Open-Ended Problem: The Design of a CFC-Free, Energy-Efficient Refrigerator

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PROBLEM INTRODUCTION AND SUGGESTED USE

The discovery that chlorofluorocarbons (CFCs), widely used as refrigerants, are causing ozone depletion, and the increasing energy efficiency awareness brought about in part by the Green House Effect (caused mainly by the burning of fossil fuels) necessitates a significantly different approach to the design of a refrigerator. This case study will address these important and current topics as well as the more traditional topics involved in the design of a refrigeration system. The following is an outline of the three sections in this case study.

SECTION I: Problem Statement

The recently announced "Golden Carrot Program" to develop a super-efficient refrigerator will be used as the problem statement. The Super Efficient Refrigerator Program, Inc. (SERP), a consortium of over thirteen electric utilities, has announced a bonus of $30 million to be awarded to the manufacturer that develops and markets by 1995, a CFC-free refrigerator that is 25-50% more efficient than the 1993 federal efficiency standard. The bonus would be paid per refrigerator sold, at $50 per 100 KWh/year energy savings over the 1993 federal standard. A benefit of using this context to pose the design problem is the introduction of the concept of Demand-Side Management (DSM).

The problem is posed in the form of two memos. The first introduces the regulatory pressure driving the conversion to the CFC-free energy efficient refrigerator. Manufacturer-supplied information in **Appendix A** may be used to supplement this memo. The second memo provides design specifications which can be used by the students to begin work on the problem. Material from **Appendix B** may be given to the students at the discretion of the instructor.

SECTION II: Teaching Aids

These are summaries of topics which are either directly related to the design project or beyond the scope of this design but also very important in that they provide either a background for the design or a future direction. These may be given before the design project is started or as it progresses. The following is a brief description of these teaching aids:

List of Acronyms

ARI Air Conditioning and Refrigeration Institute
ASHRAE American Society of Heating, Refrigerating, and Air Conditioning Engineers
AV Adjusted Volume
CFC Chlorofluorocarbon
COP Coefficient of Performance
DSM Demand-Side Management
EER Energy Efficiency Rating
GWP Global Warming Potential
HCFC Hydrochlorofluorocarbon
HFC Hydrofluorocarbon
ODP Ozone-Depleting Potential
PAG Polyalkylene glycols
RE Refrigeration Effect
SERP Super-Efficient Refrigerator Program

A BRIEF HISTORY OF REFRIGERATION

A brief survey of refrigeration technology from snow use in ancient times to the current phaseout of CFCs.

REFRIGERATOR FEATURES

A Consumer Perspective: a survey of design features that consumers are usually most concerned with: reliability, appearance, noise, access, smells, initial cost, operating cost, defrost system, temperature uniformity, safety, environmental impact, etc.

THE OZONE DEPLETION PROBLEM

The problem and how it was discovered. Mechanism of ozone depletion by CFCs. Ozone Depletion Potential (ODP) of current and prospective refrigerants.

GLOBAL WARMING AND THE REFRIGERATOR

The Green House Effect and gases contributing to it. Global Warming Potential (GWP) of current and prospective refrigerants. Energy consumption and its relation to the green house gases.

CHOOSING A REFRIGERANT

Criteria that help determine which refrigerant may be used: safety, thermodynamic efficiency, compatibility with compressor oil and material of construction, ODP, GWP, cost, availability, etc. Some potentially good choices for a refrigerant.

FACTORS WHICH INFLUENCE THE ENERGY CONSUMPTION OF A REFRIGERATOR

Insulation and gaskets, CFC-free insulation, refrigeration cycle used, electrical components.

LIFE CYCLE DESIGN

Introduction to the concept of life cycle analysis as taking a literally global perspective when evaluating a production process. The material in **Appendix C**, which gives more details on the subject, may be given to the students.

SECTION III: Design Problem Solution

This section places more emphasis on comparing different alternatives and showing their advantages and disadvantages rather than concentrating on a single design. An attempt was made to keep the focus on energy efficiency and CFC replacement as it affects energy efficiency. Graphs which can be easily used to consider a multitude of options are included in this section. Hopefully they will give the instructor and the student insights into the interplay of the environmental and design issues.

Section I:
Problem Statement

Frigid Whirl Corporation

Ann Arbor, Michigan

INTEROFFICE MEMO

DATE: December 2, 1992

TO: J. Jones
          Senior Research Engineer

FROM: S. Naser
          Manager, Project Development

RE: Design of a new energy efficient CFC-free refrigerator.

As you know, the mounting evidence that Chlorofluorocarbons (CFCs) are causing Stratospheric ozone depletion has increased the pressure to discontinue their use. This drive to phase out CFCs is consistent with both our corporate policy - we have to take care of our planet - and external demands from regulatory and public interest groups. As of July 1992, venting of CFCs to the atmosphere is illegal.[1] Pressure is also increasing to not even consider what we had counted on as short term substitutes (because of their lower ozone depletion potential) to CFCs: Hydrochlorofluorocarbons (HCFCs). I have enclosed some information furnished to us by two of our suppliers, Du Pont and ICI Americas, on the regulations regarding CFCs and HCFCs.

The recent Copenhagen meeting held in November 1992 to revise the Montreal protocol has accelerated the phaseout of CFCs and brought the schedule for this change more in accord with the deadline President Bush announced in February of 1992 for the U.S. CFC production must drop to 25% of 1986 levels by January 1994 and is to be stopped completely by January 1996. According to the Air-Conditioning and Refrigeration Institute (ARI), current world consumption is already below 50% of 1986 levels.[2] As for HCFCs, they are to be completely phased out by the year 2030. Based on these developments and our emerging pollution prevention policy, management has decided that the domestic refrigerator that we manufacture is not to use or contain any CFCs or HCFCs.

I would like you and your team to investigate alternative refrigerants for use in our product. Management is considering participating in an incentive program which takes the form of a competition for a $30 million prize. The goal is to design and build the most energy efficient, CFC-free domestic refrigerator on the market.[3] I will provide you with more detailed information as soon as a decision is made.

Frigid Whirl Corporation

Ann Arbor, Michigan

INTEROFFICE MEMO

DATE: December 9, 1992
TO: J. Jones
         Senior Research Engineer
FROM: S. Naser
         Manager, Project Development
RE: Further information regarding the design of a new energy efficient CFC-free refrigerator.

Management has approved our participation in the "Golden Carrot Program," a competition organized by the Super-Efficient Refrigerator Program, Inc (SERP). SERP is a consortium of several utility companies, including Pacific Gas and Electric Co., Southern California Edison Co., Long Island Lighting Co., Bonneville Power Administration, PacifiCorp, Sacramento Municipal Utility District, the Los Angeles Department of Water and Power, Jersey Central Power and Lighting Co., New England Electric Service, and the Wisconsin Center for Demand-Side-Management Research (itself a group of Wisconsin utilities). These utilities have already pledged $30 million as part of an effort to reduce energy consumption, a practice called Demand-Side-Management. This particular initiative is designed to encourage the development and immediate production of an energy super-efficient, CFC-free domestic refrigerator.

