86
15	14 + improved expansion valve	$744.96	$79.12	$1713.41	409.35	$36	17.06	$2,186.53	$2,115.35	$2,014.73	11.43	13.45	17.94
16	4 + fluid control valve	$521.38	$9.73	$1199.17	632.58	$56	4.57	$1,930.30	$1,820.32	$1,664.83	3.07	3.61	4.81
17	16 + vacuum panels in D & W	$570.60	$49.22	$1312.38	521.18	$46	7.40	$1,914.76	$1,824.14	$1,696.03	4.96	5.84	7.79
18	17 + decrease anti-sweat heater	$581.99	$11.39	$1338.58	501.29	$44	7.92	$1,917.97	$1,830.81	$1,707.59	5.30	6.24	8.33
19	18 + reduce gasket leak	$587.27	$5.28	$1350.73	493.33	$43	8.16	$1,920.92	$1,835.14	$1,713.88	5.47	6.44	8.59
20	19 + 6.0 EER linear compressor	$596.77	$9.50	$1372.58	481.40	$42	8.64	$1,928.97	$1,845.27	$1,726.94	5.79	6.82	9.09
21	20 + adaptive defrost	$604.50	$7.73	$1390.36	469.46	$41	8.95	$1,932.96	$1,851.33	$1,735.93	5.99	7.06	9.41
22	21 + increase condenser area	$613.48	$8.97	$1410.99	460.38	$41	9.40	$1,943.10	$1,863.05	$1,749.89	6.30	7.42	9.89
23	22 + voltage controller	$638.99	$25.52	$1469.69	440.59	$39	10.76	$1,978.91	$1,902.31	$1,794.01	7.21	8.48	11.32
24	23 + improved expansion valve	$718.12	$79.12	$1651.67	424.24	$37	15.85	$2,142.00	$2,068.24	$1,963.96	10.62	12.50	16.68

Assumptions:

(1) Energy consumptions for the baseline and for each design option were obtained from an ERA simulation of an actual 22.0 cubic foot refrigerator. A correction factor of 1.09 was applied to the ERA values in order to account for the difference between the simulated and the actual baseline usage.

(2) Manufacturer cost of the baseline unit was interpolated from the AHAM manufacturer cost vs kWh curve for this product class.
Using a linear interpolation between the two closest points on the AHAM curve to the ERA baseline consumption of 795.7 kWh, the ERA baseline cost is $492.83.

(3) Electricity cost = 0.088 $/kWh (average cost in 1998 obtained from an interpolation of the 1995 and 2000 prices of electricity forecast in DOE's Annual Energy Outlook 1993, inflated to 1992 dollars). The interpolated value (for 1998) is 0.082 $/kWh (1991$). After adjusting for inflation from 1991 to 1992, it becomes 0.085 $/kWh.
The electricity price was then adjusted by an enduse factor for refrigerators of 1.04.

(4) Installation and maintenance costs are not included in the above calculations.

(5) Lifetime =19 years.

(6) Markup factor= 2.3. This represents the average of markup factors for this product class in the 1989 TSD.

(7) Baseline: Compressor EER = 5.46; Evaporator fan motor power = 9.1W; Condenser fan motor power = 12W.
Insulation thicknesses: freezer door and sides are 1.50" and 2.00", fresh food door and sides are 1.50" and 1.57". Foam resistivity is 0.58 m2-degC/W-cm.
Evaporator and condenser areas: 2.51 sq.m and 0.70 sq.m, respectively.

(8) Vacuum panel option assumes that 50% of total wall and door surface area is covered by 1" thick vacuum panels. The delta cost (compared to foam insulation) assumes a variable cost c
$1.20 per board foot, which includes materials, installation labor and shipping. A depreciated investment cost of $10/unit was also assumed. Both costs are derived from Waldron, J.M., "Vacuum Panel and Thick Wall Foam Insulation for Refrigerators: Cost Estimates for Manufacturing and Installation", prepared for US EPA Global Change Division, EPA Project No. X818749-01-0, October 1992.

* This represents the cumulative engineering payback period.

35

Attachment 2
5/5

Life Cycle Costs and Payback Periods of Bottom-Mount Auto Defrost Refrigerator-Freezer
ERA Simulation: Draft Version

Level	Option	Manuf'r Cost (1992$)	Incr. Mfr Cost (1992$)	Retail Price (1992$)	Annual Energy Use (kWh)	Annual Energy Cost (1992$)	Cumulative Payback* (years)	Lifecycle Costs (1992$)			Cumulative CCE — (cents/kWh)		
								4%	6%	10%	4%	6%	10%
0	Baseline	$406.74	-	$935.50	700.71	$62	NA	$1,745.37	$1,623.54	$1,451.30	NA	NA	NA
1	0 + 5.46 EER compressor	$409.24	$2.50	$941.25	668.49	$59	2.03	$1,713.88	$1,597.65	$1,433.34	1.36	1.60	2.13
2	1 + 5.80 EER compressor	$413.24	$4.00	$950.45	636.27	$56	2.64	$1,685.85	$1,575.22	$1,418.82	1.77	2.08	2.77
3	2 + decr evap motor power	$419.74	$6.50	$965.40	600.03	$53	3.37	$1,658.91	$1,554.58	$1,407.09	2.26	2.66	3.55
4	3 + decr cond motor power	$424.24	$4.50	$975.75	579.89	$51	3.79	$1,645.99	$1,545.16	$1,402.62	2.54	2.99	3.98
5	4 + 1/2" insulation to doors	$429.66	$5.42	$988.22	559.76	$49	4.25	$1,635.18	$1,537.85	$1,400.26	2.85	3.35	4.47
6	5 + 1" insulation to walls	$454.38	$24.72	$1045.07	475.19	$42	5.52	$1,594.29	$1,511.67	$1,394.86	3.70	4.35	5.81
7	6 + reduce gasket leak 9.8%	$458.54	$4.16	$1054.63	463.11	$41	5.70	$1,589.89	$1,509.37	$1,395.53	3.82	4.49	5.99
8	7 + 1" insulation to doors	$463.32	$4.79	$1065.64	451.03	$40	5.92	$1,586.93	$1,508.51	$1,397.65	3.97	4.67	6.23
9	8 + increase evap area	$468.23	$4.91	$1076.93	442.97	$39	6.24	$1,588.91	$1,511.89	$1,403.01	4.18	4.92	6.56
10	9 + adaptive defrost	$475.61	$7.38	$1093.91	430.89	$38	6.67	$1,591.93	$1,517.01	$1,411.10	4.47	5.26	7.02
11	10 + 6.0 EER linear compressor	$485.11	$9.50	$1115.76	418.81	$37	7.27	$1,599.82	$1,527.00	$1,424.06	4.87	5.73	7.64
12	11 + fluid control valve	$498.24	$13.12	$1145.95	406.73	$36	8.13	$1,616.04	$1,545.32	$1,445.35	5.45	6.42	8.56
13	12 + incr condenser area	$507.76	$9.52	$1167.84	398.68	$35	8.74	$1,628 63	$1,559.31	$1,461.31	5.86	6.89	9.20
14	13 + voltage controller	$534.48	$26.73	$1229.31	381.53	$34	10.46	$1,670.28	$1,603.94	$1,510.16	7.01	8.25	11.00
15	14 + improved expansion valve	$606.21	$71.73	$1394.28	364.98	$32	15.53	$1,816.13	$1,752.67	$1,662.95	10.40	12.25	16.34
16	4 + reduce gasket leak 9.8%	$428.40	$4.16	$985.31	567.81	$50	4.26	$1,641.59	$1,542.86	$1,403.29	2.85	3.36	4.48
17	16 + adaptive defrost	$435.78	$7.38	$1002.30	547.68	$48	4.96	$1,635.30	$1,540.07	$1,405.45	3.32	3.91	5.22
18	17 + Vacuum panels in W&D	$478.14	$42.36	$1099.73	442.97	$39	7.24	$1,611.71	$1,534.69	$1,425.81	4.85	5.71	7.62
19	18 + increase evap area	$483.05	$4.91	$1111.02	434.92	$38	7.50	$1,613.69	$1,538.07	$1,431.17	5.03	5.92	7.89
20	19 + 6.0 EER linear compressor	$492.55	$9.50	$1132.87	422.84	$37	8.07	$1,621.58	$1,548.06	$1,444.13	5.41	6.37	8.49
21	20 + fluid control valve	$505.68	$13.12	$1163.06	406.73	$36	8.80	$1,633.15	$1,562.43	$1,462.46	5.89	6.94	9.25
22	21 + incr condenser area	$515.19	$9.52	$1184.95	398.68	$35	9.39	$1,645.73	$1,576.41	$1,478.42	6.29	7.40	9.87
23	22 + voltage controller	$541.92	$26.73	$1246.42	381.53	$34	11.07	$1,687.39	$1,621.05	$1,527.27	7.42	8.73	11.65
24	23 + improved expansion valve	$613.65	$71.73	$1411.39	364.98	$32	16.11	$1,833.23	$1,769.77	$1,680.06	10.79	12.70	16.95

Assumptions:

(1) Energy consumptions for the baseline and for each design option were obtained from an ERA simulation of an actual 20.0 cubic foot refrigerator. A correction factor of 1.1033 was applied to the ERA values in order to account for the difference between the simulated and the actual baseline usage.
The baseline was obtained by decreasing the wall and door thicknesses of the unit provided by Manufacturer "F" by 1/2" each in order to bring the energy use closer to the DOE standard:

(2) Manufacturer cost of the baseline unit was interpolated from the AHAM manufacturer cost vs kWh curve for this product class.
Using a linear interpolation between the two closest points on the AHAM curve to the ERA baseline consumption of 700.71 kWh, the ERA baseline cost is $406.74.

(3) Electricity cost = 0.088 $/kWh (average cost in 1998 obtained from an interpolation of the 1995 and 2000 prices of electricity forecast in DOE's Annual Energy Outlook 1993, inflated to 1992 dollars). The interpolated value (for 1998) is 0.082 $/kWh (1991$). After adjusting for inflation from 1991 to 1992, it becomes 0.085 $/kWh.
The electricity price was then adjusted by an enduse factor for refrigerators of 1.04.

(4) Installation and maintenance costs are not included in the above calculations.

(5) Lifetime =19 years.

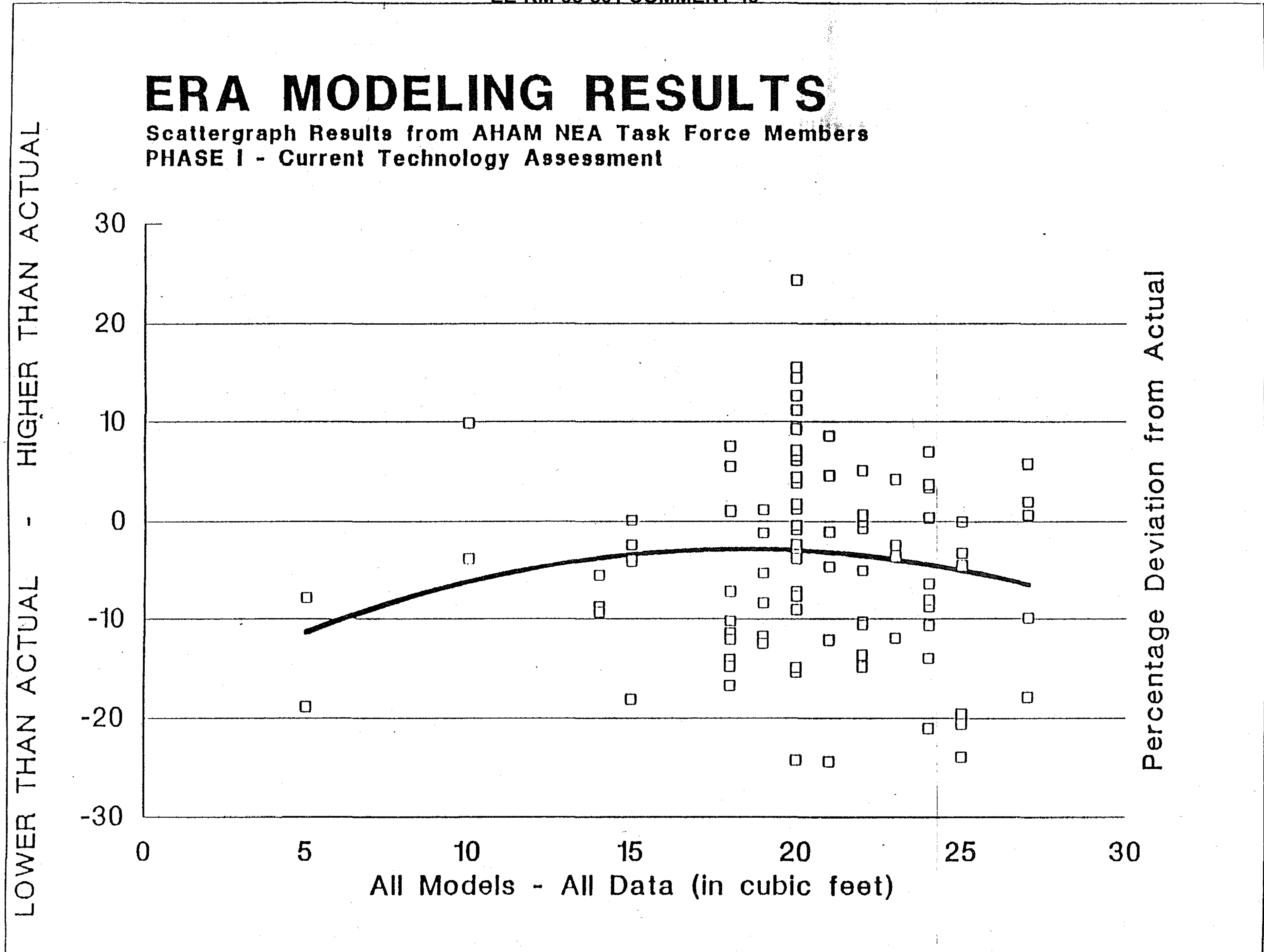
(6) Markup factor= 2.3. This represents the average of markup factors for this product class in the 1989 TSD.

(7) Baseline: Compressor EER = 5.14; Evaporator fan motor power = 10.5W; Condenser fan motor power = 10W.
Insulation thicknesses: freezer door and sides are 1.69" and 2.50", fresh food door and sides are 1.69" and 1.59". Foam resistivity is 0.55 m2-degC/W-cm in the sides and 0.53 m2-degC/W-cm in the doors.
Evaporator and condenser areas: 3.56 sq.m and 7.94 sq.m, respectively.

(8) Vacuum panel option assumes that 50% of total wall and door surface area is covered by 1" thick vacuum panels. The delta cost (compared to foam insulation) assumes a variable cost of $1.20 per board foot, which includes materials, installation labor and shipping. A depreciated investment cost of $10/unit was also assumed. Both costs are derived from Waldron, J.M., "Vacuum Panel and Thick Wall Foam Insulation for Refrigerators: Cost Estimates for Manufacturing and Installation", prepared for US EPA Global Change Division, EPA Project No. X818749-01-0, October 1992.

* This represents the cumulative engineering payback period.

Attachment 3



Attachment 4
1/18

Accuracy Analysis of the ADL/ERA Model

Prepared for

Association of Home Appliance Manufacturers
NAECA Task Force

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Background

This report was prepared at the request of the NAECA Task Force of the Association of Home Appliance Manufacturers. It is part of a larger effort by the Task Force to quantify the uncertainty associated with estimates of energy savings obtained using the USEPA's refrigerator analysis program (the ERA program). The program was developed by Arthur D. Little Inc. (1992) and will be used by DoE in the 1998 standard-setting process.[1]

The ERA model is an improved version of the model used by DoE in setting the 1993 standards (ADL, 1982). The improvements dealt mainly with the user interface, and the addition of multizone heat exchangers, alternative refrigerants and new compressor models that do not require calorimeter testing with a particular new refrigerant. AHAM task force members, engineers from the major refrigerator/freezer manufacturing companies, used the ERA model to perform the analyses reported here and provided all the necessary input data.

Purpose and scope

For many of the conservation measures likely to be considered by USDoE when setting the 1998 NAECA energy efficiency standards, adequate test data exist and enable manufacturers to guage the credibility and certainty of energy savings predicted by the ERA model. However there are little or no test data available for the measures listed in Table 1. Therefore DoE and other parties to the NAECA rulemaking may be forced to rely almost exclusively on model-based estimates for these conservation measures. This report presents results of an effort to estimate the uncertainties associated with such model-based estimates.

[1] To assist in this evaluation of the ERA model, ADL provided the AHAM Task Force members with disks containing the ERA program and input files used in the USEPA's Draft Multiple Pathways Report in 1992. Shortly after completion of the analysis reported here, AHAM was informed that USEPA was about to release an updated version (1.0) of the ERA model and user's manual. That version was said to contain a number of improvements over the version reviewed herein, but none that would substantially alter the conclusions and recommendations contained in this report.