Under this program, the winner of the competition will receive a rebate from SERP of $50 per 100 KWh/year energy savings over the 1993 federal standard[4] (to be detailed below) per refrigerator sold. The company will be required to initially supply at least 100,000 units, to be allocated to the regions of the participating utilities in proportion to their contribution to SERP. Units could be sold in other areas, but rebates would not apply to those sales. The units must have automatic defrost and have a capacity of between 17.5 and 22.4 ft^3.[5] The manufacturer must have a prototype ready by April 1993 and, if awarded the contract, start shipping units as early as 1994.[6]

I would like you to redesign our 18 ft^3 top-of-the-line model GF-222 refrigerator, equipped with a top-mounted freezer which currently uses CFC-12. Since Marketing reports that this model is selling well, we want to retain its following features:

Volume
   Freezer: 5.0 ft^3
   Fresh Food Section: 13.0 ft^3

Normal Operation Design Conditions:
   Freezer Temperature: 5 °F
   Fresh Food Temperature: 38 °F
   Ambient Temperature: 90 °F
   Pull Down Time: 2 minutes

Pull Down Time is defined as the time it takes to cool the air inside of the refrigerator from ambient to design conditions. The estimate given here is based on literature values for an empty cabinet.

The refrigerator must be able to operate satisfactorily at the following extreme conditions:
   Freezer Temperature: 0 °F
   Fresh Food Temperature: 37 °F
   Ambient Temperature: 110 °F

These extreme conditions are often used by consumer groups[7] to rate refrigerators from different manufacturers.

The federal standard referred to above is the Department of Energy (DOE) level 4 energy efficiency standard[4] which beginning in 1993 requires new refrigerators with automatic defrost to have the following maximum energy consumption:

   KWh/year = 329 + 11.8 x AV

where the Adjusted Volume (AV) is defined as:

   AV = volume of fresh food compartment + 1.63 x volume of freezer

These standards are based on DOE simulations[8] for refrigerators which utilize better insulation and more efficient compressors than are conventionally used.

I have attached some technical data from our labs and some materials furnished by our supplier which I believe will be of use to you. You should determine what changes we need to make in our refrigerator to both meet the specifications above without using CFCs or HCFCs, and produce a refrigerator with an energy efficiency that can make our company competitive in the "Golden Carrot" contest and still be commercially viable.

TECHNICAL DATA SHEET

Nomenclature

• h_i = internal heat transfer coefficient in Btu/hr-ft²-°F

• U = overall heat transfer coefficient in Btu/hr-ft²-°F

• m = mass flow rate of refrigerant in lb/hr

• C_p = heat capacity of refrigerant in Btu/lb-°F

• μ = viscosity of refrigerant in lb/ft-hr

• k = thermal conductivity of refrigerant in Btu/ft-hr-°F

Compressor

• clearance space = 5% of total volume

• compressor isentropic efficiency = 70%

• compressor motor efficiency = 80%

• volumetric efficiency (excluding losses due to clearance volume) = 90%

• speed = 60 revolutions/second

Evaporator

• $h_i = 515.2 \, (C_p/\mu)^{0.4} \, k^{0.6} \, m$

• $U = 1/((0.98/h_i) + 0.092)$

• fan power = 10 watt

CONDENSER

• $h_i = 592.5 \, (C_p/\mu)^{0.4} \, k^{0.6} \, m$

• $U = 1/((0.86/h_i) + 0.079)$

• fan power = 14 watt

DESUPERHEAT PART OF CONDENSER

• $h_i = 10.3 \, (C_p/\mu)^{0.4} \, k^{0.6} \, m$

• $U = 1/((0.86/h_i) + 0.079)$

Interchanger

• U = 40

• 14 °F superheating

DATA ON CURRENT REFRIGERATOR

• uses R-12 as refrigerant

• fresh food section insulation is 1.5 inches of R-11 blown urethane foam

• freezer insulation is 1.85 inches of R-11 blown urethane foam

• depth: 22.1 inches, width: 26.5 inches, height: 53 inches

Anti-sweat Heater

• on 30% of the time

• power = 19 watt

Cabinet Heat Gains

• internal heat transfer coefficient = 1.0 Btu/hr-ft²-°F based on inside area

• external heat transfer coefficient = 1.47 Btu/hr-ft²-°F based on outside area

• freezer gasket heat transfer coefficient = 0.0055 Btu/hr-in-°F

• fresh food section gasket heat transfer coefficient = 0.0014 Btu/hr-in-°F

• gasket heat transfer coefficents are based on gasket length

• ignore corner effects

• freezer and fresh food section are separated by 3 inches of insulation

• assume no heat transfer occurs across the insulation

Pressure Drops

• ignore pressure drops due to flow

• assume isenthalpic expansion in the expansion valve

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Section II:
Teaching Aids

A BRIEF HISTORY OF REFRIGERATION

People discovered the value of cold environments in preserving foods very early in history. The inhabitants of Crete were aware of this in 2000 B.C. It is said that Alexander the Great had his soldiers served snow-cooled drinks in the hot summers of Petra from the winter snow stored in trenches covered with branches. Many ancient civilizations where snow was not abundant used clay pottery, which allows some stored water to permeate the surface and evaporate, cooling the water inside. In areas where snow was available, it was stored for the warmer seasons or transported to warmer areas where it could be of more value. An ice transport industry flourished in the nineteenth century, taking natural ice from North America to the West Indies, Europe, India, and Australia. Ice from Norway was also transported to the warmer southern parts of Europe. The use of natural ice or snow was complemented by the addition of various salts known since antiquity (table salt, salt peter, etc.) which lower the freezing temperature of water to a degree that depends on the salt and its concentration. This practice combined with various natural insulating materials allowed temperatures below 32 °F (0 °C) to be achieved and maintained.

The birth of mechanical refrigeration occurred in the middle of the eighteenth century when W. Cullen demonstrated the making of ice by the evaporation of ethyl ether when its pressure was mechanically reduced. In 1810, Sir John Lesley used the first sulfuric acid-water pair absorption refrigeration cycle to produce ice. In 1834, J. Perkins patented the first vapor compression machine. He described his invention: "What I claim is an arrangement whereby I am enabled to use volatile fluids for the purpose of producing the cooling or freezing of fluids, and yet at the same time

constantly condensing such volatile fluids, and bringing them again and again into operation without waste." It does not seem that Perkins followed through on his invention.

Next came the air cycle (expansion and compression only with no evaporation and condensation) refrigeration machine, invented by J. Gorrie in 1845. Since the refrigerant used was air, it did not need to be in a closed cycle; thus the cold air was injected into the enclosure to be cooled. By 1860 F. Carre was selling 500-pound-a-day ammonia water absorption refrigeration machines. Interestingly, the source of energy for these machines was not mechanical but heat from firewood, coal, or gas. The refrigerant, ammonia, goes through the same steps it would in a vapor compression cycle except that there is no compressor. The ammonia vapor is condensed under elevated pressure in a condenser which rejects heat to the atmosphere, and then the liquefied ammonia is expanded through a capillary tube into the evaporator where it absorbs heat, producing the refrigeration effect and becoming a gas. The ammonia gas is then mixed with water where it is absorbed and rejects its heat of solution. The ammonia-rich solution is pumped to the generator, where a high temperature heat input causes the ammonia to desorb under elevated pressure. This high pressure ammonia gas goes to the condenser and repeats the cycle, while the weak ammonia solution is returned to the absorber and repeats its cycle.

The first commercial application of the vapor compression cycle occurred in the middle of the nineteenth century and was pioneered by A. Twining and J. Harrison, who used ethyl ether as the refrigerant. Since then, various refrigerants were used: carbon dioxide (T. Lowe, 1866), ammonia (D. Boyle, 1872), and

sulfur dioxide (R. Pictet, 1874). By the end of the nineteenth century electricity was being used to power refrigeration machinery. The first automatic household refrigerator, using sulfur dioxide, debuted in 1918. By 1930 chlorofluorocarbons (CFCs) had been proposed and demonstrated. Since then, CFCs have replaced all other refrigerants in vapor compression cycle-based machines.

REFRIGERATOR FEATURES: A CONSUMER PERSPECTIVE

Refrigerator Types

The most popular type of refrigerator has the freezer occupying the top section (top mounted freezer). This type of refrigerator is available with the widest selection of capacities, styles, and features. It generally costs less to operate than other types of refrigerators with similar features and capacities. Its claimed capacity comes closest to matching its actual capacity. The eye-level freezer makes things in it easy to reach while making vegetables and other items normally stored in the bottom shelves harder to reach. This type of refrigerator has wide shelves which makes things easy to reach in general.

The second most popular refrigerator is the side-by-side type, in which the freezer is located vertically, beside and along the fresh food compartment. It is slightly more expensive to run than the top freezer models. Its freezer is larger than comparable top or bottom freezer models. Easy access is distributed between the freezer and fresh food section. Shelves are narrower, however, making things harder to reach.

The least popular refrigerator type has the freezer occupying the bottom part of the refrigerator. Bottom freezer models of this type are not available in as wide a selection and features as the other two models. This type of refrigerator is most likely more expensive to buy and to operate. The fresh food section is easier to reach while the freezer is not. A necessary pull-out basket in the freezer reduces its capacity.

A Refrigerator's Important Features

TEMPERATURE BALANCE

A refrigerator should be able to maintain a freezer temperature of anywhere between 0-5 °F and a fresh

food compartment temperature of 37 to 38 °F.

TEMPERATURE UNIFORMITY

Unless a refrigerator can maintain a uniform temperature in its interior, some foods will freeze when they are not supposed to and others will melt. *Consumer Reports* tests[1] indicate that top freezer type models are best in this regard.

TEMPERATURE COMPENSATION

A refrigerator should have enough reserve capacity to cope with unexpected adverse conditions such as a summer heat wave or a not completely closed door.

ICE MAKING

The freezer should be able to make ice within a reasonable time. Consumer Reports rates 4 hours as excellent and 8 hours as excessively long.

CONDENSATION CONTROL

Most refrigerators have an anti-sweat heater which is an electric heating strip that prevents condensation around the doors (especially between freezer and fresh food doors) in humid weather. An "energy saver" switch can be used to turn off this heater in dry weather.

OPERATING COST

The refrigerator consumes a large portion of a household's electricity. Top freezer models cost about eighty to ninety dollars per year to operate (using the National Average Electricity Rate). Side-by-side and bottom freezer types cost a few dollars more to run. The black on yellow sticker required on all refrigerators, which is the Department of Energy's "Energy Guide," can be used by consumers to compare different refrigerator models from different manufacturers as to their energy efficiency and consequently their operating cost.

AN EPA SURVEY

Following are some of the results of a survey taken from an Environmental Protection Agency (EPA) study[2] designed to assess consumers' attitudes towards tradeoffs involved in more energy efficient refrigerators:

"What is the importance of the following features in your decision to purchase a refrigerator?" (10 = "Very important", 0 = "Not important")

Feature	Response
initial cost	9.2
better seals	9.1
storage space / interior volume	9.0
width of available kitchen space	8.8
energy efficiency	8.7
mobility	8.7
interior volume	8.6
movable shelves	8.5
operating cost	8.4
height of available kitchen space	8.4
depth of available kitchen space	8.4
freezer location (top, bottom, or side)	8.4
doesn't break easily	8.4
sturdier doors	8.2
easier to clean underneath	8.1
easier to clean seals	8.0
deep door shelves	8.0
type of shelves	7.8
kick plate is secure	7.7
environmental impact	7.4
freezer room	6.9
larger crispers	6.7
makes more ice	5.4
controls odors	5.4
changeable color panels	4.4
ice/water service	4.2
bottom freezer	3.0
makes different shaped ice	1.9

THE OZONE DEPLETION PROBLEM

Today's computer images of the ozone hole over Antarctica drive home the reality of the problem which was suspected, in theory, twenty years ago. In 1974 two chemists, Dr. F. Sherwood Rowland and Dr. Mario Molina, theorized that chlorofluorocarbons (CFCs) could be destroying the stratospheric ozone and thus depleting the earth's protective shield against ultraviolet radiation from the sun. Ozone is made up of three oxygen atoms. It is formed in the stratosphere by the sun's radiation, which breaks up an oxygen molecule into its two constituent atoms. These atoms, being very reactive, immediately react with oxygen molecules to form ozone. Ozone absorbs ultraviolet radiation in the wavelength range of 290-320 nanometers. This radiation is harmful not only to earth's surface life, but also its aquatic life. It could cause skin cancer in humans, retard plant growth, and harm near-surface marine life. To understand the Rowland-Molina theory, which is still valid today, we need to take a closer look at CFCs.

CFCs are simple compounds which contain only chlorine, fluorine, and carbon atoms. Another related family of compounds is hydrochlorofluorocarbons (HCFCs), which contains hydrogen in addition to the ozone-destroying atom chlorine. HCFCs are not as damaging to the ozone layer as CFCs are. Two of the most commonly used CFCs are R-11 which has the structure CCl_3F, and R-12 whose structure is CCl_2F_2. Currently, R-11 is used in air conditioning and industrial chillers while R-12 is used in domestic and industrial refrigeration. CFCs are very stable, non-toxic, non-corrosive, and nonflammable compounds with excellent thermodynamic properties, all reasons for their widespread use in refrigeration, air conditioning, insulation material manufacture, and as propellants in some countries. Their stability close to the earth's surface (lower atmosphere) is in part to blame for their troublesome nature. CFCs diffuse to the upper atmosphere (the stratosphere, 15-40 kilometers above the earth's surface), where exposure to the strong levels of radiation present in the stratosphere causes these normally stable compounds to break up and release reactive chlorine atoms. The chlorine acts as a catalyst which, in its reaction-regeneration cycle, both destroys an ozone molecule by taking its third oxygen, and prevents one from forming by reacting with atomic oxygen. At the end of this process, the chlorine atom is ready to repeat the cycle again and again.