• Gasket improvements
• Vacuum insulation
• Alternative refrigerants & mixtures
• Lorenz cycle
• Variable speed compressors
• Reduce cycling losses
• Dual evaporator

Table 1. Conservation measures considered

A simple example will demonstrate the importance of quantifying the uncertainties associated with an ERA estimate of energy use. For a base case refrigerator, suppose that the model predicts an energy use of 2.00 kWh/day, and 1.80 kWh/day after the design has been modified to incorporate a conservation measure. In the absence of an uncertainty analysis, one would conclude that savings of 0.20 kWh/day could be achieved. However if the uncertainties on both the badeline and altered results are about 5%, say ±0.10 kWh/day, then the estimated energy savings ΔE could be said to lie in the interval: $0 < \Delta E < 0.4$ kWh/day.[2] The error, ± 0.2 kWh/day, is as large as the original estimate (nominal value) for energy savings that was obtained initially by running the model twice. This is a common problem encountered when subtracting two large numbers, especially when the large numbers are only known to three significant figures, as the ERA model is programmed to display its calculated energy use prediction.

This example illustrates the importance of knowing what the uncertainties are, and how they propogate through the model to influence the results. In nonlinear models input uncertainties and errors may be magnified or shrunk, and they may add in a worst-case fashion or cancel one another (Porter and Bullard, 1992).

The paucity of data on the effectiveness of these conservation options has another important implication. It means that some of the equations used to quantify the energy savings may not yet have been subjected to rigorous validation. Therefore in the following analysis three sources of inaccuracy are considered: 1) uncertainties in the parameters specified as inputs to the ERA model; 2) assumptions embedded in ERA's equations; and 3) terms inadvertently omitted from ERA equations. In most cases errors of types #2 and #3 cannot be quantified in this report due to lack of data, but in some cases it was possible to use limited proprietary data from manufacturers to test for the presence of these types of errors.

Overview of methodology

The general approach is straightforward. It is based on the concept that one can identify independent sources of error. Each of these errors propagates through the system (or in our case the set of

[2] The maximum savings could be 2.1 - 1.7 = 0.4 kWh/d and the minimum could be 1.9 - 1.9 = 0 kWh/d.

nonlinear simultaneous equations describing the system) and affects the values of the output variables (in our case we are primarily interested in the effect on only one of them, the energy use E). Since these input errors are independent, their effects can be added: the resultant uncertainty is given by a root-sum-squared equation. Given that

$$E = E(x_1, x_2, \dots x_n)$$

and given the uncertainties $\Delta x_1, \Delta x_2, \dots \Delta x_n$ on each of the independent variables, then the uncertainty in E is given by

$$\Delta E = \{[(\partial E/\partial x_1)\Delta x_1]^2 + [(\partial E/\partial x_2)\Delta x_2]^2 + \dots + [(\partial E/\partial x_n)\Delta x_n]^2]\}^{0.5} \qquad (1)$$

In our case we will define Δx such that we are 95% certain that the true value of x lies within the range $x \pm \Delta x$. Then there is a 95% chance that the actual value of E lies within the range $E \pm \Delta E$. If these uncertainties are normally distributed, the upper and lower limits of x may be considered to be specified at the $\pm 2\sigma$ level.

Note from eq. (1) that the result will be dominated by the largest term inside the [ ] because each is squared. It is therefore possible to limit the analysis to the relatively few large terms, and to ignore the more numerous smaller ones. For example the sum of a 10% error due to one of the inputs plus twenty-one 1% errors will result in only an 11% error on E, the annual energy use.[3] Therefore attention was focused on obtaining accurate estimates of only the largest terms contributing to the total uncertainty.

This kind of analysis is called "single-sample uncertainty analysis" because it defines the range within which the result will lie, even if only a few measurements and variables x_i are involved.[4] The result is therefore very conservative, mainly because not enough data exist for mean values to be known.[5] To define a more realistic range on E would require two things: 1) more extensive data on the energy-conserving designs to permit accurate estimation of the mean values of the parameters describing the improved systems; and 2) a Monte Carlo analysis in which hundreds of hypothetical refrigerators were "constructed" from inputs x_i randomly selected from "bins" containing values in the range $x_i \pm \Delta x_i$. The resulting E_i would lie within a much narrower range than $E \pm \Delta E$ if the errors combined so as to cancel one another. We will take the simpler root-sum-squared approach outlined above. It is generally not advisable to start with a Monte Carlo analysis; it is better to learn first which sources of uncertainty are the most important to model.

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[3] The combined error is given by $[10^2 + 11(1^2)]$

[4] For a detailed explanation of this kind of uncertainty analysis see R. J. Moffat, "Describing uncertainties in experimental results", *Experimental Thermal and Fluid Science* 1988: 1:3-17; or S. J. Kline and F. A. McClintock, "Describing uncertainties in single-sample experiments", *Mech. Eng.*, 3-8, January 1953.

[5] It is not as conservative as a worst case analysis in which all inputs x were set at their extreme values in such a manner that all the errors added. The approach used here accounts for the fact that the sources of uncertainty are independent and are therefore unlikely to add in a worst-case manner.

Application to conservation options

To estimate the uncertainty relating to a particular conservation option, we will ignore for the moment the effect of uncertainties on parameters describing parts of the system that are not changed. For example in the case of adding vacuum panels, only part of the foam is replaced. What is relevant to the accuracy of the model is how well the mean values of the replaced foam and the vacuum panel are known. In the case of R11 foam it might be argued that the thermal conductivity k and the resulting R-value for the wall panel known, due to years of experience and extensive testing of this material. In fact it might be reasonable to assume that the *mean value* is known with perfect accuracy. In that case the only uncertainty introduced by the replacement of foam by vacuum panels would be those associated with the thermal conductivity and thickness of the vacuum panels. Because of the lack of extensive testing and data, such uncertainties are analyzed using the single-sample methods described above.

For other conservation measures, however, there exists substantial uncertainty about the mean values of parameters describing the base case design. Examples include heat exchanger conductances and gasket heat leaks. Despite years of testing and analysis, uncertainties remain due to measurement limitations and the lack of standardized tests. For conservation measures involving these components, uncertainties in the base case estimates are considered explicitly.

In the following subsections the conservation measures are analyzed individually for all three types of error sources: 1) uncertainties in the inputs specified by the user of the ERA model; 2) assumptions embedded in ERA's equations; and 3) terms inadvertently omitted from ERA equations. Type 1 errors are quantified where possible, while the others are discussed in qualitative terms.

Gasket heat leak

The ERA model allows the user to input a gasket heat leak, and it includes a calculational assist feature to calculate that figure from information about the gasket material and geometry. Typical values for the main parameters affecting this calculation are shown below:

Internal exposed width	0.95 ± 0.25 cm
Internal heat transfer coefficient	5.0 ± 1.0 W/m^2°C
Cabinet skin thickness	1.30 ± 0.38 mm
Gasket thickness	0.43 ± 0.13 mm

An ERA simulation was conducted using these values. It showed that the ERA-calculated heat transfer is dominated by the first two terms, and that the aggregate effect of all the uncertainties is about 2.6%, or about 15 kWh/y for a baseline 18 cubic foot model. These results are highly suspect for several reasons. First, conduction through the cabinet flange is modeled in ERA, while conduction through the door flange is apparently ignored. Second, the ERA model does not allow the user to specify the most important factors affecting gasket region heat leak, namely the relative positions of the gasket and flanges. It appears that ERA attempts to account for most of the heat gain through the door edge region as direct heat transfer through the gasket, while in fact conduction through the sheet metal flanges is the critical factor.

Flynn, et. al. (1992) confirmed this fact by demonstrating through finite element modeling that it is the interaction between the flanges and the gasket, rather than the dimensions of either, that determine the heat transfer through the door edge region. ERA only allows the user to change the dimensions of the flanges and gasket, not to change their interactions in the ways demonstrated by Flynn, et. al.. The work of Boughton et. al. (1992) and that of Flynn, et. al. (1992) indicate that simply changing gasket dimensions could yield savings greater or less than the magnitude predicted by ERA, depending on the extent to which the flanges protrude past the edge of the gasket into the refrigerator. If as the ERA users manual states, the gasket heat leak calculations do in fact yield accurate results for the base case, there is no guarantee that calculations of energy savings due to redesign will be correct. For example if the steel flange protrudes past the gasket, most of the heat will travel through the flange and warm the internal exposed surface of the gasket. On the other hand if the flange terminates halfway through the gasket the internal exposed surface of the gasket will be much colder, and the heat transfer through the gasket itself will be much larger. The energy savings due to changing the gasket material, its internal exposed width, or the flange thickness will differ greatly among these cases, which cannot be distinguished by ERA.

ERA users must also be aware of other interactions besides those addressed by Flynn, et. al. (1992), who concentrated mainly on the extent to which the gasket overlapped the ends of the metal flanges. Heat leaks through the gasket and the sheet metal flanges may be exacerbated by the presence of mullion heaters and anti-sweat heaters, which are modeled independently in ERA. Also, three-dimensional heat transfer occurs wherever the sheet metal door or cabinet skin protrudes into the cold interior to provide for the magnetic door seal. The wall heat transfer equations in ERA do not account for this 3-D effect, which will lower the temperature of the outer skin and thereby reduce 2-D heat transfer through the wall.

Since ERA apparently ignores conduction through the door flange, a crude approach to modeling this effect using ERA would be to add this value to the user-specified gasket heat leak, and use that instead of the calculational assist.

Another pathway for heat gain through the gasket region, air infiltration, is also ignored in the ERA model. Manufacturers report that this factor is highly variable, and contributes significantly to the total gasket-region heat gain of 20 to 25% of total cabinet heat load. (Note that this percentage was applicable to 1990-vintage cabinets; it could be proportionately larger for 1993 or vacuum-panel cabinets in the future.) A crude way to account for air infiltration using the ERA model would be to specify some "equivalent number of door-openings." Note also that heat leaks occuring on the DoE test involve relatively dry 90°F air, while in actual refrigerators the infiltrating air will be somewhat cooler but wetter thereby increasing defrost energy.

The reports by Flynn, et. al. and Boughton et. al. were based on laboratory studies aimed at isolating gasket-region heat leaks from the remainder of the cabinet loads. The more typical situation, however, is that a manufacturer will test an entire cabinet and compare the result to the ERA prediction. One manufacturer did this, using the ERA calculational assist to quantify the gasket-region contribution, and compared the totals. The results are shown in the Table below.

Model Type And Size	Cabinet Heat Leak (w/o Fans, Defrost, Door Openings, Etc.) (Watts)	Energy Usage of Cabinet and System (Fully modeled) (kWh/day)		
	(Actual - Predicted)/Actual (%)	Predicted	Actual	% Error
Top Mount 15 Cu. Ft.	21.2	1.71	2.37	27.8
Side-By-Side 20 Cu. Ft.	24.4	2.37	3.11	23.8
Side-By-Side 24 Cu. Ft.	25.3	2.57	3.39	24.2

Vacuum panel insulation

A quantitative analysis using ERA was conducted by one manufacturer for a particular configuration of vacuum panels in a refrigerator having a baseline energy consumption of 686 kWh/y. Energy savings of 106 kWh/y due to the vacuum panels were estimated by the model. Uncertainties about the enclosure thickness (± 18%) and panel resistivity (± 10%) combined to place the result in the following range: 97 < 106 < 115 kWh/y.

This preliminary result was based on the assumption that the mean value of the foam resistivity (R = 0.58 m^2-°C/W-cm) was known with perfect certainty. While this may arguably be true for R11 foam (individual panels may vary due to manufacturing tolerances etc. but the mean is by now well known) the same may not be true for non-CFC foams. If DoE were to simply use a laboratory test result for a new (R141b) foam to calculate energy savings, the input value of k might be in error by as much as ± 10% because of yet-unknown differences between laboratory specimens and actual blowing around vacuum panels; the difference between laboratory test temperatures (75°F) and actual temperatures of foam in a refrigerator operating in a 90°F test chamber; etc. Factoring this uncertainty into both the base-case and vacuum-panel refrigerator changes the result dramatically to 75 < 106 < 137 kWh/y.

Comparison of these results illustrates another potential source of error: careless extrapolation of limited test data by the analyst running the ERA model. The ± 10% uncertainty on foam conductivity may be reduced considerably over time as more is learned about the process of foaming around vacuum panels, the temperature dependence of k-values, and other characteristics of foams made with non-CFC blowing agents. Until these uncertainties are resolved, the uncertainty analysis must address all three parameters (panel resistivity; enclosure thickness; foam resistivity); the latter cannot be neglected as in the case of R11-blown foam.

Other uncertainties unique to vacuum panels cannot be analyzed quantitatively using the ERA model. For example the "bimetallic effect" of a vacuum panel in a door may cause some warping of the door and lead to increased air infiltration. Data from one manufacturer indicates the existence of such an effect, but much more data would be required to quantify it. The ERA model would then need to be modified to account for the resulting infiltration through the gasket region.

Heat leaks through the gasket region are also underestimated by the ERA model, as described in the previous section. It was suggested that an estimate of heat conduction through the sheet metal door flange could be included in the user-specified gasket heat leak. However if this were done, *and* the model were then used to estimate energy savings due to vacuum panels, the energy savings would be overestimated because heat flux through the flange would increase. The vacuum panels (or any other kind of increased wall insulation) would increase the temperature of the sheet metal to a level approaching approaching that of the test chamber, while the temperature inside the refrigerator remained constant. The higher temperature differential would cause more heat to be transferred through the flange than in the base case without vacuum panels. Preliminary tests conducted by this manufacturer for four configurations showed that ERA-predicted energy savings exceeded measured energy savings by 18%, 20%, 47% and 60%, respectively. Part of these differences may be attributable to 3-D effects such as flange heat leaks not modelled by ERA, while the rest is probably attributable to the factors quantified above.

This kind of three-dimensional effect, ignored in ERA, can be analyzed using a much more complex finite-difference model. One manufacturer did so for two configurations and found that ERA under-predicted energy savings by 73% and 74%, respectively. A laboratory test was then conducted for one of the refrigerators, and the actual savings were actually 60% greater than predicted by ERA. The lesson here is that more accurate 3-D models can guide the placement of vacuum panels to maximize 3-D energy savings and minimize 3-D energy losses. The fact remains, however, that the crude assumptions in the ERA model may combine with uncertain input parameters to greatly overestimate or underestimate the energy savings achievable with vacuum panels.

Alternative refrigerants and mixtures

The thermodynamic property data used by ERA are the CSD equations of state obtained from NIST REFPROP 3 program (Morrison and McLinden, 1986). They could be a significant source of uncertainty because they are relatively crude equations requiring only minimal experimental data. They were used in ERA because unlike other more accurate equations of state they can be easily combined to approximate the properties of mixtures. However ERA uses them for pure fluids as well.

Preliminary comparisons have shown that the latent enthalpy of R-134a predicted by the CSD equations of state to contain to vary by about 1 to 4% from the Martin-Hou and Benedict-Webb-Rubin (BWR) equations.[6] Similar discrepancies are found in the near-superheat regime. The magnitude of these errors were confirmed by NIST (Gallagher, 1993) who estimated that: the CSD equations estimate the vapor pressure curve quite accurately (±0.2%); for latent enthalpy errors in the 3-5% range are entirely possible; and that the NIST BWR equations are the most accurate for R-134a. Therefore the only way to remove this source of uncertainty from the model is to replace the CSD equations of state by the NIST BWR equations when simulating systems using R-134a.

Similar errors may exist for other pure refrigerants. Note that this error tends not to be randomly distributed about a mean, but introduces a clear bias in the positive direction. The direction may be different for other refrigerants.

The inaccuracy of the latent enthalpy data for R-134a will propagate through the ERA model to affect its estimates of annual energy use almost proportionately. That is, a 1% error in latent enthalpy at the evaporator temperature will cause ERA to alter the run time and hence the energy use by an equal amount. There will be some second-order effects in the compressor model and due to the way in which ERA handles cycling losses, but these are probably insignificant compared to the 1% error in annual energy use that could result from inaccuracies in the property data at the evaporating temperature. Similarly, a 4% error in latent enthalpy at the condensing temperature will alter the ΔT at the condenser (and therefore the EER) as the model holds the user-specified values of UA and subcooling constant.

Additional errors will be introduced when ERA is used to simulate mixtures of refrigerants. The interaction coefficients in the NIST property routines are intended to account for such factors as zeotropic behavior. No quantitative analysis of the associated uncertainty could be conducted, however, because of a lack of data on the uncertainty of the interaction coefficients themselves, and on latent enthalpy and other key properties of the refrigerants most likely to be used in mixtures for refrigerator-freezers.

[6] For example at a condensing temperature of 130°F the CSD equations' value for latent enthalpy is about 4% larger than the value given by the Martin-Hou equations of state. At an evaporating temperature of -20°F the error is about 1%.

Attachment 4
9/18

The ERA model's ability to deal with alternative refrigerants and mixtures is further limited by inadequacies in the compressor models (see below). Reliable results can probably only be based on maps of compressor calorimeter data obtained with the refrigerant or blend in question.

Lorenz cycle

Efforts to conduct an uncertainty analysis of the Lorenz cycle feature of the ERA model began with a simple attempt to compare model results to the results of a test conducted by one manufacturer using a mixture of 65% R22 and 35% R141b. The energy savings estimated by ERA were discarded as not credible because the program predicted temperature glides of only 0.2°C and 1.2°C for the condenser and evaporator, respectively. These differed greatly from the glides of 30°C and 37°C predicted by the REFPROP program from NIST. The Task Force member conducting the analysis reported that the results did not change when the interaction coefficient from REFPROP was entered into the ERA program.