Between 1974 and 1978 mounting evidence on the potential destructive effect of releasing CFCs into the atmosphere and increasing public pressure resulted in a ban on the use of CFCs as propellants in aerosol products (e.g. spray cans) by the U.S. and some other governments. This resulted in the elimination (in theory) of one of the major sources of CFCs in the atmosphere. The problem lay dormant with many countries continuing to use CFCs in aerosols and other applications. By 1985, annual worldwide CFC production was rising by 3%.[3] In 1986, 23% of all CFCs produced were being used as refrigerants, 28% as propellants, 26% for foam insulation blowing, and 21% as solvents for cleaning applications.

The increased use of CFCs led to the 1985 Vienna Convention, which called on participants to formulate a plan for action to determine the danger CFCs posed to the atmosphere and means of dealing with that threat. Unfortunately, an outcome which reflected the lack of urgency and ignorance of the enormity of the problem prevailed on that occasion. That same year, a team of British scientists published data which showed that an ozone hole had been developing over Antarctica since 1980. Their findings were confirmed by others. This discovery brought a sense of urgency to deal with the problem.

With the effects of CFCs on the ozone layer established, unprecedented international action started to take place. In 1987 the Montreal Protocol was signed by most of the world's industrialized countries. The agreement called for a freeze on the production and consumption of CFCs starting in 1989, and a gradual phaseout to end in the year 2000 with the total elimination of CFCs. In the U.S., the new Clean Air Act of 1990 mandated sharper cuts in CFC production but maintained the total phaseout date of the year 2000. In February of 1992, President Bush moved up the total phaseout date for the U.S. to the end of 1995. In November of 1992, the Copenhagen Revision of the Montreal Protocol brought the world phaseout schedule in line with that of the U.S.[4] The U.S. Environmental Protection Agency (EPA) is formulating regulations and certification procedures [5,6] for the recovery and recycle of CFCs since venting of ozone-depleting compounds became illegal at the start of July 1992 (violations carry fines of up to $25,000 a day[7]). Several companies are already advertising CFC Banks and recycle programs in anticipation of a CFC "crunch."[8,9,10,11]

How could these man-made compounds have such a profound effect on the atmosphere? Why doesn't the ozone layer replenish itself? How does the ozone layer deal with chlorine released from natural sources such as sea water and volcanoes? The concentration of ozone in the stratosphere is controlled by a photo-chemical steady state in which the ultraviolet radiation produces ozone, which is consumed by various other reactions resulting in a constant steady state ozone concentration.[12] The introduction of chlorine into the stratosphere causes this steady state to shift to a lower ozone concentration dependent on how much chlorine is present. Since a chlorine atom is not consumed in any of these reactions, it can destroy an estimated 100,000 ozone molecules. It also means that there is a cumulative effect as more chlorine from CFCs enters the stratosphere to add to that which is already present. The atmospheric life of CFCs ranges from 60-500 years, which means that many CFCs released will eventually find their way to the stratosphere.[13] Chlorine from natural sources such as volcanoes never reaches the stratosphere[12] because of its affinity for water. This affinity causes it to dissolve and eventually fall as rain. CFCs act as a Trojan Horse by allowing chlorine to enter the stratosphere before it becomes active.

Alternatives to CFCs

As replacements for CFCs, another class of haloginated compounds which is less harmful to the ozone layer, hydrochlorofluorocarbons (HCFCs), has been proposed as a short term solution. HCFCs have an atmospheric life of 2-20 years as opposed to the 60-500 years of CFCs. It appears, however, that both industry and government are moving to bypass HCFCs and use alternatives to CFCs which have no effect on the ozone layer. This is evidenced in the recent Copenhagen Revision to the Montreal Protocol. These revisions have put in place binding restrictions on HCFCs which will cap their consumption beginning in 1996 to the reported level of use at that time plus 3% of the CFC use level.[4] A complete phaseout is mandated by the year 2030. Countries not complying will have trade sanctions imposed on them. The outlook for eliminating CFCs is good: worldwide CFC output has dropped from 1.13 million metric tons in 1986 down to 680 thousand metric tons in 1991.[8] Industry has already introduced compounds which have no effect on the ozone layer and is investing and committing to finding and using alternatives to CFCs.[14,15]

The class of replacement compounds that industry has focused on is hydrofluorocarbons (HFCs). Because they contain no chlorine atoms, these compounds have no effect on the ozone layer. They have many similarities to CFCs but also several differences. Other alternative refrigerants outside of the fluorocarbon family have been largely ignored; some postulate the reason being industry's desire to sell patentable chemicals. One such alternative is propane. Work in England and Germany on refrigerators which use propane[16,17,18] indicates the feasibility of using this substitute, whose only problem is flammability. Developers claim, however, that the amounts used in a domestic refrigerator are so small (on the order of what is present in two disposable cigarette lighters) as to preclude the risk of explosion.

To compare refrigerants with respect to their effect on the ozone layer, a numeric parameter has been developed which incorporates: a molecule's potential to participate in the ozone depletion process, its atmospheric life span, and the time horizon considered for future ozone depletion.[13] This parameter, normalized with respect to R-11, is called the Ozone Depletion Potential (ODP) of a compound. As this parameter is still evolving, there will be some inconsistency in reported values. The following are some ODPs for various refrigerants:[13,14]

Table 1. Environmental Effects of Various Refrigerants

Refrigerant	Formula	ODP	Est. Atmospheric Life Span (years)
R-11	CCl_3F	1.00	59
R-12	CCl_2F_2	0.99	122
R-13	$CClF_3$	0.45	?
R-113	CCl_2FCClF_2	0.83	98
R-114	$CClF_2CClF_2$	0.71	244
R-115	$CClF_2CF_3$	0.38	539
R-21	$CHCl_2F$	0.04	?
R-22	$CHClF_2$	0.05	18
R-123	$CHCl_2CF_3$	0.02	2
R-142b	CH_3CClF_2	0.06	?
R-125	CHF_2CF_3	0.00	
R-134a	CF_3CH_2F	0.00	
R-143a	CH_3CF_3	0.00	
R-152a	CH_3CHF_2	0.00	
R-290	$CH_3CH_2CH_3$	0.00	
R-717	NH_3	0.00	
R-500	azeotrope: 73.8% R-12 26.2% R-152a	0.74	
R-502	azeotrope: 51.2% R-115 48.8% R-22	0.22	

Source: Epstein, G. J. and S. P. Manwell. "Environmental Tradeoffs between CFCs and Alternative Refrigerants." *ASHRAE Journal*, January 1992; "Industry Backs Bush on CFC-ban Speedup." *Appliance Manufacturer*, April 1992.