A second apparent error in the ERA program was discovered when examining the effect of heat exchanger geometry. To fully exploit the thermodynamic benefits of the Lorenz cycle a counterflow heat exchanger is required. However due to the difficulty of achieving this goal in practice given the packaging constraints in refrigerator/freezers, an attempt was made to compare ERA's estimates of the performance of crossflow and counterflow designs. The starting point was input file B6_05LRZ.ERA which was used by USEPA (1992) in its "multiple pathways" report on super efficient refrigerators. Strangely, that input file specified crossflow rather than counterflow heat exchangers. When the inputs were changed to counterflow and the program re-run, the resulting estimate of energy use was identical.

Both of these findings indicate the presence of at least one serious error in the ERA program, or at a minimum a serious problem with the user interface that would allow experienced designer to make data input errors that would lead to such erroneous results. The error(s) in the program might conceivably have a common cause, because in the absence of a temperature glide one would expect to see no significant difference between the performance of crossflow and counterflow geometries at the specified design condition (minimal superheat or subcooling).

Because of these problems with ERA's ability to properly model the behavior of mixtures and the effect of counterflow heat exchanger geometries, the uncertainty analysis was terminated at this point.

There is also cause for concern about approximations made by ADL in calculating the effectiveness of heat exchangers in the Lorenz cycle. The equation for effectiveness depends critically on the ratio of heat capacities (product of mass flow and specific heat) of the refrigerant and air streams. ERA calculates this ratio using an *average* specific heat which it determines from refrigerant inlet and outlet enthalpies and temperatures. However the actual value of this specific heat can vary by as much as a factor of seven or more through the heat exchanger (Conklin and Vineyard, 1992), depending on the refrigerant mixture selected. This in turn affects the ratio of heat capacities by the same factor. If the effectiveness of the heat exchanger is small, this error in the ratio of heat capacities will have only a small effect on the comparison between crossflow and counterflow designs. On the other hand if the effectiveness is large, substantial errors may be introduced into comparisons of crossflow and counterflow geometries.

Other simplifying assumptions made in the ERA model will also introduce uncertainties into estimates of energy savings achievable through the Lorenz cycle. They were identified in the User's Manual (ADL, 1992); they are not addressed in detail here because the accuracy analysis was terminated at an earlier stage for the reasons stated above. For example, the heat transfer correlations used to calculate conductance were originally determined for pure fluids, and should not be used for refrigerant mixtures because they are known to behave very differently.

Finally the ERA program skirts the major design problem associated with Lorenz cycle refrigerators — cabinet temperature control. If evaporator areas are optimized for a particular design operating condition, compartment temperatures could float to (cold) levels that would waste energy or (warm) levels that would spoil food, depending on climate and door-opening conditions experienced under actual operating conditions. Both of ERA's "control" options, involving the discharge of freezer air to the fresh food compartment or bypassing part of the evaporator area, would waste energy relative to the nominal condition at which the evaporators were designed to operate.

Variable-speed compressors

Recent developments in electric motor technology have created the opportunity for using variable-speed drives on compressors as well as the fans that move air over heat exchangers. A variable-speed compressor could eliminate cycling losses, track loads induced by usage patterns and climate, and increase operating efficiency by reducing condensing temperatures and increasing evaporating temperatures. That is one reason for developing the capability for modeling such equipment.

A second reason for developing better ways of modeling compressors is to be able to predict a given compressor's performance with alternative refrigerants. Since compressor map equations are both compressor- and refrigerant-specific, the ERA model includes two other compressor models. Both require the user to input such physical parameters as displacement volume and speed (rpm), which are needed to predict how a particular compressor will perform with a variable-speed drive, or with an alternative refrigerant having a different specific volume and other thermodynamic properties.

Compressor rating point model The rating point model requires the user to input the EER only at a single rating point, and then proceeds to calculate its performance throughout the 'map' range of evaporating and condensing pressures. However these calculations are based on empirical correlations obtained for compressors using R-12. Since the slopes of the compressor performance maps vary significantly for different refrigerants, one would not expect the ERA rating point model to yield accurate results for anything but R-12. One Task Force member simulated two refrigerators using an actual compressor map obtained with R-134a, and repeated the simulations using the ERA rating point model. The energy requirements predicted by the rating point model were 5% and 8% lower than those predicted using the map-based model. It is difficult to draw a generalized conclusion from this limited information. However it is clear that this data, shown in the table below, provides no evidence that the rating point model is as accurate as the more widely accepted map-based models.

	Normalized Energy	Run Time %	Operating Capacity Btu/h	Operating EER Btu/h/W
Top Mount 19 Cu. Ft.				
Compressor map method	1.0	48.0	667	4.75
Rating point model	0.948	51.8	613	5.12
Side-by-Side 22 Cu. Ft.				
Compressor map method	1.0	49.4	826	5.05
Rating point model	0.920	51.0	776	5.43

Compressor physical model ADL's documentation of the compressor physical model was reviewed by Task Force members who design compressors, in anticipation of conducting a detailed accuracy analysis. The ERA model requires the user to specify, among other parameters, an "isentropic compression efficiency" which appears to refer to a parameter known in the industry as "piston work isentropic compression efficiency". This parameter equals the standard "isentropic compressor efficiency" defined in thermodynamics texts, modified for motor losses and friction and windage losses. In addition, the ERA model demands that the user specify input parameters that differ from the standard parameters familiar to designers (e.g. motor-pump efficiency; can loss as percent of power; discharge line loss). Because of the industry's unfamiliarity with these parameters and resultant lack of reliable data, Task Force members concluded that the uncertainty on these input parameters would introduce an intolerable amount of error into the ERA simulation. They suggested that a more accurate approach would be to simply specify the standard "isentropic compressor efficiency" as the input parameter that would be used to calculate the power required.

In contrast to power input, the compressor mass flow rate is computed in ERA's "compressor physical model" in a much more straightforward manner. It follows directly from the user-specified values of displacement, clearance volume, rpm and a relation between volumetric efficiency and pressure ratio that is hard-wired into the program. This type of model places most of the burden on the user to supply the correct inputs, but there are still uncertainties about the accuracy of the volumetric efficiency formula and its applicability to alternative refrigerants. The greatest uncertainty, however, is tied into the extremely complex heat transfer phenomena that occur inside the compressor shell between the exit of the suction line and the suction port of the cylinder itself (in a reciprocating compressor). The ERA model does not claim to be able to simulate rotary vane compressors.

Neither ADL nor EPA have published data validating either the physical model or the rating point model, for example by attempting to reproduce compressor maps for the standard matrix of test conditions. This is the minimal amount of validation required before such models could be substituted for maps for constant-speed operation with alternative refrigerants. If such data were available, it might be reasonable to expect the physical and rating point models to more accurately simulate compressor operation at conditions significantly different from the 90°F ambient air and refrigerant inlet temperatures on which the performance maps are based. However, before using such models to predict the effects of variable-speed operation, there should be some validation done under those conditions because the compressor can heat transfer (and the closely-related heat loss from the motor windings and friction losses) may differ greatly from the full-load case.

Because of these deficiencies in the rating point and physical models, the Task Force members have no confidence in them. In the absence of validating data it is recommended that neither model be used to predict the power, mass flow or can heat loss as a function of refrigerant properties or compressor speed. In the meantime simulations involving alternative refrigerants or variable-speed compressors ought to be based on the map-based model using data obtained for the specific refrigerant (or blend) at the particular rotational speed being simulated.

Reducing cycling losses

Cycling losses are estimated by the ERA model as a part of each simulation. However the model is set up to analyze only one conservation measure aimed at reducing such losses: a check valve that would prevent the refrigerant from migrating to the evaporator during the off-cycle.[7] Other approaches to dealing with cycling losses (e.g. reducing the refrigerant charge) cannot be simulated by ERA.

The ERA model simulates steady-state operation at standard design conditions. It then deals with cycling losses by applying a correction factor taken by ADL (1992) from a report by Janssen et. al. (1990). If the correction factor formulae were credible, it would make sense to examine the sensitivity

[7] One or two valves would be required, depending on whether the compressor is a rotary or reciprocating type.

of model results to uncertainty in the input variables relating to the correction factors. This type of analysis cannot be done in this case because the correction factor is not credible. Its estimates of cycling losses are grossly inconsistent with data provided by refrigerator manufacturers.

ERA's cycling loss correction factor is composed of two parts. The first is intended to capture the effect of refrigerant migration by decreasing COP by 1.1% times the number of cycles per hour.[8] It is applied to all refrigerators that do not contain a shutoff valve. The magnitude of this error is obviously dependent on the quantity of refrigerant charge that migrates, it will therefore vary significantly among refrigerators. ADL provides no justification for applying this value to all refrigerators, regardless of charge. Moreover, there is no justification for applying the same factor to refrigerators equipped with rotary vs. reciprocating compressors. Rotaries have a high-side sump and it takes longer for the refrigerant (which dissolves in the oil during the off-cycle) to vaporize and begin circulating during the on-cycle (it must wait for the can to heat up). Since ERA ignores such factors as compressor thermal mass it cannot model this difference, and the correction factor cannot be altered by the user to account for the presence of a rotary compressor.

AHAM Task Force members indicated that no correction factor that is independent of run time (duty cycle) can be credible. On theoretical grounds alone, the percentage error due to refrigerant migration must increase as run time decreases. To illustrate this effect one manufacturer provided experimental data showing measured COP for two refrigerators at 40% < runtime < 60% and 45 min < cycle length < 75 min. These results showed that runtime affected the "cycling loss correction factor" under these conditions by 10% for reciprocating compressors and by as much as 100% for rotaries. Clearly the effect of runtime cannot be ignored as in the ERA model. Moreover the variation between rotary and reciprocating compressors demonstrates that factors other than refrigerant migration may have greater effects on cycling losses.

The second part of the cycling correction applies only to systems having a shutoff valve. It actually increases the COP as a function of duty cycle to approach asymptotically the hypothetical case in which the increase in cycling frequency combines with the thermal mass of the heat exchangers to hold evaporating and condensing temperatures at levels corresponding to the case of a variable-speed compressor operating continuously with no cycling loss. ADL uses a ratio of Carnot COP's to adjust the actual COP for the change in evaporating and condensing temperatures. Again, ADL offers no justification for applying to all refrigerators the quantitative correction observed by Janssen et. al. (1990) in a single experimental apparatus containing heat exchangers of unknown thermal mass.

To test the efficacy of the ERA correction factor for systems equipped with a shutoff valve, the tests described above were expanded by adding a shutoff valve to the refrigerators with rotary compressors.

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[8] This value is taken directly from the reference to the single experiment reported by Janssen et. al. (1990).

Substantial energy savings were achieved, but cycling performance was still worse than steady-state. In contrast the ERA correction factors, computed for these same refrigerators, predicted a 1-3% *increase* in COP over the steady-state value.

Because of these obvious defects in the structure of the equations dealing with cycling losses, the accuracy analysis was terminated at this point.

Dual evaporator

ERA represents the dual evaporator system as a special case of the Lorenz cycle, keeping both evaporators in series and simply eliminating one of the intercoolers. The particular configuration considered in this analysis has no air exchange between the two compartments, and assumes The primary mode of energy conservation in this case would be the elimination of the evaporator fan power and the compressor energy to remove that heat from the cabinet. The only other potential for conserving energy would seem to be savings in defrost energy, since no humid air from the fresh food compartment would contact the freezer evaporator. However such savings would not be observed on the DoE test, or in simulations of closed-door operation. It appears that ERA is unable to model other dual-evaporator configurations that might exploit other types of energy conservation opportunities.

The base case refrigerator for this experiment was assumed to have a single evaporator, and subject to the following uncertainties. The conductance, or U-value for two-phase operation was assumed to be known within ± 10%.[9] The volumetric air flow rate over the evaporator is also known within 10%, and the evaporator fan power within 5%. The combined effect of these evaporator-specific uncertainties propagated through the model to produce a ± 0.73% uncertainty in ERA's estimate of energy use.

The dual evaporator system contains none of the aforementioned components, which were replaced by two static evaporators defined in terms of their conductances and their areas. To ensure that the cabinet temperatures were comparable between the single- and dual-evaporator simulations, the areas of the two evaporators were defined to achieve the proper temperatures in each compartment under closed-door conditions in a 90°F environment.[10] ERA's calculational assist menus were used to calculate the conductances of these natural-draft evaporators, which are assumed to be known only within ±15%

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9 Actually a typical evaporator calorimeter is able to compare evaporators within ± 5%, but the absolute value is more difficult to determine. Since a different type of calorimeter would have to be used for static evaporators, it is appropriate to compare the absolute values.

10 This assumption helps the energy portion of the analysis, but introduces great uncertainty into the cost analysis. The required area might be substantially larger or smaller than the value predicted by ERA. It is made here to enable the analysis to proceed and yield some insight into the sensitivity of model results to this parameter. However it recognizes that the cost difference between a single-evaporator and dual-evaporator systems will require a detailed cost analysis anyway.

because of the relative difficulty of the calorimetry problem. This produced a ± 1.5% uncertainty in energy use.

The energy savings associated with eliminating the fan were approximately 13%. Considering the uncertainties, the savings are estimated in this quantitative analysis to lie within the 11-15% range. However, this quantitative analysis may not address the most important sources of uncertainty.

For example the areas of the plate-style static evaporators are much greater than the area of the single forced-draft evaporator. If the resulting charge requirements are also proportionately larger, cycling losses due to refrigerant migration will also increase.

The greatest uncertainty, however, is associated with the temperature control problem, which the ERA model ignores. Overcoming this problem will certainly make the system more costly, for example by adding the cost of refrigerant switching valves, sensors and controllers to the extra cost of evaporator surface areas. It may also increase energy requirements, depending on the control option chosen. For example exchanging air between the compartments may erode defrosting savings, fans might be needed to prevent stratification, and pulldown requirements might demand a larger compressor thereby shortening the duty cycle.

Conclusions and recommendations

This report described both the qualitative and quantitative aspects of the accuracy analysis conducted for the conservation options listed in Table 1. The options were selected because there is inadequate test data available to validate the parts of the ERA model used to calculate energy savings. For a few of these conservation options, limited amounts of test data were provided by manufacturers on a confidential basis. In some cases it was possible to conclude, based on this admittedly sparse and confidential data, that certain parts of the ERA program must be improved before they can be used to accurately predict energy savings associated with certain conservation options. Except for the gasket heat leak and vacuum panel insulation analysis, the quantitative part of the analysis could not be completed because of erroneous assumptions and omissions in the ERA model itself. The findings reported above lead to the following conclusions and recommendations:

- Estimates of energy savings obtainable from gasket improvements should be determined directly by experiment. ERA's calculational assist yields results that fail to agree with published data and experiments conducted by manufacturers, and may lead to errors of 30% or more.
- ERA's estimates of energy savings due to vacuum panel insulation may be in error by ±10% due to uncertainties about enclosure thickness and and panel resistivity alone. Careless use of laboratory-reported k-values for non-CFC foam can increase these errors to ± 50%. ERA's failure to deal with 3-D effects apparently led to even larger errors in tests conducted by one manufacturer.

- The NIST thermodynamic property routines used by ERA may lead to errors in energy use exceeding 1% for R-134a. More accurate routines are available from NIST in identical format and should be used for all R-134a simulations.
- Serious errors apparently exist in the part of ERA that simulates Lorenz cycles. Repeated attempts by an experienced refrigerator designer failed to produce a result showing a temperature glide of the magnitude predicted by the NIST property data. Also ERA's use of a constant specific heat to approximate the behavior of zeotropes causes the specific heat ratio to be in error by as much as a factor of 7 as mixture composition changes through the evaporator. This can lead to serious errors in high-effectiveness heat exchangers.
- For dual-evaporator systems as well as Lorenz-cycle systems, ERA's inability to model cabinet temperature control strategies and off-design performance introduces great uncertainty into any estimates of energy savings. A system optimized for operation in a 90°F environment is almost certain to experience control problems at lower ambients and in response to realistic door-opening schedules, and the energy cost of eliminating these problems is not included in the model.
- ERA's ability to predict the effects of alternative refrigerants or zeotropic blends is further compromised by ERA's compressor rating point model because it contains empirical relations that are specific to R-12 and leading to errors of 5-8% in predicting energy use for R-134a. Likewise the compressor physical model is deficient because it demands that the user specify parameters that are unfamiliar to compressor designers, thereby leading to input errors. A simple model requiring the user to specify a constant isentropic efficiency might be more accurate. It is recommended that neither the physical model nor the rating point model be used; compressor calorimeter data should be obtained for analysis of any conservation options involving alternative refrigerants, mixtures or variable-speed compressors.
- The cycling loss correction in the ERA model should be removed, and refrigerators compared on the basis of steady-state performance. The cycling loss correction in the model extrapolates results obtained from a single test refrigerator, fails to account properly for significant variables (e.g. refrigerant/oil solubility; runtime), and yields results that differ greatly from test data obtained by manufacturers.
- The output format should be changed to display daily energy use to more significant digits. This would facilitate analyses of parametric uncertainty, and would help designers evaluate more accurately the derivative of energy use with respect to input variables.
- The user input menu should be changed to accept at least three significant digits for the value of foam conductivity. The two digits currently accepted are inadequate to support even the three-digit results calculated for daily energy use, which is extremely sensitive to foam conductivity.

The preceding suggestions are necessarily conservative and negative because they are based on very limited data. Only few data points are needed to reject a hypothesis, while many are required to confirm it. The ERA model represents a significant improvement over its predecessor, and might only require modest improvements in order to become a useful tool for analyzing impacts of these energy-conserving designs. However it will take considerable time and effort to amass the data needed to improve and validate many of the features evaluated here. Data on this set of conservation options are very scarce at this time, and are therefore held as proprietary information by the companies that developed it. Until more experience is acquired with these technologies and data become widely available in the public domain, the predictions of the ERA model must be interpreted with great care.

A final cautionary note is necessary as a reminder that the ERA model is a design model, not a simulation model. Even if all inputs and equations were known with perfect certainty, the model would only be capable of representing performance at a single design point. It could not simulate off-design performance (e.g. the effect of changing ambient temperature, the effect of additional cabinet loads due to door openings.) The user-specified input values for evaporator superheat and condenser subcooling are intended to compensate for ERA's lack of equations describing the mass flow-pressure drop relation for the capillary tube, and equations that would keep track of the refrigerant charge. Therefore when a user specifies a different kitchen temperature, for example, the ERA model assumes that the refrigerator has been recharged and fitted with a new capillary tube designed to produce the user-specified superheat and subcooling at that new ambient temperature. Therefore a refrigerator design predicted by ERA to perform well at the design point (90°F ambient; 5°F freezer; 40°F fresh food) will not necessarily keep food from spoiling as the ambient temperature changes. To predict off-design performance characteristics such as this requires a true simulation model that models explicitly the behavior of the capillary tube and the charge inventory.

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Attachment 5
1/2

AHAM NAECA Engineering Analysis Task Force

TOP MOUNT AUTO-DEFROST NON-DISPENSER MODEL

Ranking per Design Option (1=highest feasibility : 45=lowest)	Design Feasibility RANK		Marketing Utility RANK		Energy Savings RANK	
1a) Increased Cabinet Insul. - 1/2"	2	99.3%	36	31.7%	5	65
1b) Increased Cabinet Insul. - 1"	6	95.0%	37	20.0%	2	95
1c) Increased Cabinet Insul.-1 1/2"	9	87.5%	38	11.7%	1	123
2a) Increased Door Insulation - 1/2"	1	99.3%	30	75.8%	18	27
2b) Increased Door Insulation - 1"	5	95.0%	34	55.8%	14	36
2c) Increased Door Insulation - 1 1/2"	10	87.5%	35	34.2%	10	47
3) Improved Foam Insul.(ie.microcell)	12	76.3%	21	100.0%	16	33
4) Evac. Insul. Panels (all types)	19	49.8%	2	100.0%	4	71
5) Gas Filled Panels (inert gas)	32	18.8%	1	100.0%	8	51
6) Improved Gaskets	18	53.3%	28	83.3%	27	12
7) Double Door Gaskets	14	56.0%	33	57.5%	13	36
8) Reduced Heat Load for Dispensers	44	0.0%	41	0.0%	40	0
9) Reduction in Electric Anti-sweat	21	37.5%	10	100.0%	24	17
10) Subst. of Hot Gas Anti-sweat	22	36.0%	11	100.0%	9	49
11) Reduction in Auto-defrost Energy	24	32.5%	7	100.0%	23	17
12) Subst. of Hot Gas Defrost	38	14.5%	15	100.0%	26	12
13) Adaptive Defrost Systems	3	98.8%	18	100.0%	21	18
14) Improved Compressor Efficiency	11	87.3%	24	95.7%	6	60
15) Two Compressor System	17	55.0%	32	70.0%	44	(3)
16) Variable Speed Compressor	16	55.3%	6	100.0%	20	25
17a) Improved Evap Fan Mtr Effic.	4	95.0%	16	100.0%	11	37
17b) Improved Cond. Fan Mtr Effic.	20	47.5%	3	100.0%	19	25
18a) Improved Evap Fan Effic.	34	16.5%	4	100.0%	35	4
18b) Improved Cond. Fan Effic.	43	0.0%	39	0.0%	41	0
19) Variable Speed Fans	27	27.0%	9	100.0%	22	17

Design Feasibility/Marketing Utility = 100%(Highly Feasible/Saleable) / 0%(Not Feasible/Saleable) : Energy Savings (in kWh/yr

Attachment 5
2/2

20) Two Stage Two Evap System	23	34.5%	31	73.3%	3	78
21a) Other Cycles - Lorenz	25	32.3%	29	80.8%	12	37
21b) Other Cycles - Stirling	41	0.3%	17	100.0%	7	54
21c) Other Cycles - Gas Absorption	39	14.3%	44	0.0%	36	3
21d) Other Cycles - Thermoacoustic	40	0.3%	14	100.0%	15	34
22a) Impr Evap HX - Increased Area	7	92.5%	26	95.0%	29	10
22b) Impr Evap HX - Enhanced Surface	15	55.5%	5	100.0%	28	11
22c) Impr Evap HX - High Thermal Mass	29	22.0%	25	95.0%	31	7
22d) Impr Evap HX - Integrated Surf	28	24.8%	22	97.8%	32	6
23a) Impr Cond HX - Increased Area	8	91.3%	23	97.5%	30	10
23b) Impr Cond HX - Enhanced Surface	31	20.3%	12	100.0%	34	5
23c) Impr Cond HX - High Thermal Mass	30	20.8%	27	95.0%	38	2
23d) Impr Cond HX - Integrated Surf	26	27.3%	8	100.0%	37	3
24) Alternative Refrigerant	45	0.0%	40	0.0%	42	0
25) Impr Expansion Valve (electronic)	35	16.3%	20	100.0%	25	15
26) Fluid Control Valves	13	66.0%	13	100.0%	17	29
27) Location of Compressor & Fans	33	17.8%	45	0.0%	33	5
28) Use of Natural Convection	37	16.0%	43	0.0%	45	(15)
29) Electrohydrodynamic Enhanced HX	42	0.0%	42	0.0%	39	0
30) Voltage Controller	36	16.0%	19	100.0%	43	0

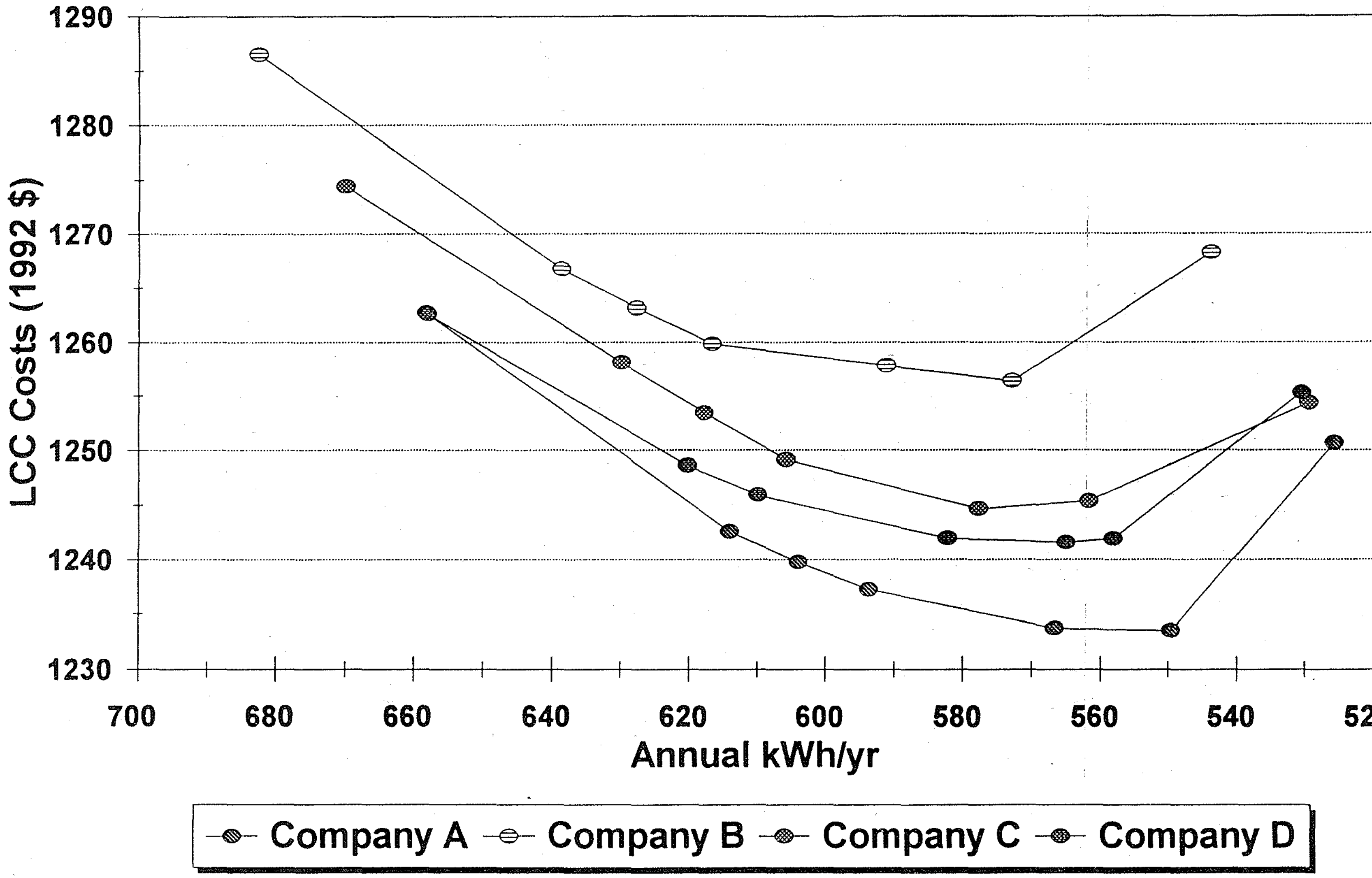
L.J. Swatkowski - AHAM Chicago 03/11/94

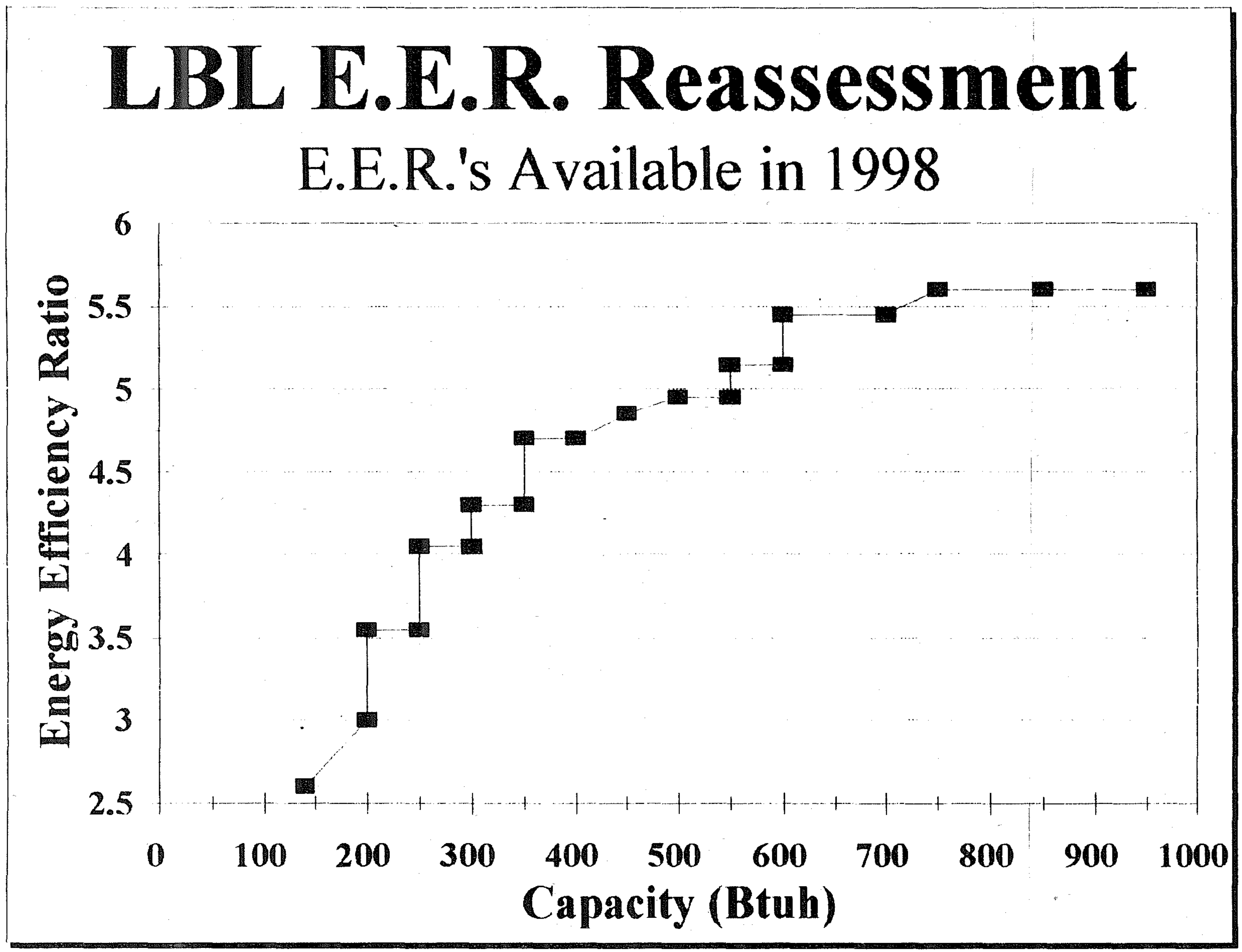
100% (Highly Feasible/Saleable) / 0%(Not Feasible/Saleable) : Energy Savings (in kW

Attachment 7

LBL/ERA Analysis (12.23.93)

AHAM Data Request




LBL E.E.R. Reassessment
E.E.R.'s Available in 1998
Energy Efficiency Ratio
6
5.5
5
4.5
4
3.5
3
2.5
0
100
200
300
400
500
600
700
800
900
1000
Capacity (Btuh)

INDUSTRY PROPOSED 1998 ENERGY EFFICIENCY STANDARDS
FOR REFRIGERATOR/FREEZERS

Product Categories	Adjusted Volume	Equation	Percent Below 1993 Standards
Top-mount without dispenser	21.19	11.0 AV + 315	21%
Top-mount with dispenser	24.94	11.0 AV + 385	20%
Side-by-side with dispenser	26.05	11.0 AV + 446	23%
Side-by-side without dispenser	27.39	5.0 AV + 514	21%
Bottom-mounts	24.72	5.0 AV + 496	20%

Attachment 10

Life Cycle Costs and Payback Periods of 5.5 cuft Manual-Defrost Refrigerator
ERA Simulation: Draft Version

Level	Option	Manuf'r Cost (1992$)	Incr. Mfr Cost (1992$)	Retail Price (1992$)	Annual Energy Use (kWh)	Annual Energy Cost (1992$)	Cumulative Payback (years)	Lifecycle Costs (1992$)			Cumulative CCE (cents/kWh)			Duty Cycle (%)
								4%	6%	10%	4%	6%	10%	
0	BASELINE	$95.65	-	$220.00	367.74	$32.36	NA	$509.65	$480.15	$433.39	NA	NA	NA	57.2%
1	0 + 3.35 EER Compressor	$99.95	$4.30	$229.89	275.63	$24.26	1.22	$446.98	$424.88	$389.82	$1.20	$1.34	$1.63	56.9%
2	1 + Enhanced Evaporator HT Surface	$100.56	$0.60	$231.28	265.66	$23.38	1.26	$440.52	$419.22	$385.43	$1.23	$1.37	$1.68	49.9%
3	2 + Enhanced Condenser HT Surface	$101.16	$0.60	$232.67	261.28	$22.99	1.35	$438.47	$417.51	$384.28	$1.33	$1.48	$1.81	49.4%
4	3 + Reduce Gasket Heat Leak	$102.16	$1.00	$234.97	255.94	$22.52	1.52	$436.56	$416.03	$383.48	$1.50	$1.67	$2.03	48.4%

Assumptions:

(1) Energy consumptions for the baseline and for each design option were obtained from an ERA simulation of an actual 5.5 cubic foot manual-defrost refrigerator. A correction factor of 0.666 was applied to the ERA values in order to account for the difference between the simulated and the actual baseline usage.

(2) Manufactures cost of the baseline unit was obtained by dividing the retail price of a 6 cuft manual-defrost refrigerator-freezer from 1992/93 fall/winter SEARS catalogue by the the markup factor.

(3) Electricity cost = 0.088 $/kWh (average cost in 1998 obtained from an interpolation of the 1995 and 2000 prices of electricity forecast in DOE's Annual Energy Outlook 1993, inflated to 1992 dollars). The interpolated value (for 1998) is 0.082 $/kWh (1991$). After adjusting for inflation from 1991 to 1992, it becomes 0.085 $/kWh. The electricity price was then adjusted by an enduse factor for refrigerators of 1.04.