GLOBAL WARMING AND THE REFRIGERATOR

The refrigerator contributes to global warming in two ways. First it is an electricity consumer, thus a major part of the energy it consumes comes indirectly from fossil fuels. Fossil fuel combustion generates carbon dioxide, which is the primary gas causing the greenhouse effect and global warming. The second contribution of the refrigerator to global warming is due to chlorofluorocarbons (CFCs) used as refrigerants, and hydrochlorofluorocarbons (HCFCs) used as blowing agents in the manufacture of foams for insulation.

Refrigeration applications in general consume about one fifth of the total electricity generated in the U.S. In 1988 this was 603 billion Kilowatt-hours, 23.4% of the total electricity generated.[19] Refrigerators and freezers consumed 6.9%, air conditioners and heat pumps 10.2%, and commercial and industrial refrigeration 6.3%. To reduce global warming, energy consumption of refrigerators must be reduced. This can be achieved by increasing the mechanical thermodynamic efficiency of the refrigeration system, and by using more and/or better insulation. The U.S. government is moving aggressively in this area. The new 1993 energy standards for refrigerators require 30% more efficiency than the 1990 standard.[20] A study by the Department of Energy (DOE)[21,22] estimates that by the year 2010 the 1993 standards, if not amended, will result in less than a 2% decrease each in the emissions of carbon dioxide, sulfur dioxide, and nitrogen dioxide.

Even though CFC and HCFC emissions are small compared to those of carbon dioxide, their chemical properties make them orders of magnitude more efficient in absorbing infrared radiation and thus contributing to the greenhouse effect. In addition, CFCs absorb infrared radiation in a range of wavelengths where carbon dioxide and water do not absorb, thus compounding the problem.[19] The impending phaseout of CFCs and the introduction of replacements which are less energy efficient introduces an interesting dilemma. By replacing a CFC refrigerant with one which has no greenhouse effect but is less efficient, you would increase the energy consumption of a refrigerator and thus increase the amount of carbon dioxide produced as a result of burning more fossil fuel to supply the increased energy demand. It is thus important not to compromise energy efficiency when switching over to the new generation of refrigerants.

The following list illustrates the Global Warming Potential (GWP) of different refrigerants relative to that of carbon dioxide:[13]

Table 2.
Global Warming Potential of Various Refrigerants

Compound	Formula	GWP
Carbon dioxide	CO_2	1
R-11	CCl_3F	1,300
R-12	CCl_2F_2	3,700
R-11	CCl_2FCClF_2	1,900
R-114	$CClF_2CClF_2$	6,400
R-115 13,800	$CClF_2CF_3$	
R-22	$CHClF_2$	510
R-123	$CHCl_2CF_3$	28
R-134a	CF_3CH_2F	400
R-152a	CH_3CHF_2	46
R-290	$CH_3CH_2CH_3$	0
R-717	NH_3	0
R-500 2,700	azeotrope:	
	73.8% R-12	
	26.2% R-152a	
R-502 7,300	azeotrope:	
	51.2% R-115	
	48.8% R-22	

Source: Epstein, G. J. and S. P. Manwell. "Environmental Tradeoffs between CFCs and Alternative Refriger-

CHOOSING A REFRIGERANT

THERMODYNAMICS

Most current refrigeration systems use the vapor compression cycle which in its most ideal state can be thought of as a reversed Carnot engine. Work is input to the system via the compressor, producing a net effect of pumping heat from a low temperature to a higher temperature. How well a refrigeration cycle operates depends, among other things, on the thermodynamic properties of the working fluid, the refrigerant. The thermodynamic properties of the refrigerant must also be compatible with the operating temperature range. Thus the freezing point of the refrigerant must be well below the operating temperature of the evaporator at the working evaporator pressure; otherwise the refrigerant could freeze in the evaporator. Similarly, the operating pressure in the condenser should be below the critical pressure of the refrigerant, otherwise no condensation will occur.

From an energy efficiency point of view, the most important thermodynamic measure of a refrigerant is its Coefficient Of Performance (COP), which is defined as follows:

$$COP = \frac{\text{Net refrigeration effect of cycle}}{\text{Work input to refrigeration cycle}}$$

The COP provides a measure of how much work needs to be supplied to the system to achieve a given refrigeration effect. Thus the higher the COP of a refrigerant for a given application (the temperature extrema of the cycle), the more energy efficient it is. For a reversed Carnot engine, a purely theoretical concept, the COP is independent of the refrigerant and depends only on the temperatures between which the cycle operates. It can mathematically be derived from the above definition of the COP and is:

$$\begin{matrix} COP = \\ \text{(reversed Carnot)} \end{matrix} \quad \frac{\text{Low temperature of cycle}}{\begin{matrix}\text{Difference between} \\ \text{temperature extrema of cycle}\end{matrix}}$$

This is an idealized measure which serves to define the maximum attainable performance for a given refrigeration application no matter what type of process or working fluid is used.

In practice, other thermodynamic properties come into play which affect the design and consequently the efficiency of a refrigeration cycle. Using a refrigerant with a higher latent heat, for example, means that less of the refrigerant needs to be used to remove a certain quantity of heat. A refrigerant which operates with a large pressure difference between the condenser and the evaporator (high compression ratio) causes the efficiency of the compressor to be lower and its exit temperature to be higher, both undesirable operating features. Reciprocating positive displacement compressors (the type used in the domestic refrigerator) have a space between the piston head and the casing to prevent damage to the piston head in the compression stroke. This space is called the Clearance Volume and is usually 4-15% of the total volume. The Clearance Volume contains refrigerant gas which expands and contracts with every stroke of the piston, which results in a lowering of the actual volume of the refrigerant that is compressed. A quantity which measures this inefficiency is the compressor Clearance Volumetric Efficiency (N_{cv}) which is defined as follows:[23]

$$N_{cv} = 1 - \text{Clearance Volume fraction} \times \left(\frac{\text{inlet gas density}}{\text{outlet gas density}} - 1 \right)$$

As can be seen from the above expression, the higher the inlet to outlet density ratio (or in terms of pressure, the higher the compression ratio), the lower the Clearance volumetric Efficiency. Other factors which contribute to lowering the Total Volumetric Efficiency (actual volume of gas/volume swept by piston) of a compressor are: leakage losses past the piston and valves, throttling in the valves, absorption of the gas in the lubricating oil, and heat exchange losses from the compressor.