(4) Installation and maintenance costs are not included in the above calculations.

(5) Lifetime =11.3 years.

(6) Markup factor= 2.3. This represents the average of markup factors for this product class in the 1989 TSD.

(7) Baseline evaporator area is 0.448 sqm; condenser area is 0.22 sqm.

(8) Baseline compressor EER is 2.65

Attachment 11
1/7

Life Cycle Costs and Payback Periods of 2.9 cuft Partial Auto-Defrost Top_Mount Refrigerator-Freezer
ERA Simulation: Draft Version

Level	Option	Manufr Cost (1992$)	Incr. Mfr Cost (1992$)	Retail Price (1992$)	Annual Energy Use (kWh)	Annual Energy Cost (1992$)	Cumulative Payback (years)	Lifecycle Costs (1992$)			Cumulative CCE (cents/kWh)			Duty Cycle (%)
								4%	6%	10%	4%	6%	10%	
0	BASELINE	$95.65	-	$220.00	410.07	$36.09	NA	$542.98	$510.10	$457.95	NA	NA	NA	78.2%
1	0 + 3.55 EER Compressor	$97.75	$2.10	$224.83	363.67	$32.00	1.18	$511.27	$482.10	$435.85	$1.16	$1.29	$1.58	78.1%
2	1 + Enhanced Condenser HT Surface	$98.46	$0.71	$226.47	355.56	$31.29	1.35	$506.52	$478.00	$432.79	$1.33	$1.48	$1.80	76.8%
3	2 + Reduce Gasket Heat Leak	$100.19	$1.73	$230.44	348.01	$30.63	1.91	$504.55	$476.63	$432.38	$1.88	$2.09	$2.55	75.2%
4	3 + Increased Evaporator Area	$105.53	$5.34	$242.72	339.63	$29.89	3.66	$510.22	$482.98	$439.79	$3.60	$4.01	$4.89	71.3%
5	4 + Reduce Anti-Sweat Heat	$116.09	$10.56	$267.01	339.07	$29.84	7.52	$534.08	$506.88	$463.76	$7.40	$8.24	$10.08	71.3%

Assumptions:

(1) Energy consumptions for the baseline and for each design option were obtained from an ERA simulation of an actual 2.9 cubic foot partial auto-defrost top-mount refrigerator-freezer. A correction factor of 0.766 was applied to the ERA values in order to account for the difference between the simulated and the actual baseline usage.

(2) Manufacturer cost of the baseline unit was obtained by dividing the retail price of a 2. cuft manual-defrost top-mount refrigerator-freezer from 1992/93 fall\winter SEARS catalogue by the the markup factor.

(3) Electricity cost = 0.088 $/kWh (average cost in 1998 obtained from an interpolation of the 1995 and 2000 prices of electricity forecast in DOE's Annual Energy Outlook 1993, inflated to 1992 dollars). The interpolated value (for 1998) is 0.082 $/kWh (1991$). After adjusting for inflation from 1991 to 1992, it becomes 0.085 $/kWh. The electricity price was then adjusted by an enduse factor for refrigerators of 1.04.

(4) Installation and maintenance costs are not included in the above calculations.

(5) Lifetime =11.3 years.

(6) Markup factor= 2.3. This represents the average of markup factors for this product class in the 1989 TSD.

(7) Baseline evaporator area is 0.096 sqm; condenser area is 0.28 sqm.

(8) Baseline compressor EER is 3.11

Attachment 11
2/7

Life Cycle Costs and Payback Periods of 3.5 cuft Auto-Defrost All-Refrigerator.
ERA Simulation: Draft Version

Level	Option	Manuf'r Cost (1992$)	Incr. Mfr Cost (1992$)	Retail Price (1992$)	Annual Energy Use (kWh)	Annual Energy Cost (1992$)	Cumulative Payback (years)	Lifecycle Costs (1992$)			Cumulative CCE (cents/kWh)			Duty Cycle (%)
								4%	6%	10%	4%	6%	10%	
0	BASELINE	$76.52	-	$176.00	585.29	$51.51	NA	$637.00	$590.05	$515.62	NA	NA	NA	69.6%
1	0 + 3.55 EER Compressor	$78.62	$2.10	$180.83	519.63	$45.73	0.84	$590.11	$548.43	$482.35	$0.82	$0.92	$1.12	69.3%
2	1 + Reduce Condenser Fan Motor Power	$83.12	$4.50	$191.18	468.96	$41.27	1.48	$560.55	$522.94	$463.30	$1.46	$1.62	$1.98	69.3%
3	2 + Increase Condenser Area	$88.53	$5.41	$203.63	447.89	$39.41	2.29	$556.40	$520.48	$463.52	$2.25	$2.50	$3.05	66.9%
4	3 + Increase Evaporator Area	$99.36	$10.83	$228.53	421.95	$37.13	3.65	$560.88	$527.03	$473.37	$3.59	$4.09	$4.88	55.3%
5	4 + Reduce Gasket Leak	$104.63	$5.27	$240.65	416.68	$36.67	4.36	$568.84	$535.42	$482.43	$4.28	$4.77	$5.82	54.6%

Assumptions:
(1) Energy consumptions for the baseline and for each design option were obtained from an ERA simulation of an actual 3.5 cubic foot cyclic-defrost refrigerator. A correction factor of 1.11 was applied to the ERA values in order at account for the difference between the simulated and the actual baseline usage.
(2) Manufacturer cost of the baseline unit was obtained by dividing the manufacturer suggested retail price by the the markup factor.
(3) Electricity cost = 0.088 $/kWh (average cost in 1998 obtained from an interpolation of the 1995 and 2000 prices of electricity forecast in DOE's Annual Energy Outlook 1993, inflated to 1992 dollars). The interpolated value (for 1998) is 0.082 $/kWh (1991$). After adjusting for inflation from 1991 to 1992, it becomes 0.085 $/kWh. The electricity price was then adjusted by an enduse factor for refrigerators of 1.04.
(4) Installation and maintenance costs are not included in the above calculations.
(5) Lifetime =11.3 years.
(6) Markup factor= 2.3. This represents the average of markup factors for this product class in the 1989 TSD.
(7) Baseline: Compressor EER 3.11. Evaporator and condenser areas, 0.209 and 0.76 sqm respectively. Condenser fan motor power, 12.0W

Life Cycle Costs and Payback Periods of 3.5 cuft Auto-Defrost Refrigerator.
ERA Simulation: Draft Version

Level	Option	Manuf'r Cost (1992$)	Incr. Mfr Cost (1992$)	Retail Price (1992$)	Annual Energy Use (kWh)	Annual Energy Cost (1992$)	Cumulative Payback (years)	Lifecycle Costs (1992$)			Cumulative CCE (cents/kWh)			Duty Cycle (%)
								4%	6%	10%	4%	6%	10%	
0	BASELINE	$76.52	-	$176.00	418.72	$36.85	NA	$505.80	$472.21	$418.96	NA	NA	NA	100.0%
1	0 + 3.55 EER Compressor	$78.62	$2.10	$180.83	380.22	$33.46	1.43	$480.30	$449.81	$401.45	$1.40	$1.56	$1.90	100.0%
2	1 + Reduce Condenser Fan Motor Power	$83.12	$4.50	$191.18	331.54	$29.18	1.98	$452.31	$425.72	$383.56	$1.95	$2.17	$2.64	100.0%

Assumptions:

(1) Energy consumptions for the baseline and for each design option were obtained from an ERA simulation of an actual 3.5 cubic foot cyclic-defrost refrigerator. A correction factor of 0.872 was applied to the ERA values in order to account for the difference between the simulated and the actual baseline usage.

(2) Manufacturer cost of the baseline unit was obtained by dividing the manufacturer suggested retail price by the the markup factor.

(3) Electricity cost = 0.088 $/kWh (average cost in 1998 obtained from an interpolation of the 1995 and 2000 prices of electricity forecast in DOE's Annual Energy Outlook 1993, inflated to 1992 dollars). The interpolated value (for 1998) is 0.082 $/kWh (1991$). After adjusting for inflation from 1991 to 1992, it becomes 0.085 $/kWh. The electricity price was then adjusted by an enduse factor for refrigerators of 1.04.

(4) Installation and maintenance costs are not included in the above calculations.

(5) Lifetime =11.3 years.

(6) Markup factor= 2.3. This represents the average of markup factors for this product class in the 1989 TSD.

(7) Baseline: Compressor EER 3.11, Condenser fan motor power 12.0W

Attachment 11
4/7

Life Cycle Costs and Payback Periods of 5 cuft Manual Defrost Chest Freezer
ERA Simulation: Draft Version

Level	Option	Manufr Cost (1992$)	Incr. Mfr Cost (1992$)	Retail Price (1992$)	Annual Energy Use (kWh)	Annual Energy Cost (1992$)	Cumulative Payback (years)	Lifecycle Costs (1992$)			Cumulative CCE (cents/kWh)			Duty Cycle (%)
								4%	6%	10%	4%	6%	10%	
0	BASELINE	$96	-	$220	253.43	$22.30	NA	$419.61	$399.28	$367.05	NA	NA	NA	65.6%
1	0 + 4.30 EER Compressor	$98	$2.79	$226	227.85	$20.05	2.85	$405.88	$387.61	$358.63	$2.80	$3.12	$3.81	65.5%
2	1 + Add 1" Insulation to Walls	$125	$26.77	$288	186.47	$16.41	11.54	$434.86	$419.91	$396.20	$11.35	$12.63	$15.40	54.6%
3	2 + Add 1" Insulation to Door	$135	$10.20	$311	178.79	$15.73	13.93	$452.28	$437.94	$415.21	$13.69	$15.24	$18.58	52.4%
4	3 + Reduce Gasket Heat Leak	$138	$2.83	$318	177.17	$15.59	14.60	$457.50	$443.29	$420.76	$14.35	$15.98	$19.48	51.9%

Assumptions:

(1) Energy consumptions for the baseline and for each design option were obtained from an ERA simulation of an actual 5 cubic foot manual-defrost chest freezer. A correction factor of 0.91 was applied to the ERA values in order to account for the difference between the simulated and the actual baseline usage. A correction factor of 0.7 was also applied as specified in the test procedure for chest freezers.

(2) Manufacturer cost of the baseline unit was obtained by dividing the retail price of a 5.3 cuft chest freezer from 1992/93 fall/winter SEARS catalogue by the the markup factor.

(3) Electricity cost = 0.088 $/kWh (average cost in 1998 obtained from an interpolation of the 1995 and 2000 prices of electricity forecast in DOE's Annual Energy Outlook 1993, inflated to 1992 dollars). The interpolated value (for 1998) is 0.082 $/kWh (1991$). After adjusting for inflation from 1991 to 1992, it becomes 0.085 $/kWh. The electricity price was then adjusted by an enduse factor for refrigerators of 1.04.

(4) Installation and maintenance costs are not included in the above calculations.

(5) Lifetime =11.3 years.

(6) Markup factor= 2.3. This represents the average of markup factors for this product class in the 1989 TSD.

(7) Baseline: Compressor EER 3.88; evaporator area is 0.76 sqm. Door and walls (average of sides and bottom) insulation thickness 4.60 cm (1.81") and 6.37 cm (2.51"), respectively.

(8) Resistivity of wall and door insulation: 0.55 sqm-C/W-cm

Attachment 11
5/7

Life Cycle Costs and Payback Periods of 5.0 cuft Manual-Defrost Upright Freezer.
ERA Simulation: Draft Version

Level	Option	Manuf'r Cost (1992$)	Incr. Mfr Cost (1992$)	Retail Price (1992$)	Annual Energy Use (kWh)	Annual Energy Cost (1992$)	Cumulative Payback (years)	Lifecycle Costs (1992$)			Cumulative CCE (cents/kWh)			Duty Cycle (%)
								4%	6%	10%	4%	6%	10%	
0	BASELINE	$95.65	-	$220.00	410.71	$36.14	NA	$543.49	$510.55	$458.32	NA	NA	NA	80.6%
1	0 + 4.70 EER Compressor	$104.99	$9.34	$241.48	318.31	$28.01	2.64	$492.19	$466.66	$426.18	$2.60	$2.89	$3.53	80.5%
2	1 + Enhanced Condenser HT Surface	$106.06	$1.07	$243.95	311.59	$27.42	2.75	$489.37	$464.38	$424.75	$2.70	$3.01	$3.66	78.9%
3	2 + Reduce Gasket Heat Leak	$107.13	$1.07	$246.41	306.22	$26.95	2.87	$487.60	$463.04	$424.10	$2.82	$3.14	$3.83	77.5%

Assumptions:

(1) Energy consumptions for the baseline and for each design option were obtained from an ERA simulation of an actual 5.0 cubic foot manual-defrost upright freezer. A correction factor of 0.866 was applied to the ERA values in order to account for the difference between the simulated and the actual baseline usage. A correction factor of 0.85 was also applied as specified in the test procedure for upright freezers.

(2) Manufacturer cost of the baseline unit was obtained by dividing the retail price of a 5 cuft manual-defrost upright refrigerator-freezer from 1992/93 fall\winter SEARS catalogue by the the markup factor.

(3) Electricity cost = 0.088 $/kWh (average cost in 1998 obtained from an interpolation of the 1995 and 2000 prices of electricity forecast in DOE's Annual Energy Outlook 1993, inflated to 1992 dollars). The interpolated value (for 1998) is 0.082 $/kWh (1991$). After adjusting for inflation from 1991 to 1992, it becomes 0.085 $/kWh. The electricity price was then adjusted by an enduse factor for refrigerators of 1.04.

(4) Installation and maintenance costs are not included in the above calculations.

(5) Lifetime =11.3 years.

(6) Markup factor= 2.3. This represents the average of markup factors for this product class in the 1989 TSD.

(7) Baseline condenser area is 0.29 sqm.

(8) Baseline compressor EER is 3.65

Life Cycle Costs and Payback Periods of 1.7 cuft Manual Defrost Refrigerator
ERA Simulation: Draft Version

Level	Option	Manufr Cost (1992$)	Incr. Mfr Cost (1992$)	Retail Price (1992$)	Annual Energy Use (kWh)	Annual Energy Cost (1992$)	Cumulative Payback (years)	Lifecycle Costs (1992$)			Cumulative CCE (cents/kWh)			Duty Cycle (%)
								4%	6%	10%	4%	6%	10%	
0	BASELINE	$76.52	-	$176.00	277.20	$24.39	NA	$394.33	$372.09	$336.84	NA	NA	NA	67.1%
1	0 + Enhanced Condenser HT Surface	$76.92	$0.40	$176.93	271.09	$23.86	1.73	$390.45	$368.71	$334.23	$1.70	$1.90	$2.31	66.2%
2	1 + Enhanced Evaporator HT Surface	$77.53	$0.60	$178.32	264.99	$23.32	2.16	$387.03	$365.78	$332.08	$2.12	$2.37	$2.88	63.6%
3	2 + Reduce Gasket Heat Leak	$78.53	$1.00	$180.63	259.85	$22.87	3.03	$385.30	$364.45	$331.41	$2.98	$3.32	$4.05	62.4%

Assumptions:

(1) Energy consumptions for the baseline and for each design option were obtained from an ERA simulation of an actual 1.7 cubic foot manual-defrost refrigerator. A correction factor of 0.88 was applied to the ERA values in order to account for the difference between the simulated and the actual baseline usage.

(2) Manufacturer cost of the baseline unit was obtained by dividing the manufacturer suggested retail price by the the markup factor.

(3) Electricity cost = 0.088 $/kWh (average cost in 1998 obtained from an interpolation of the 1995 and 2000 prices of electricity forecast in DOE's Annual Energy Outlook 1993, inflated to 1992 dollars). The interpolated value (for 1998) is 0.082 $/kWh (1991$). After adjusting for inflation from 1991 to 1992, it becomes 0.085 $/kWh. The electricity price was then adjusted by an enduse factor for refrigerators of 1.04.

(4) Installation and maintenance costs are not included in the above calculations.

(5) Lifetime = 11.3 years.

(6) Markup factor = 2.3. This represents the average of markup factors for this product class in the 1989 TSD.

(7) Evaporator area is 0.15 sqm and Condenser area is 0.22 sqm.

Attachment 11
7/7

Attachment 13
1/2

AHAM Compact/Undercounter Task Force

Design Option	Design Feasibility		Energy Savings		Marketing Utility	
	RANK		RANK		RANK	
1a)Increased Cabinet Insul. - 1/2"	6	57.5%	3	5.4%	19	25.0%
1b)Increased Cabinet Insul. - 1"	15	39.2%	1	7.8%	25	16.7%
2a)Increased Door Insul. - 1/2"	2	65.8%	6	3.5%	16	40.3%
2b)Increased Door Insul. - 1"	12	41.7%	7	3.3%	17	28.8%
3)Improved Foam Ins.(ie,microcell)	10	46.7%	10	2.5%	3	83.3%
4)Evac. Insul. Panels (all types)	26	8.5%	5	4.5%	10	50.8%
5)Gas Filled Panels (inert gas)	27	7.7%	8	3.3%	11	49.2%
6)Improved Gaskets	4	60.0%	12	2.4%	2	85.0%
7)Double Door Gaskets	14	40.8%	11	2.5%	6	59.2%
8)Reduced Heat Load for Dispensers	41	0.0%	39	0.0%	33	10.0%
9)Reduction in Elec. Anti-sweat	19	22.0%	30	0.0%	22	23.0%
10)Subst. of Hot Gas Anti-sweat	35	2.0%	34	0.0%	18	27.0%
11)Reduction in Auto-Defrost Energy	25	9.0%	23	0.5%	8	53.0%
12)Subst. of Hot Gas Defrost	20	19.0%	15	1.5%	7	55.0%
13)Adaptive Defrost Systems	16	27.0%	17	1.2%	5	63.0%
14)Improved Compressor Efficiency	1	82.1%	2	7.4%	1	90.0%
15)Two Compressor System	36	1.0%	25	0.4%	27	11.0%
16)Variable Speed Compressor	30	3.5%	22	1.0%	14	43.3%
17)Improved Fan Motor Efficiency	3	65.0%	4	5.2%	4	74.0%
18)Improved Fan Efficiency	18	25.0%	18	1.2%	23	21.0%
19)Variable Speed Fans	34	2.0%	33	0.0%	28	11.0%
20)Two Stage Two Evaporative Syste	31	3.0%	32	0.0%	35	4.0%
21a)Other Cycles - Lorenz	40	0.0%	38	0.0%	37	2.0%
21b)Other Cycles - Stirling	39	0.0%	37	0.0%	41	0.0%
21c)Other Cycles - Gas Absorption	24	15.0%	41	-5.0%	38	2.0%
21d)Other Cycles - Thermoacoustic	38	0.0%	36	0.0%	40	0.0%
22a)Impr Evap HX - Increased Area	11	45.8%	14	2.2%	13	47.0%

Attachment 13
2/2

AHAM Compact/Undercounter Task Force

Design Option	Design Feasibility RANK		Energy Savings RANK		Marketing Utility RANK	
22b)Impr Evap HX - Enhanced Surfac	13	40.8%	16	1.2%	24	19.0%
22c)Impr Evap HX - High Therm Mass	22	18.3%	27	0.2%	30	10.0%
22d)Impr Evap HX - Integr Surfaces	33	2.0%	29	0.2%	32	10.0%
23a)Impr Cond HX - Increased Area	9	48.3%	9	2.8%	9	52.0%
23b)Impr Cond HX - Enhanced Surfac	8	48.8%	13	2.4%	15	42.0%
23c)Impr Cond HX - High Therm Mass	21	18.3%	26	0.2%	29	10.0%
23d)Impr Cond HX - Integr Surfaces	32	2.0%	28	0.2%	31	10.0%
24)Alternative Refrigerant	5	58.2%	40	-0.6%	12	49.0%
25)Impr Expansion Valve(electronic)	28	6.7%	31	0.0%	34	5.0%
26)Fluid Control Valves	17	25.8%	20	1.0%	20	25.0%
27)Location of Comprsr. & Fans	23	15.0%	21	1.0%	36	3.0%
28)Use of Natural Convection	7	50.0%	19	1.0%	21	24.0%
29)Electrohydrodynmc Enhanced HX	37	0.0%	35	0.0%	39	0.0%
30)Voltage Controller	29	4.0%	24	0.4%	26	15.0%

18-Aug-93 LJS

Feasibility Ranking of 41 design Options for the 1998 NAECA Rulemaking

6/14/94

Life Cycle Costs and Payback Periods of 15 cuft Manual Defrost Chest Freezer
ERA Simulation: Draft Version

Level	Option	Manuf'r Cost (1992$)	Incr. Mfr Cost (1992$)	Retail Price (1992$)	Annual Energy Use (kWh)	Annual Energy Cost (1992$)	Cumulative Payback (years)	Lifecycle Costs (1992$) 4%	Lifecycle Costs (1992$) 6%	Lifecycle Costs (1992$) 10%	Cumulative CCE (cents/kWh) 4%	Cumulative CCE (cents/kWh) 6%	Cumulative CCE (cents/kWh) 10%
0	BASELINE	$170	-	$392	471.71	$42	NA	$937	$855	$739	NA	NA	NA
1	0 + ADD 1/2" INS TO WALLS	$179	$9.03	$413	392.91	$35	2.99	$867	$799	$702	$2.01	$2.36	$3.15
2	1 + ADD 1" INS TO WALLS	$186	$6.36	$427	345.76	$30	3.19	$827	$767	$682	$2.14	$2.52	$3.36
3	2 + 4.56 EER COMPRESSOR	$191	$5.52	$440	314.55	$28	3.48	$804	$749	$672	$2.33	$2.74	$3.66
4	3 + 4.95 EER COMPRESSOR	$197	$5.25	$452	289.18	$25	3.74	$786	$736	$665	$2.51	$2.95	$3.94
5	4 + ADD 1" INS TO DOOR	$201	$4.08	$462	278.63	$25	4.09	$784	$735	$667	$2.74	$3.23	$4.31
6	5 + REDUCE GASKET LEAK	$202	$1.76	$466	274.59	$24	4.24	$783	$735	$668	$2.84	$3.35	$4.46
7	6 + INCREASE EVAP AREA	$204	$2.07	$470	273.02	$24	4.48	$786	$738	$671	$3.00	$3.53	$4.71
8	7 + ENHANCED COND HT SURFACE	$220	$15.36	$506	266.28	$23	6.29	$813	$767	$702	$4.21	$4.96	$6.61

Assumptions:

(1) Energy consumptions for the baseline and for each design option were obtained from an ERA simulation of an actual 15 cubic foot chest manual-defrost freezer.

(2) Manufacturer cost of the baseline unit was interpolated from the AHAM manufacturer cost vs kWh curve for this product class.
Using a linear interpolation between the two closest points on the AHAM curve to the ERA baseline consumption of 471.71 kWh, the ERA baseline cost is $170.42.

(3) Electricity cost = 0.088 $/kWh (average cost in 1998 obtained from an interpolation of the 1995 and 2000 prices of electricity forecast in DOE's Annual Energy Outlook 1993, inflated to 1992 dollars). The interpolated value (for 1998) is 0.082 $/kWh (1991$). After adjusting for inflation from 1991 to 1992, it becomes 0.085 $/kWh. The electricity price was then adjusted by an enduse factor for refrigerators of 1.04.

(4) Installation and maintenance costs are not included in the above calculations.

(5) Lifetime =19 years.

(6) Markup factor= 2.3. This represents the average of markup factors for this product class in the 1989 TSD.

(7) Baseline: Compressor EER = 4.15.
Insulation thicknesses: freezer door and sides are 6.40 cm (2.52") and 5.46 cm (2.15"). Resistivity of door and walls is 0.55 and 0.55 m2-degC/W-cm.
Evaporator area is 1.56 sqm and Condenser area is 1.08sqm.

Attachment 14
1/4

6/14/94

Life Cycle Costs and Payback Periods of 14 cuft Manual Defrost Upright Freezer
ERA Simulation: Draft Version

Level	Option	Manufr Cost (1992$)	Incr. Mfr Cost (1992$)	Retail Price (1992$)	Annual Energy Use (kWh)	Annual Energy Cost (1992$)	Cumulative Payback (years)	Lifecycle Costs (1992$)			Cumulative CCE (cents/kWh)		
								4%	6%	10%	4%	6%	10%
0	BASELINE	$183	-	$420	482.94	$42	NA	$978	$894	$775	NA	NA	NA
1	0 + FOAM INS TO DOOR	$184	$1.82	$424	424.31	$37	0.81	$915	$841	$736	$0.54	$0.64	$0.85
2	1 + 5.15 EER COMPRESSOR	$190	$5.25	$436	361.15	$32	1.52	$854	$791	$702	$1.02	$1.20	$1.60
3	2 + ENHANCED COND HT	$190	$0.80	$438	353.78	$31	1.59	$847	$785	$698	$1.07	$1.26	$1.68
4	3+ REDUCE GASKET LEAK	$193	$2.33	$443	343.30	$30	1.91	$840	$780	$696	$1.28	$1.51	$2.01
5	4 + ADD 1" INS TO DOOR	$199	$5.99	$457	318.09	$28	2.57	$825	$769	$691	$1.72	$2.02	$2.70
6	5 + ADD 1" INS TO WALLS	$219	$20.23	$504	275.04	$24	4.58	$822	$774	$706	$3.07	$3.61	$4.82
7	6 + INCREASE EVAP AREA	$223	$4.05	$513	273.90	$24	5.06	$830	$782	$715	$3.39	$3.99	$5.32

Assumptions:

(1) Energy consumptions for the baseline and for each design option were obtained from an ERA simulation of an actual 14 cubic foot upright manual-defrost freezer.

(2) Manufacturer cost of the baseline unit was interpolated from the AHAM manufacturer cost vs kWh curve for this product class.
Using a linear interpolation between the two closest points on the AHAM curve to the ERA baseline consumption of 482.94 kWh, the ERA baseline cost is $182.57.

(3) Electricity cost = 0.088 $/kWh (average cost in 1998 obtained from an interpolation of the 1995 and 2000 prices of electricity forecast in DOE's Annual Energy Outlook 1993, inflated to 1992 dollars). The interpolated value (for 1998) is 0.082 $/kWh (1991$). After adjusting for inflation from 1991 to 1992, it becomes 0.085 $/kWh. The electricity price was then adjusted by an enduse factor for refrigerators of 1.04.

(4) Installation and maintenance costs are not included in the above calculations.

(5) Lifetime =19 years.

(6) Markup factor= 2.3. This represents the average of markup factors for this product class in the 1989 TSD.

(7) Baseline: Compressor EER = 4.40.
Insulation thicknesses: freezer door and sides are 6.35 cm (2.50") and 3.81 cm (1.50"). Resistivity of door and walls is 0.286 and 0.521 m2-degC/W-cm.
Evaporator area is 2.15 sqm and Condenser area is 2.15sqm.

Attachment 14
2/4

6/14/94

Life Cycle Costs and Payback Periods of 15 cuft Auto Defrost Upright Freezer:
ERA Simulation: Draft Version

Level	Option	Manufr Cost (1992$)	Incr. Mfr Cost (1992$)	Retail Price (1992$)	Annual Energy Use (kWh)	Annual Energy Cost (1992$)	Cumulative Payback (years)	Lifecycle Costs (1992$)			Cumulative CCE (cents/kWh)		
								4%	6%	10%	4%	6%	10%
0	BASELINE	$296	-	$681	694.24	$61	NA	$1,484	$1,363	$1,192	NA	NA	NA
1	0 + FOAM INSULATION ON DOOR	$298	$1.82	$685	600.78	$53	0.51	$1,380	$1,275	$1,128	$0.34	$0.40	$0.54
2	1 + 5.60 EER COMPRESSOR	$307	$9.05	$706	545.78	$48	1.91	$1,337	$1,242	$1,108	$1.28	$1.51	$2.01
3	2+ ADD 1" INS TO DOOR	$316	$8.93	$727	501.93	$44	2.69	$1,307	$1,220	$1,096	$1.80	$2.12	$2.83
4	3+ DEC EVAP MOTOR POWER	$323	$6.50	$742	481.55	$42	3.23	$1,298	$1,215	$1,096	$2.17	$2.55	$3.40
5	4 + REDUCE GASKET LEAK	$324	$1.94	$746	476.93	$42	3.40	$1,297	$1,215	$1,097	$2.28	$2.68	$3.57
6	5 + ADD 1" INS TO WALLS	$360	$35.84	$829	415.01	$37	6.00	$1,308	$1,236	$1,134	$4.02	$4.73	$6.31
7	6 + ENHANCED EVAP HT	$363	$2.61	$835	411.55	$36	6.16	$1,310	$1,239	$1,138	$4.13	$4.86	$6.49
8	7 + ADAPTIVE DEFROST	$371	$8.28	$854	402.70	$35	6.72	$1,319	$1,249	$1,150	$4.50	$5.30	$7.07
9	8 + ENHANCED COND HT	$374	$2.82	$860	400.39	$35	6.92	$1,323	$1,253	$1,155	$4.63	$5.46	$7.28

Assumptions:

(1) Energy consumptions for the baseline and for each design option were obtained from an ERA simulation of an actual 15 cubic foot auto-defrost upright freezer.

(2) Manufacturer cost of the baseline unit was interpolated from the AHAM manufacturer cost vs kWh curve for this product class.
Using a linear interpolation between the two closest points on the AHAM curve to the ERA baseline consumption of 694.24 kWh, the ERA baseline cost is $296.22

(3) Electricity cost = 0.088 $/kWh (average cost in 1998 obtained from an interpolation of the 1995 and 2000 prices of electricity forecast in DOE's Annual Energy Outlook 1993, inflated to 1992 dollars). The interpolated value (for 1998) is 0.082 $/kWh (1991$). After adjusting for inflation from 1991 to 1992, it becomes 0.085 $/kWh. The electricity price was then adjusted by an enduse factor for refrigerators of 1.04.

(4) Installation and maintenance costs are not included in the above calculations.

(5) Lifetime =19 years.

(6) Markup factor= 2.3. This represents the average of markup factors for this product class in the 1989 TSD.

(7) Baseline: Compressor EER = 5.05
Insulation thicknesses: freezer door and sides are 3.81 cm (1.50") and 6.35 cm (2.50"). Resistivity of door and walls is 0.286 and 0.521 m2-degC/W-cm.
Evaporator UA product is 21.9 W/C and Condenser area is 2.15sqm.

Attachment 14
3/4

6/14/94

Life Cycle Costs and Payback Periods of 15 cuft Auto Defrost Upright Freezer :
ERA Simulation: Draft Version

Level	Option	Manufr Cost (1992$)	Incr. Mfr Cost (1992$)	Retail Price (1992$)	Annual Energy Use (kWh)	Annual Energy Cost (1992$)	Cumulative Payback (years)	Lifecycle Costs (1992$)			Cumulative CCE (cents/kWh)		
								4%	6%	10%	4%	6%	10%
0	BASELINE	$268.43	-	$617.39	759.24	$67	NA	$1,494.91	$1,362.90	$1,176.28	NA	NA	NA
1	0+ DEC EVAP MOTOR POWER	$274.93	$6.50	$632.34	711.14	$63	3.53	$1,454.26	$1,330.62	$1,155.82	2.37	2.79	3.72
2	1 + 5.60 EER COMPRESSOR	$281.68	$6.75	$647.87	674.62	$59	4.09	$1,427.59	$1,310.29	$1,144.46	2.74	3.23	4.31
3	2+ EVAP ENHANCED HT SURFACE	$284.29	$2.61	$653.87	667.96	$59	4.54	$1,425.89	$1,309.75	$1,145.56	3.04	3.58	4.78
4	3 + ADD 1" INS TO WALLS	$320.13	$35.84	$736.29	580.15	$51	7.54	$1,406.83	$1,305.96	$1,163.35	5.06	5.95	7.94
5	4 + REDUCE GASKET LEAK	$322.06	$1.94	$740.75	575.23	$51	7.62	$1,405.59	$1,305.57	$1,164.18	5.10	6.01	8.01
6	5+ ADD 1" INS TO DOOR	$330.99	$8.93	$761.28	554.36	$49	7.98	$1,402.00	$1,305.61	$1,169.35	5.35	6.29	8.40
7	6 +COND HOT GAS FOR ANTI SWEAT	$351.51	$20.52	$808.46	526.25	$46	9.32	$1,416.70	$1,325.20	$1,195.85	6.24	7.35	9.80
8	7 +ADAPTIVE DEFROST	$359.78	$8.28	$827.50	515.53	$45	9.80	$1,423.34	$1,333.71	$1,206.99	6.56	7.73	10.31
9	8 + INCREASED COND AREA	$367.00	$7.22	$844.10	506.55	$45	10.20	$1,429.56	$1,341.49	$1,216.98	6.83	8.04	10.73

Assumptions:

(1) Energy consumptions for the baseline and for each design option were obtained from an ERA simulation of an actual 15 cubic foot upright auto-defrost freezer.

(2) Manufacturer cost of the baseline unit was interpolated from the AHAM manufacturer cost vs kWh curve for this product class.
Using a linear interpolation between the two closest points on the AHAM curve to the ERA baseline consumption of 759.24 kWh, the ERA baseline cost is $268.43

(3) Electricity cost = 0.088 $/kWh (average cost in 1998 obtained from an interpolation of the 1995 and 2000 prices of electricity forecast in DOE's Annual Energy Outlook 1993, inflated to 1992 dollars). The interpolated value (for 1998) is 0.082 $/kWh (1991$). After adjusting for inflation from 1991 to 1992, it becomes 0.085 $/kWh. The electricity price was then adjusted by an enduse factor for refrigerators of 1.04.

(4) Installation and maintenance costs are not included in the above calculations.

(5) Lifetime =19 years.

(6) Markup factor= 2.3. This represents the average of markup factors for this product class in the 1989 TSD.

(7) Baseline: Compressor EER = 5.27; Evaporator fan motor power = 9 W; Condenser fan motor power = 0 W.
Insulation thicknesses: freezer door and sides are 7.01 cm (2.76") and 4.90 cm (1.93"). Foam resistivity is 0.53 m2-degC/W-cm.
Evaporator and condenser areas: 1.051 sq.m and 1.22 sq.m, respectively.