Transport Properties

A major departure from the ideality of a reversed Carnot cycle is that the refrigerant has to exchange heat with the surroundings. This requires a finite temperature difference across which heat transfer can occur. The refrigerant flows through pipes, heat exchangers, and expansion valves, all of which produce a pressure drop. The heat transfer in the heat exchangers ties a refrigerant's performance to its heat transfer characteristics: heat capacity, thermal conductivity, density and viscosity. Higher values for heat capacity, thermal conductivity, and density, and lower values for viscosity result in improved heat transfer.

Pressure drop due to the flow of the refrigerant, although not very significant, is related to the viscosity and density of the refrigerant. High density and low viscosity reduce the piping pressure drop due to flow for a particular refrigerant. A refrigerant's transport properties, as well as its thermodynamic properties, are influenced by the type of compressor lubricating oil used.[24] Since the domestic refrigerator uses hermetically sealed compressors, part of the oil (which is totally miscible with the refrigerant) circulates with the refrigerant. This oil-refrigerant solution has markedly different properties from just a pure refrigerant. The compressor lubricating oil is, therefore, an important part of the refrigeration process and is discussed below.

Compatibility with Compressor Oil

In a hermetic reciprocating compressor, the oil serves several purposes:

1. lubricating moving parts in the compressor.

2. providing a seal of the gas between the suction and discharge sides.

3. removing heat from the bearings and crankcase.

4. reducing noise generated by the moving parts.

The oil in a hermetic compressor must possess, in addition to the required lubrication performance at the operating temperatures, the following characteristics:

1. electrical insulation.

2. miscibility with refrigerant, particularly at the low temperatures encountered in the evaporator, where immiscibility will cause reduced heat transfer and poor oil return to the compressor.

3. chemical stability, to last the expected lifetime of the sealed system and withstand the range of operating temperatures encountered.

An important parameter used to characterize a lubricant's miscibility with the refrigerant is the Lower Critical Solution Temperature, defined as the temperature below which immiscibility occurs.

Environmental Impact

With the discovery of the danger CFCs pose to the ozone layer and the ensuing public awareness and government regulations, the environmental impact of using a particular refrigerant has become of paramount importance. The contribution of a refrigerant to global warming, although not regulated yet, is also becoming an important issue. **Table 3** shows current and prospective refrigerants and their Ozone Depletion Potential (ODP), and Global Warming Potential (GWP).

Table 3. Environmental impact parameters of various refrigerants.

Refrigerant	Compound class	ODP	WP
R-11	CFC	1.00	1,300
R-12	CFC	0.99	3,700
R-13	CFC	0.45	?
R-113	CFC	0.83	1,900
R-114	CFC	0.71	6,400
R-115	CFC	0.38	13,800
R-21	HCFC	0.04	?
R-22	HCFC	0.05	510
R-123	HCFC	0.02	28
R-142b	HCFC	0.06	?
R-125	HFC	0.00	?
R-134a	HFC	0.00	400
R-143a	HFC	0.00	?
R-152a	HFC	0.00	46
R-290	hydrocarbon	0.00	0
R-717	ammonia	0.00	0
R-500	azeotrope:	0.74	2,700
	73.8% R-12		
	26.2% R-152a		
R-502	azeotrope:	0.22	7,300
	51.2% R-115		
	48.8% R-22		

Source: Epstein, G. J. and S. P. Manwell. "Environmental Tradeoffs between CFCs and Alternative Refrigerants." *ASHRAE Journal*, January 1992.

Note that ODP is used to calculate the increased taxes on ozone depleting chemicals. Progressively higher taxes and restrictions are being placed on CFCs and HCFCs. Venting of ozone depleting chemicals is currently illegal in the U.S. With current regulations CFCs will be banned by 1996, and HCFCs by 2030.

Current Alternatives

R-12 (DICHLORODIFLUOROMETHANE)

Is the refrigerant currently used in the domestic refrigerator. It has good thermodynamic and transport properties, is non-toxic, nonflammable, and chemically stable. It is, however, a CFC and one of the most harmful to the ozone layer. It has one of the highest GWPs of the refrigerants listed. It will be completely phased out by the end of 1995.

R-134A (TETRAFLUOROETHANE):

Is the leading contender to replace R-12. It has a similar (slightly lower) thermodynamic performance to R-12. It has a zero ozone depletion potential and a small global warming potential. It is nonflammable and very stable. Tests on its toxicity are still underway, but they seem to indicate low[25] or no toxicity.[26] The major problem with R-134a has been its incompatibility with mineral oils used in current refrigeration equipment.[27] R-134a has a low solubility in mineral oils and as a result "drops out" in the evaporator and causes compressor oil starvation. A substitute class of oils, polyalkylene glycols (PAGs), which have good miscibility with R-134a, has fallen out of favor because of its high moisture absorption tendency, its somewhat low electrical resistivity, and its tendency to decompose at high temperatures (about 390°F).[28] Currently, polyol esters seem to offer the solution: they are miscible with R-134a, they are less hygroscopic than PAGs, and they decompose at high temperatures only in the presence of steel. R-134a seems to be on its way to replacing R-12. Ford Motor Company recently announced that Taurus cars are now being built with air conditioners which use R-134a.[29]

R-290 (PROPANE)

This is the refrigerant that is being supported by environmentalists. Thermodynamically, it is equivalent to R-12 and even superior in some aspects. The problem with propane is that it is flammable. Proponents of using R-290 maintain that because its refrigeration effect per pound is more than twice that of R-12, only half the mass of R-12 is required if propane is used. Developers of propane-based refrigerators[17,18] claim the amount of propane is so small that it would need to leak into a space as small as the refrigerator itself for the gas to explode (R-290 explosive limits in air are 2.3-7.3 % by volume[30]). They add that modern systems are so well built that the possibility of leakage is virtually non-existent. The major barrier to the development of propane-based refrigerators seems to be regulatory. In England, for example, standard BS4434 bans hydrocarbons in domestic refrigerators, a restriction Greenpeace is lobbying to eliminate.[17] In the U.S. several gallons of propane can be legally transported and kept indoors. But once the propane is in a fixed installation such as the refrigerator, even when it is in such a small quantity, it is subject to the safety rules of a large facility.[18]

R-152A (DIFLUOROETHANE)

Has been used so far as one of two components in the azeotrope R-500. Its investigation and use had been encouraged at one time by the EPA because it has a lower Global Warming Potential than R-134a.[19] Its thermodynamic performance is slightly better than R-134a and it is compatible with the mineral-based lubricants.[31] A major impediment to using R-152a is its flammability: its explosive limits in air are 5.1-17.1% by volume.[30]

R-717 (AMMONIA)

Has been used mainly as a refrigerant in industrial applications. It is a good refrigerant in many respects. It has a very high refrigeration effect per pound, its heat transfer properties permit use of smaller heat exchangers, and its high critical temperature and low freezing point make it suitable for a wide range of applications. It is readily available and inexpensive. R-717, however, reacts with the copper and aluminum tubing used in current refrigerators, thus requiring the use of more expensive and less conducting iron.[32] Ammonia also requires a high compression ratio which results in a high compressor temperature. R-717 is an

irritant, whose odor can be detected at concentrations of less than 20 ppm.[33] Exposure to concentrations of 1,700 ppm for more than half an hour could be lethal.[23] R-717 is flammable; its explosive limits in air are from 16-25% by volume.[30] Ignition would require a high temperature such as an open flame.[23]

FACTORS WHICH INFLUENCE THE ENERGY CONSUMPTION OF A REFRIGERATOR

Insulation

A refrigerator's main function is to keep an enclosed space at a temperature which is below its surroundings. If a perfect insulator was available, then once the desired low temperature was reached, no energy would need to be expended to keep the contents at their low temperature. Unfortunately, there is no perfect insulator, and the refrigerator must be reasonably priced and have a door. The presence of a door necessitates having some seal, which is usually a gasket. A gasket provides less than ideal insulation and thus, even though it has a small exposed area, accounts for 10-20% of the cabinet's total heat gain.[31] A possible solution: the use of a double gasket which would improve insulation and reduce moisture, reducing the need for the anti-sweat heater and consequently producing more energy savings.

Insulation currently used for refrigerators has a thermal resistivity of R7 per inch (i.e., a thermal conductivity of 1/7 Btu-inch/hr-ft²-°F) and is made of urethane foam which is blown using CFC R-11. With the pending ban of CFCs, alternative blowing agents are being considered, namely R-141b and R-123, both HCFCs.[34] The foam blown using these HCFCs has an insulating value that is approximately 7% less than that blown with R-11. The fact that HCFCs are also ozone depleting, contribute to global warming, and are to be eventually phased out has led to the consideration of carbon dioxide as a blowing agent. Foam made using carbon dioxide has an insulating value which is approximately 16% lower than that made using R-11.[35]

An emerging technology may provide the answer to the quest for a CFC-free, energy efficient insulator. This technology relies on vacuum panels which use various filler materials (powder, aerogel, fiberglass) for support.[36] The most developed of these is the powder

filled vacuum insulation panels. The powder they contain is precipitated silica. Aerogel vacuum insulation panels contain silica glass with a porosity of 90-95%.[35] There are several disadvantages to using these insulation panels: they are difficult to install in the mass production of the refrigerator, expensive, and considerably heavy. **Table 4** provides a rough comparison between the various insulating materials.

Table 4. Insulating parameters of various materials.

Material cost per Relative volume (lb/ft³) unit	R value	Density per inch
Urethane foam with 0.16 R-11 as blowing agent	7	2
Urethane foam with carbon 0.4 dioxide as blowing agent	6	1.7
Fiberglass 0.04	4	1.5
Vacuum insulation 1 (powder, 7.6 torr)	25	12
Vacuum insulation 2 (aerogel, 76 torr)	20-30	6

Source: Feldman, K. Thomas Jr. "Advanced Insulations for Refrigerated Shipping Containers." *ASHRAE Journal*, February 1993.

Refrigeration Cycle Design

Domestic refrigerators currently use the vapor compression cycle and operate with one evaporator removing heat from the freezer, and a condenser rejecting heat to the atmosphere. The theoretical efficiency of the cycle can be improved if heat is removed from a higher temperature and/or rejected to a lower temperature (the ideal reversed Carnot cycle).

Since these temperatures are determined in part by a tradeoff between the evaporator and condenser heat exchange area and the temperature difference between the working fluid and the freezer or fresh food section, improvements in the design of these heat exchangers would allow a reduction in the driving temperature difference, and thus a better cycle efficiency. Another possible refrigerator design modification which would exploit this principle is the use of a dual cycle: one cycle for the freezer and another with higher efficiency for the fresh food section. Such a design could employ separate refrigerants for each cycle and consequently a separate compressor, evaporator, and condenser for each cycle. Alternatively, a single compressor and refrigerant could be used, with separate evaporators for the freezer and, at a higher operating temperature, the fresh food section. The two evaporators would operate at different pressures. A recent theoretical study which evaluated the dual cycle design with different refrigerants predicts energy savings of up to 23%.[37] A complete analysis of these options can be found elsewhere.[34]

A refrigeration cycle which preceded the vapor compression cycle in practical application is the absorption cycle. The first commercial machines, developed by F. Carre in 1850, used ammonia and water as the absorption pair. Interestingly, the source of energy for these machines was not mechanical but thermal, from firewood, coal, or gas. In a typical absorption refrigeration cycle, the refrigerant, ammonia for example, goes through the same steps it would if it were in a vapor compression cycle except that there is no compressor. The ammonia vapor is condensed under elevated pressure in a condenser which rejects heat to the atmosphere; this liquefied ammonia is expanded through a valve into the evaporator where it absorbs heat, producing the refrigeration effect and becoming a gas. The ammonia gas is then mixed with water, where it is absorbed and rejects its heat of solution. The ammonia rich solution is pumped to the generator where a high temperature heat input causes the ammonia to desorb under elevated pressure. This high pressure ammonia gas goes to the condenser and repeats the cycle. The weak ammonia solution returns to the absorber and repeats its cycle. The energy input to the cycle occurs in the generator in the form of heat input rather than work as is the case in the vapor compression cycle. This provides a degree of flexibility: first of all, "low quality" energy can be used; second, the energy can be supplied directly from the

fuel without having to go through several intermediate steps such as electricity generation and mechanical compression as is the case in the vapor compression cycle. Currently, the only widespread use of the absorption cycle is in Japanese air conditioners and in mini-bars in hotel rooms.[16] The absence of a compressor makes possible the quiet operation required of the second application.

Individual Component Efficiencies

The efficiencies of the individual electric devices that a refrigerator contains also influence its energy consumption. The major consumer is the compressor and its driving motor. A standard rating number used to describe the efficiency of a compressor is the Energy Efficiency Rating (EER), which is defined as follows:

$$EER = \frac{\text{Refrigeration effect achievable by compressor in Btu/hr}}{\text{Power input to the compressor in Watts}}$$

Note that the definition of EER shows a dependence on the refrigeration effect, which depends on the refrigerant used in the cycle and the operating temperatures of the cycle. Compressors rated between 4.0 EER and 4.5 EER have been available for some time. Recently compressors rated from 4.5 EER to 5.5 EER have become available.[31,38]

Minor electrical components such as the condenser and evaporator fans are another potential area of improvement. Currently, low cost, low efficiency (10%) motors are generally used.[2] The potential exists for using induction motors (30% efficiency) or small DC motors (65% efficiency). Another possible technology to improve efficiency is Variable-Speed drive, which could increase motor efficiency by 10-15%.[31]

LIFE CYCLE DESIGN

At a recent exhibition in Europe a prototype refrigerator named "Green Frost" was demonstrated.[39] The entire cabinet, including external structural parts and insulation, is made of polystyrene. The insulation is made from expandable CFC-free polystyrene which is foamed into the cabinet. The use of a single material in most of the refrigerator makes recycling not only possible but also profitable. Modular design is

followed and design for ease of assembly/disassembly insures that most of the refrigerator is recoverable.