Life Cycle Costs and Payback Periods of Upright Freezer - Automatic Defrost (AHAM)

Percent below baseline	Mfr. Cost (1992$)	Incr. Mfr. Cost (1992$)	Uncertainty in Incr. Mfr Cost (+/- %)	Retail Price (1992$)	Annual Energy Use (kWh)	Uncertainty in Energy Use (+/- %)	Cumulative Payback (years)	Lifecycle Costs (1992$) 4%	Lifecycle Costs (1992$) 6%	Lifecycle Costs (1992$) 10%	Cost of Conserved Energy (cents/kWh) 4%	Cost of Conserved Energy (cents/kWh) 6%	Cost of Conserved Energy (cents/kWh) 10%	Design Option Feasibility	Marketing Utility (Medium Vol.)
Baseline	$261.46	-	-	$601.35	784.00	-	NA	$1,569.25	$1,412.98	$1,198.04	NA	NA	NA	-	-
5%	$272.44	$10.98	NA	$626.61	744.80	1.0%	7.32	$1,546.12	$1,397.66	$1,193.47	4.59	5.48	7.45	90.0%	100.0%
10%	$287.21	$14.77	NA	$660.59	705.60	1.2%	8.59	$1,531.70	$1,391.05	$1,197.61	5.39	6.42	8.74	87.0%	78.8%
15%	$318.28	$31.06	NA	$732.04	666.40	2.2%	12.63	$1,554.75	$1,421.92	$1,239.22	7.92	9.45	12.85	56.3%	71.3%
20%	$359.73	$41.45	NA	$827.38	627.20	4.4%	16.38	$1,601.70	$1,476.68	$1,304.73	10.28	12.25	16.67	46.0%	57.7%
25%	$376.39	$16.66	NA	$865.70	588.00	5.3%	15.33	$1,591.62	$1,474.42	$1,313.21	9.61	11.46	15.59	27.7%	50.7%
30%	$388.19	$11.80	NA	$892.83	548.80	7.9%	14.08	$1,570.36	$1,460.97	$1,310.52	8.83	10.53	14.33	21.3%	32.5%
35%	* $416.87	$28.68	NA	$958.80	509.60	9.0%	14.80	$1,587.94	$1,486.36	$1,346.65	9.29	11.07	15.06	15.0%	27.0%
40%	* $428.69	$11.82	NA	$986.00	470.40	11.0%	13.94	$1,566.74	$1,472.97	$1,344.01	8.74	10.43	14.18	9.8%	19.0%
45%	* $439.42	$10.73	NA	$1010.67	431.20	13.0%	13.18	$1,543.01	$1,457.06	$1,338.85	8.27	9.86	13.41	9.8%	10.7%
50%	* $502.39	$62.97	NA	$1155.50	392.00	16.0%	16.06	$1,639.45	$1,561.31	$1,453.84	10.08	12.02	16.35	0.8%	5.3%

Assumptions:

(1) Total Volume = 15.7 cu ft (AHAM 1993 average for this product class).

(2) Baseline energy use is for AHAM's 1993 "adjusted model" (includes adjustments for CFC impact and '93 standard).

(3) Baseline retail price is from the 1992-1993 Fall/Winter Annual Sears catalog, page 1580. A price of $560 was listed for a Kenmore upright automatic defrost, 15.0 cu ft freezer. This model does not meet the 1993 standard. The Sears cost was increased by AHAM's cost adjustment of $41.35 for CFC impact and the1993 standard.
Baseline manufacturer cost was calculated from the retail cost, assuming a markup factor of 2.3 (see note 6).

(4) Price of electricity = 8.8 cents/kWh.

(5) Installation and maintenance costs are not included in the above calculations.

(6) Markup factor is assumed to be 2.3. This represents the average of markup factors for this product class in the 1989 TSD.

(7) Lifetime is 21 years.

(8) Uncertainty in energy use is from AHAM's 1998 Energy Option Cumulative Analysis and represents two standard deviations.

(9) Design Option Feasibility (AHAM): 100% = highly feasible; 0% = not feasible.

(10) Marketing Utility at Medium Volume (AHAM): 100% = no negative impact; 0% = not marketable.

*Not all companies submitted data after this point, having exceeded maximum technically feasible designs.

Attachment 15
1/3

11/18/93

Life Cycle Costs and Payback Periods of Upright Freezer - Manual Defrost (AHAM)

Percent below baseline	Mfr. Cost (1992$)	Incr. Mfr. Cost (1992$)	Uncertainty in Incr. Mfr Cost (+/- %)	Retail Price (1992$)	Annual Energy Use (kWh)	Uncertainty in Energy Use (+/- %)	Cumulative Payback (years)	Lifecycle Costs (1992$) 4%	6%	10%	Cost of Conserved Energy (cents/kWh) 4%	6%	10%	Design Option Feasibility	Marketing Utility (Medium Vol.)
Baseline	$167.08	-	-	$384.29	506.00	-	NA	$1,008.98	$908.12	$769.40	NA	NA	NA	-	-
5%	$184.06	$16.98	NA	$423.34	480.70	1.5%	17.54	$1,016.80	$920.98	$789.20	11.00	13.12	17.85	92.5%	100.0%
10%	$193.46	$9.40	NA	$444.97	455.40	2.0%	13.63	$1,007.19	$916.42	$791.57	8.55	10.19	13.87	92.5%	69.0%
15%	$199.94	$6.47	NA	$459.85	430.10	2.3%	11.31	$990.84	$905.11	$787.19	7.10	8.46	11.51	80.0%	52.0%
20%	$209.46	$9.52	NA	$481.75	404.80	4.1%	10.94	$981.50	$900.82	$789.84	6.86	8.19	11.14	65.8%	38.3%
25% *	$219.42	$9.97	NA	$504.68	379.50	7.3%	10.81	$973.19	$897.55	$793.51	6.78	8.09	11.00	47.0%	25.5%
30% *	$228.43	$9.01	NA	$525.39	354.20	8.5%	10.56	$962.68	$892.07	$794.97	6.63	7.90	10.75	32.3%	12.8%
35% *	$262.07	$33.64	NA	$602.76	328.90	11.3%	14.02	$1,008.81	$943.25	$853.08	8.79	10.49	14.26	15.3%	6.3%
40% *	$271.08	$9.01	NA	$623.48	303.60	20.0%	13.43	$998.29	$937.78	$854.54	8.42	10.06	13.66	6.6%	0.0%
45% *	$275.90	$4.82	NA	$634.56	278.30	NA	12.49	$978.14	$922.67	$846.37	7.83	9.34	12.71	2.3%	0.0%
50% *	$330.58	$54.69	NA	$760.34	253.00	NA	16.89	$1,072.69	$1,022.26	$952.90	10.59	12.63	17.19	0.9%	0.0%

Assumptions:

(1) Total Volume = 14.1 cu ft (AHAM 1993 average for this product class).
(2) Baseline energy use is for AHAM's 1993 "adjusted model" (includes adjustments for CFC impact and '93 standard).
(3) Baseline retail price is from the 1992-1993 Fall/Winter Annual Sears catalog, page 1581. A price of $350 was listed for each of two Kenmore upright manual defrost freezers of size 13.3 cu ft and 16 cu ft. Neither model met the 1993 standard. The Sears cost was increased by AHAM's cost adjustment for CFC impact and the 1993 standard of $34.29. Baseline manufacturer cost was calculated from the retail cost, assuming a markup factor of 2.3 (see note 6).
(4) Price of electricity = 8.8 cents/kWh.
(5) Installation and maintenance costs are not included in the above calculations.
(6) Markup factor is assumed to be 2.3. This represents the average of markup factors for this product class in the 1989 TSD.
(7) Lifetime is 21 years.
(8) Uncertainty in energy use is from AHAM's 1993 Energy Option Cumulative Analysis and represents two standard deviations.
(9) Design Option Feasibility (AHAM): 100% = highly feasible; 0% = not feasible.
(10) Marketing Utility at Medium Volume (AHAM): 100% = no negative impact; 0% = not marketable.
*Not all companies submitted data after this point, having exceeded maximum technically feasible designs.

Life Cycle Costs and Payback Periods of Chest Freezer - Manual Defrost (AHAM)

Percent below baseline	Mfr. Cost (1992$)	Incr. Mfr. Cost (1992$)	Uncertainty in Incr. Mfr Cost (+/- %)	Retail Price (1992$)	Annual Energy Use (kWh)	Uncertainty in Energy Use (+/- %)	Cumulative Payback (years)	Lifecycle Costs (1992$) 4%	Lifecycle Costs (1992$) 6%	Lifecycle Costs (1992$) 10%	Cost of Conserved Energy (cents/kWh) 4%	Cost of Conserved Energy (cents/kWh) 6%	Cost of Conserved Energy (cents/kWh) 10%	Design Option Feasibility	Marketing Utility (Medium Vol)
Baseline	$174.53	-	-	$401.43	465.00	-	NA	$975.50	$882.82	$755.33	NA	NA	NA	-	-
5%	$188.77	$14.23	NA	$434.16	441.75	1.5%	16.00	$979.53	$891.48	$770.37	10.03	11.97	16.28	100.0%	80.0%
10%	$200.47	$11.71	NA	$461.09	418.50	2.0%	14.58	$977.75	$894.33	$779.60	9.14	10.91	14.83	87.5%	58.0%
15%	$205.21	$4.74	NA	$471.99	395.25	2.3%	11.49	$959.95	$881.16	$772.80	7.21	8.60	11.70	62.8%	47.8%
20%	$221.76	$16.55	NA	$510.06	372.00	3.0%	13.27	$969.32	$895.17	$793.18	8.33	9.93	13.51	52.0%	40.8%
25%	$232.90	$11.13	NA	$535.66	348.75	6.3%	13.12	$966.22	$896.70	$801.09	8.23	9.82	13.35	35.8%	35.3%
30%	* $246.41	$13.52	NA	$566.75	325.50	8.1%	13.47	$968.60	$908.72	$814.49	8.45	10.07	13.70	12.0%	0.3%
35%	* $322.32	$75.91	NA	$741.34	302.25	9.4%	23.73	$1,114.48	$1,054.24	$971.37	14.89	17.75	24.15	7.5%	0.0%
40%	* $340.45	$18.13	NA	$783.02	279.00	11.0%	23.31	$1,127.47	$1,071.86	$995.37	14.62	17.44	23.72	5.0%	0.0%
45%	* $429.66	$89.21	NA	$988.21	255.75	14.0%	31.87	$1,303.95	$1,252.98	$1,182.86	19.99	23.84	32.42	1.5%	0.0%
50%	* $499.17	$69.51	NA	$1148.08	232.50	16.0%	36.49	$1,435.12	$1,388.77	$1,325.03	22.89	27.30	37.13	0.0%	0.0%

Assumptions:

(1) Total Volume = 16.0 cu ft (AHAM 1993 average for this product class).

(2) Baseline energy use is for AHAM's 1993 "adjusted model" (includes adjustments for CFC impact and '93 standard).

(3) Baseline retail price is from the 1992-1993 Fall/Winter Annual Sears catalog, page 1583. A price of $370 was listed for a Kenmore chest manual defrost, 15.8 cu ft freezer. This model does not meet the 1993 standard. The Sears cost was increased by AHAM's cost adjustment of $31.43 for CFC impact and the 1993 standard. Baseline manufacturer cost was calculated from the retail cost, assuming a markup factor of 2.3 (see note 6).

(4) Price of electricity = 8.8 cents/kWh.

(5) Installation and maintenance costs are not included in the above calculations.

(6) Markup factor is assumed to be 2.3. This represents the average of markup factors for this product class in the 1989 TSD.

(7) Lifetime is 21 years.

(8) Uncertainty in energy use is from AHAM's 1998 Energy Option Cumulative Analysis and represents two standard deviations.

(9) Design Option Feasibility (AHAM): 100% = highly feasible; 0% = not feasible.

(10) Marketing Utility at Medium Volume (AHAM): 100% = no negative impact; 0% = not marketable.

*Not all companies submitted data after this point, having exceeded maximum technically feasible designs.

Attachment 15
3/3

1998 NAECA Negotiation Summary

Automatic Defrost Refrigerator/Freezers

	Original 1993 NAECA			1998 NAECA/RSG Final		HCFC	HCFC-free/RSG Final	
	Slope	y-inter	% below 1993	Slope	y-inter	Phaseout	Slope	y-inter
Top Frz Auto	16.00	355.0	29.6%	9.80	276.0	-10.0%	10.78	303.6
Top Frz Disp	17.60	391.0	26.0%	10.20	356.0	-10.0%	11.22	391.6
Side Frz Auto	11.80	501.0	22.0%	4.91	507.5	-10.0%	5.40	558.3
Side Frz Disp	16.30	527.0	29.3%	10.10	406.0	-10.0%	11.11	446.6
Bottom Frz Auto	16.50	367.0	26.0%	4.60	459.0	-10.0%	5.06	504.9

Compact Refrigerators (7.75 cu ft or smaller)*

	Original 1993 NAECA			1998 NAECA/RSG Final		HCFC	HCFC-free/RSG Final	
	Slope	y-inter	% below 1993	Slope	y-inter	Phaseout	Slope	y-inter
Manual Defr R/F	13.50	299.0	5.0%	10.70	299.0	-10.0%	11.77	328.9
Partial Defr R/F	10.40	398.0	5.0%	7.00	398.0	-10.0%	7.70	437.8
Top Frz Auto	16.00	355.0	5.0%	12.70	355.0	-10.0%	13.97	390.5
Side Frz Auto	11.80	501.0	5.0%	7.60	501.0	-10.0%	8.36	551.1
Bottom Frz Auto	16.50	367.0	5.0%	13.10	367.0	-10.0%	14.41	403.7
Upright Frz Auto	14.90	391.0	5.0%	11.40	391.0	-10.0%	12.54	430.1
Upright Frz Man	10.30	264.0	5.0%	9.78	250.8	-10.0%	10.76	275.9
Chest Frz Manual	11.00	160.0	5.0%	10.45	152.0	-10.0%	11.50	167.2

Freezer Products (full-sized, larger than 7.75 cu ft)

	Original 1993 NAECA			1998 NAECA/RSG Final		HCFC	HCFC-free/RSG Final	
	Slope	y-inter	% below 1993	Slope	y-inter	Phaseout	Slope	y-inter
Upright Frz Auto	14.90	391.0	16.6%	12.43	326.1	-10.0%	13.67	358.7
Upright Frz Man	10.30	264.0	14.2%	7.55	258.3	-10.0%	8.31	284.1
Chest Frz Man	11.00	160.0	10.2%	9.88	143.7	-10.0%	10.87	158.1

Manual/Partial Defrost - Free Standing Products (Higher than 36")

	Original 1993 NAECA			1998 NAECA/RSG Final		HCFC	HCFC-free/RSG Final	
	Slope	y-inter	% below 1993	Slope	y-inter	Phaseout	Slope	y-inter
Manual Defr R/F	13.50	299.0	24.4%	8.82	248.4	-10.0%	9.70	273.2
Partial Defr R/F	10.40	398.0	30.9%	8.82	248.4	-10.0%	9.70	273.2

***Compact Refrigerators or Freezers are defined as being LESS THAN 7.75 cu ft AHAM (FTC reported) volume AND LESS THAN 36" in HEIGHT.**

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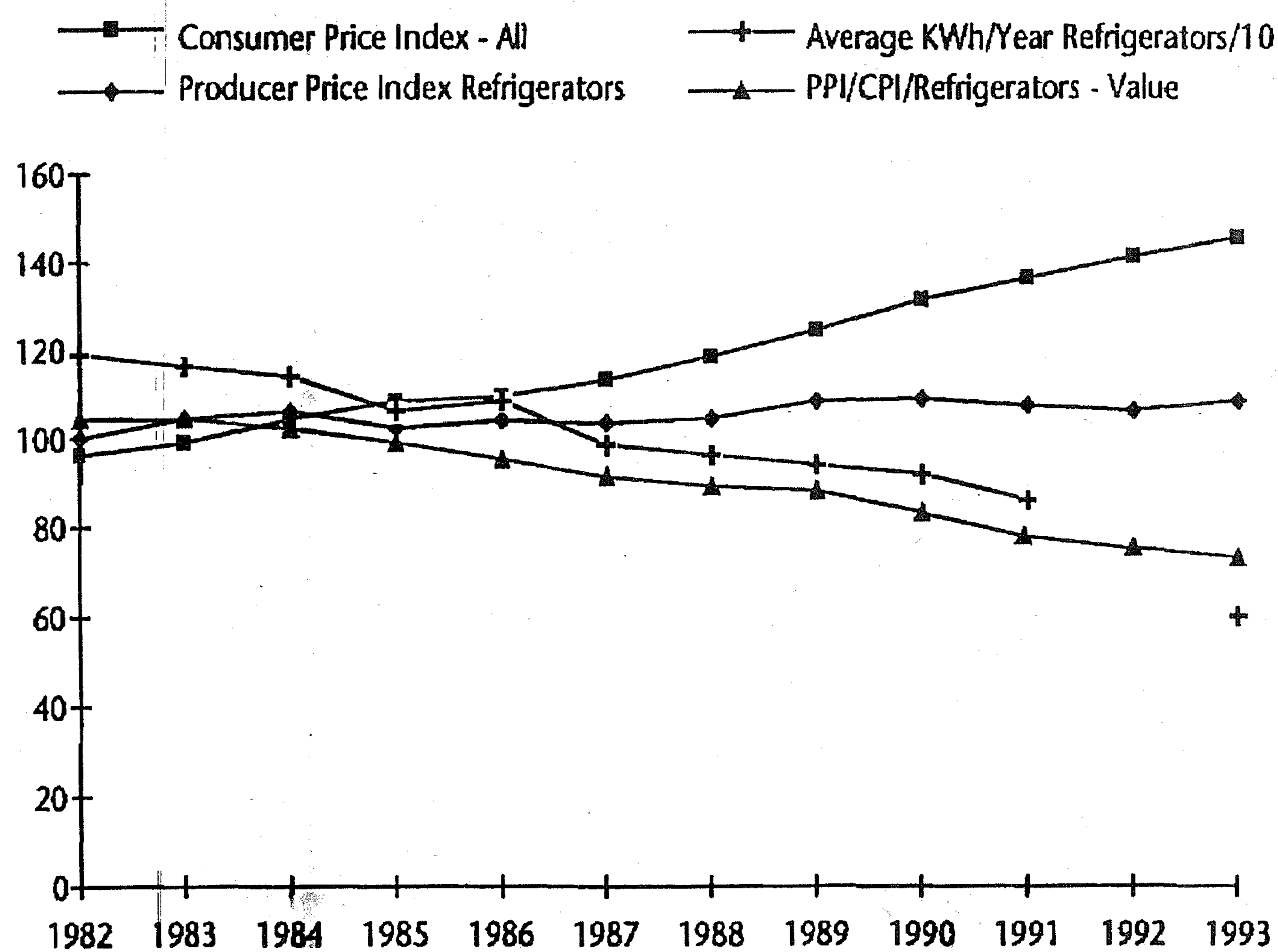
Attachment 16

Figure 1

Change in Refrigerators for Ex-Manufacturer Price Versus Consumer Prices Index

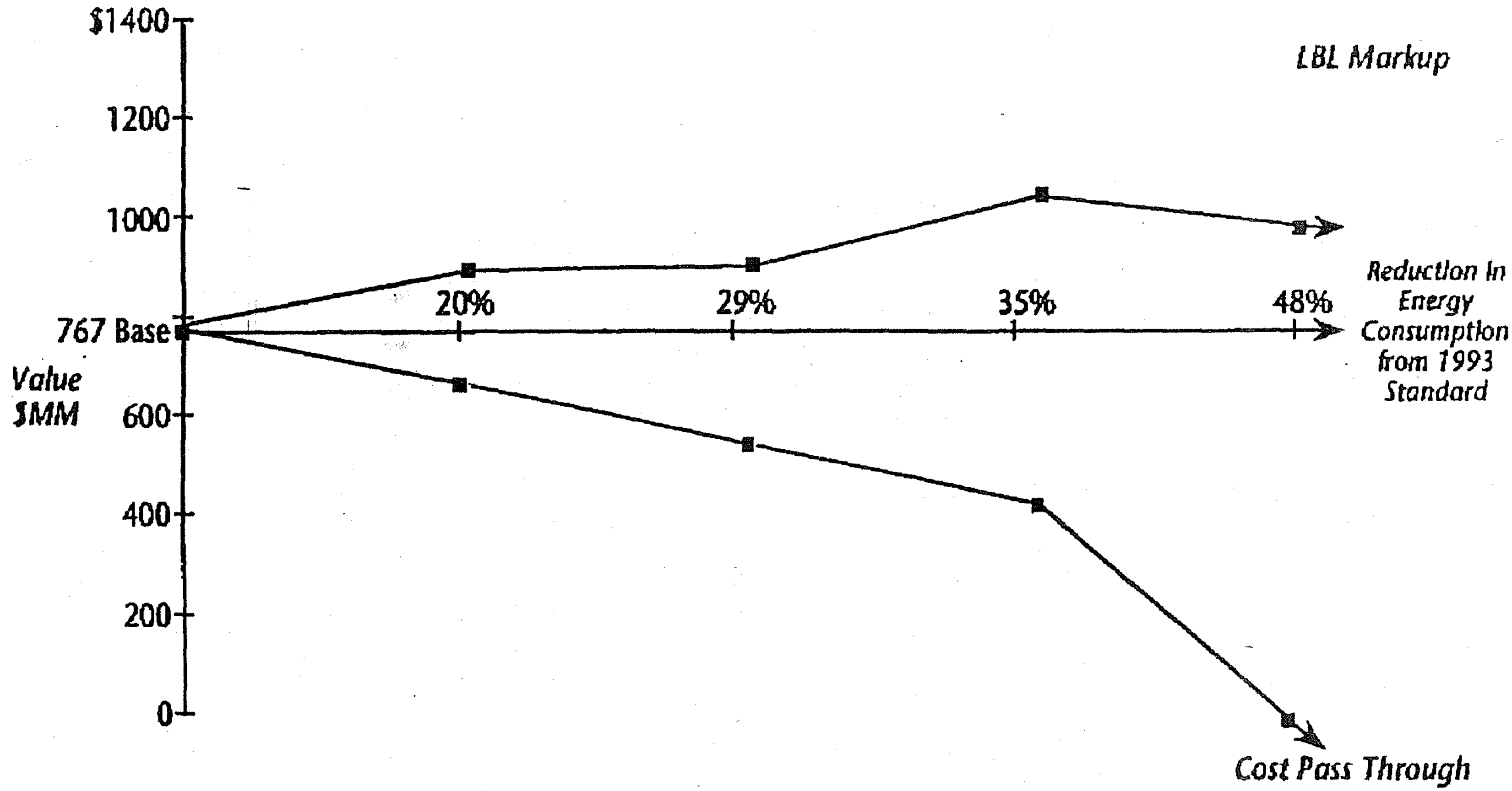


Attachment 17

Figure 2

Refrigerator Industry

Value at Various Standard Levels - Automatic Defrost Refrigerators



Note: "LBL markup" represents a 30% markup from manufacturer cost to manufacturer price and is derived from LBL published markup from manufacturer costs to retail price

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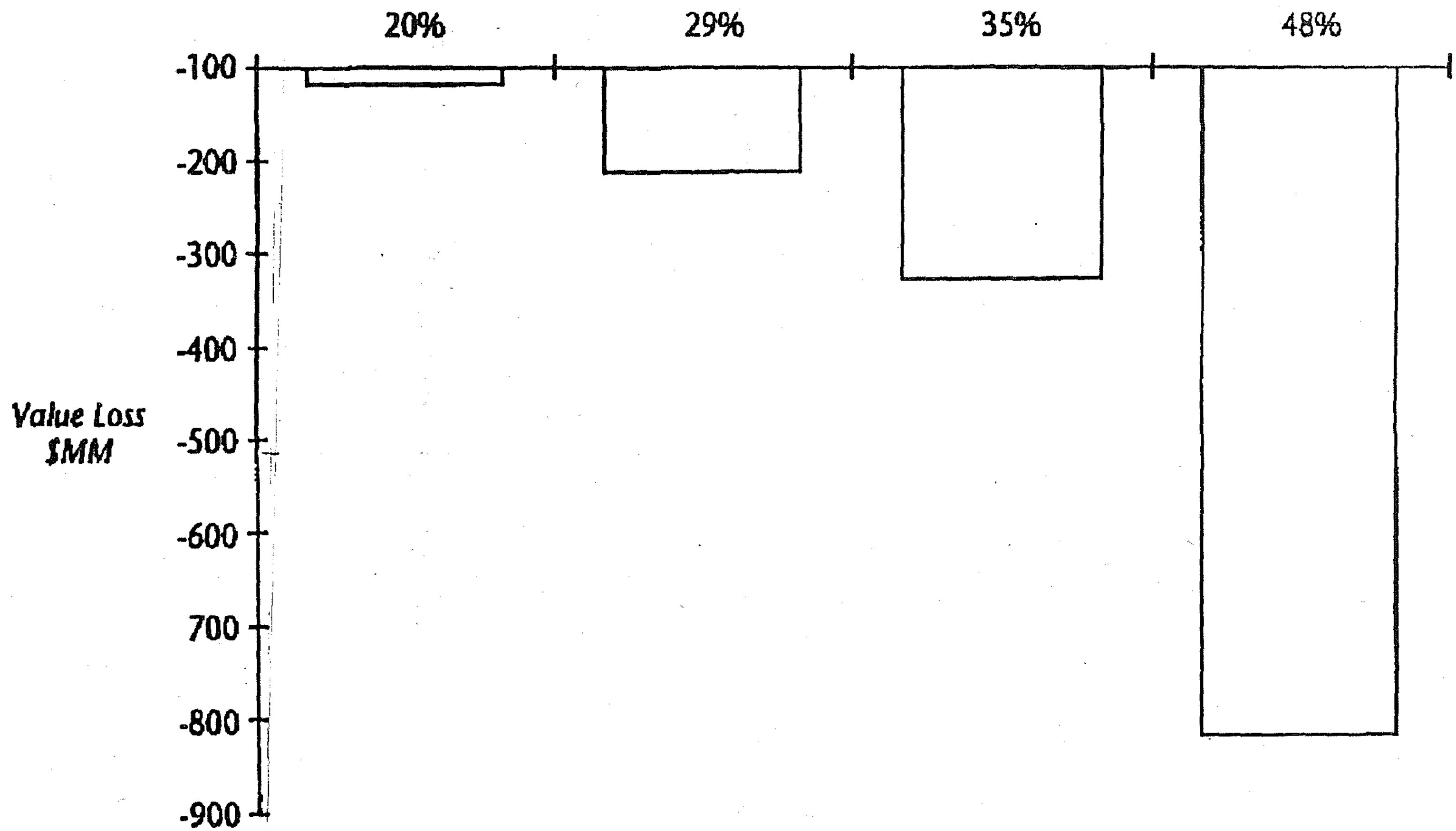
Attachment 18

Figure 3

Refrigerator Industry

Value Loss at Various Standard Levels - Automatic Defrost Refrigerators

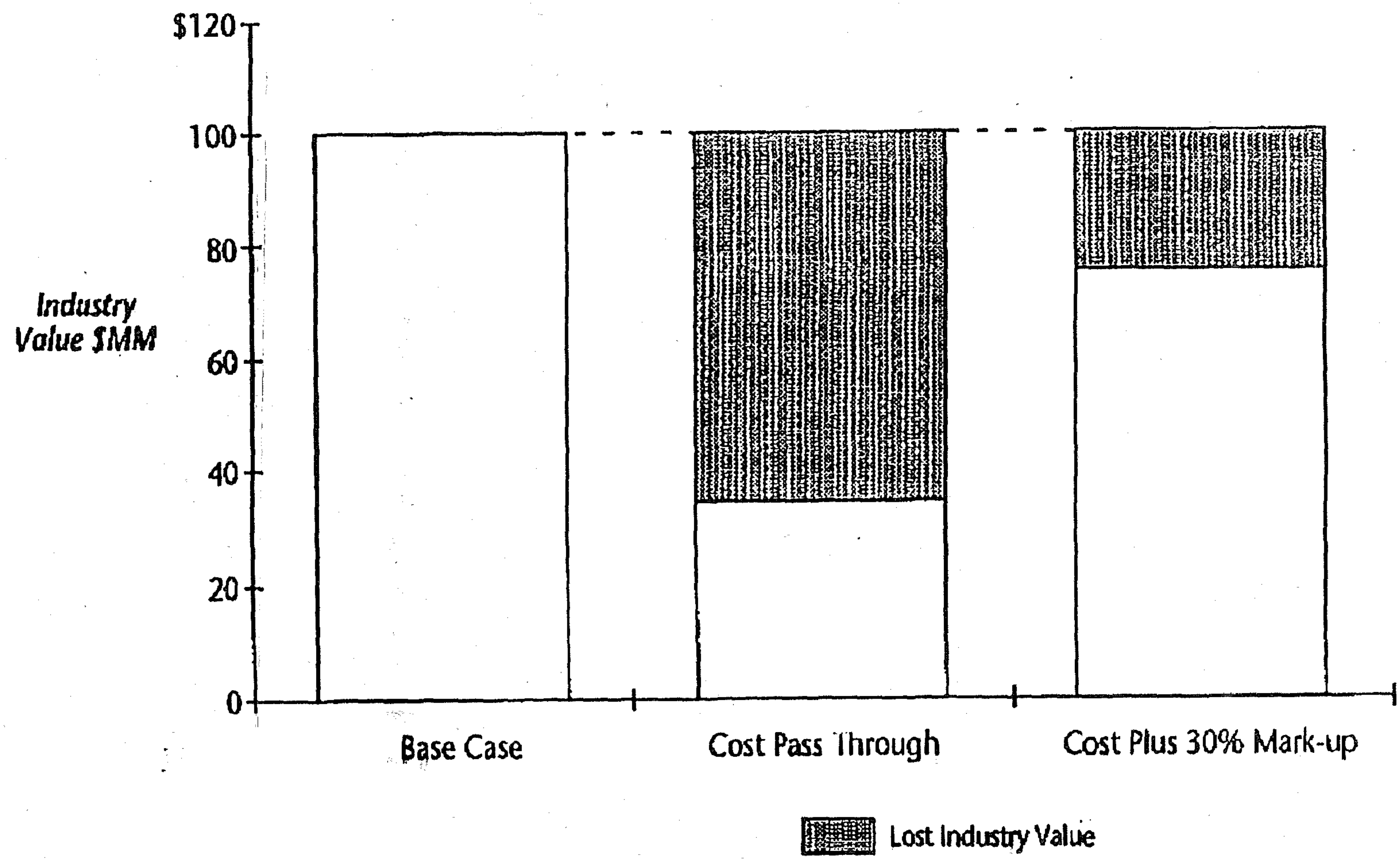


Attachment 19

Figure 4

Freezer Industry

Value at Proposed Standard Level



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Energy Impacts of 1998 NAECA Negotiations

	Annual Prod*	AV**	1993	RSG 1998	Annual Watt/hr savings
Top Frz Auto	4,506,765	21.7	702.20	488.66	962,374,598,100
Top Frz Disp	75,772	25.4	838.04	615.08	16,894,125,120
Side Frz Auto	399,291	26.3	811.34	636.63	69,758,932,737
Side Frz Disp	984,793	25.4	941.02	662.54	274,245,154,640
Bottom Frz Auto	93,213	24.8	776.20	573.08	18,933,424,560
Single Door R/F	160,000	16.2	542.09	467.14	11,992,640,000
Compact R/F	812,012	7.0	433.27	411.61	17,591,171,543
Upright Frz Auto	155,798	26.0	778.40	649.28	20,116,637,760
Upright Frz Man	471,256	24.4	515.32	442.52	34,307,436,800
Chest Frz Manual	803,621	26.0	446.00	400.58	36,500,465,820

Total Watt hour Savings (@ outlet) = 1,462,714,587,080

Total Energy Consumption of ALL Ref/Frz and Frz (at utility):
2.017 QUADS

Convert to Btu's - 16 years - 30.8% Utility Efficiency
259,261,411,486,623 BTU's

FIVE AUTO-DEFROST Refrigerator/Freezer Product Classes

Total Energy Consumption of Auto-defrost :
1.278 QUADS (at utility)

Total Energy Consumption if ALL Auto-defrost @ 1993:
0.755 QUADS (at utility)

Total Energy Consumption if ALL Auto-defrost @ RSG 1998:
0.532 QUADS (at utility)

RSG 1998 Savings = .223 Quads

SINGLE DOOR REFRIGERATOR/FREEZERS

Total Energy Consumption of Single door Ref/Frz :
0.151 QUADS (at utility)

Total Energy Consumption of Single Door Ref/Frz if ALL @ 1993:
0.088 QUADS (at utility)

Total Energy Consumption of Single Door Ref/Frz if ALL @ RSG 1998:
0.084 QUADS (at utility)

RSG 1998 Savings = .004 Quads

COMPACT/UNDERCOUNTER REFRIGERATOR/FREEZERS

Total Energy Consumption of Compact Ref/Frz :
0.052 QUADS (at utility)

Total Energy Consumption of Compact Ref/Frz if ALL @ 1993:
0.042 QUADS (at utility)

Total Energy Consumption of Compact Ref/Frz if ALL @ RSG 1998:
0.040 QUADS (at utility)

RSG 1998 Savings = .002 Quads

THREE FREEZER PRODUCT CLASSES (8 cu ft +)

Total Energy Consumption of Freezers :
0.535 QUADS (at utility)

Total Energy Consumption of Freezers if ALL @ 1993:
0.260 QUADS (at utility)

Total Energy Consumption of Freezers if ALL @ AHAM:
0.230 QUADS (at utility)

RSG Savings = .030 Quads

TOTAL RSG 1998 Savings = .259 Quads

** Figures are from the 1991 and 1992 AHAM Energy Audits.*
*** Adjusted Volumes used for Standard Estimates*

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