The above demonstrates a growing trend, further along in Europe than the U.S., which attempts to look at a process or product design through a total systems approach which is not confined to the immediate perimeter of the process or product. Essential to such an approach is total cost assessment, which incorporates hidden, liability, and often ignored environmental costs into the accounting procedure.[40,41] Such a procedure will eventually become standard as the movement grows to make manufacturers responsible for their products "from the cradle to the grave."

New laws enacted recently in Germany require manufacturers to take back their products. The German automobile manufacturer BMW has a program in place whereby its customers can return their old automobiles for recycling. BMW is designing its automobiles to enhance recyclability by using such simple techniques as color coding and parts labeling, and such complex techniques as redesign to use compatible materials. An approach such as this not only reduces the waste stream that eventually must be disposed of, but also slows resource depletion.

Basic to accepting life cycle design is the understanding that the Earth's resources are finite. There is a limited quantity of raw materials, a limited quantity of energy, and limited disposal space (land, water, air).

Processing of raw materials requires energy conversion, which produces pollution. The limitations of recycling necessitate waste disposal. Such disposal requires space in landfills or poisons the air and water, and consumes energy. Thus the need for pollution prevention at the source to avoid the generation of the waste in the first place. Once waste is generated it will eventually end up in one form or another in either the air, water, or land. Waste treatment, which usually shifts waste from one medium to the other (e.g. incineration reduces solid waste but increases air pollution) does not solve the problem.

In order for life cycle design to work, it must be implemented very early in the life of a process or product and not done as an afterthought. A systematic approach which keeps in mind the limited resources of the planet (raw materials, energy, disposal space), and the fact that ultimately regulations and public pressure will protect these limited resources must be followed to avoid increased financial and public relations costs. In design there is no magic recipe; tradeoffs must be made between capital expenditures, operating costs, regulations, customer preferences, etc. What life cycle design practices attempt to do is insure that environmental liabilities and benefits are accounted for. In many cases this results in dividends not only to the environment, but also to the manufacturer.

Recommended Reading

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Section III:
The Design

INTRODUCTION

This design addresses two major issues: the first is the replacement of R-12, the refrigerant that has been used in all domestic refrigerators to date, with a refrigerant that is not ozone depleting. From the literature survey summarized in the teaching aids, R-134a seems to be the refrigerant of choice for most of the industry. This, however, does not and should not preclude the possibility of choosing other refrigerants since a great many factors influence this choice. The second issue this design addresses is energy efficiency. The deciding factors in this issue are insulation, thermodynamic performance of the refrigerant, and the efficiency of the various power-consuming devices that make up the refrigerator such as the compressor; and the evaporator and condenser fans. In this study, we will focus on insulation and on the thermodynamic performance of the refrigerant.

ALTERNATIVE REFRIGERANTS TO BE CONSIDERED

We will evaluate R-12 as a base case. R-134a will be evaluated since it seems to be the leading candidate to replace R-12. Other refrigerants which pose undesirable properties that are unrelated to their thermodynamic performance will also be evaluated. These are R-290, R-717, and R-152a. Finally, two patented azeotropes — AZ-20[1] and AZ-50[2] — which are not intended for the operating range of the domestic refrigerator will be evaluated to provide insight into the reasons for their unsuitability.

REFRIGERATION CYCLE COMPONENTS

Figure 1 illustrates the main components of the refrigeration cycle in the refrigerator. Cool gas from the evaporator (stream 1) which is close to the freezer temperature is superheated by warm liquid (stream 4), close to ambient temperature, from the condenser. This superheating is necessary to insure that no liquid refrigerant is carried over to the compressor. The superheated gas (stream 2) pressure is brought up to the condensing pressure by the compressor. As the result of compression, the gas (stream 3) is superheated even further. In the condenser, this gas is first desuperheated and then condensed to a saturated liquid (stream 4) by rejecting heat to the outside air. The saturated liquid goes through the interchanger where it is subcooled (stream 5). The expansion valve drops the pressure of the liquid to the evaporator pressure (stream 6). In the evaporator the liquid evaporates by absorbing heat from the inside of the refrigerator (the freezer). The saturated refrigerant (stream 1) leaves the evaporator and repeats the cycle.
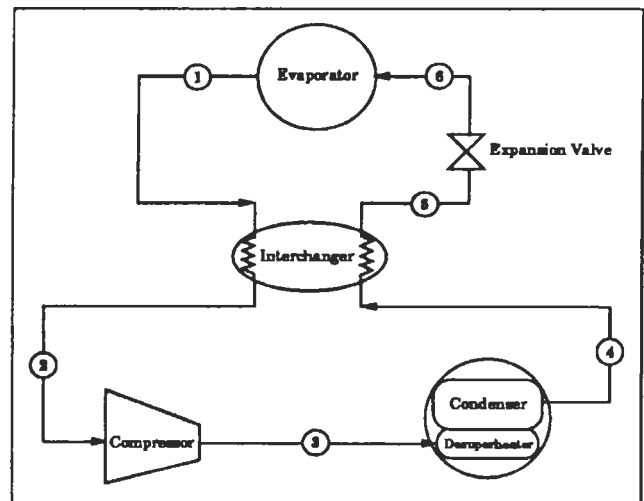

Figure 1. Major components in a refrigerator.

In **Figure 2**, the vapor compression cycle described above is shown on an Enthalpy-Pressure Diagram. The cycle shown ignores pressure drops due to the flow of the refrigerant through piping and heat exchangers. It also assumes isentropic (constant entropy) compression in the compressor. The numbered points on the diagram refer to the numbered streams in **Figure 1**. Point 1 represents the saturated vapor leaving the evaporator. This vapor is superheated in the interchanger to point 2, thus going from the saturated vapor temperature at the evaporator pressure to the temperature given by the constant temperature line on the diagram. The gas is compressed isentropically along the constant entropy line up to the condenser pressure. The temperature of this superheated gas can be found from the constant temperature line at point 3. The condenser brings the

temperature of the gas down to the saturation temperature at the condenser pressure. This is the point where the upper isobar intersects the saturated vapor curve. The saturated vapor then condenses at constant temperature to a saturated liquid at point 4. This liquid is subcooled in the interchanger from point 4 to point 5. Since the heat gained in going from point 1 to point 2 is the same heat quantity rejected in going from point 4 to point 5, and since the flow rate is the same (closed system), the length of the segment from point 1 to point 2 is equal to the length of the segment from point 4 to point 5. The expansion valve drops the pressure isenthalpically from point 5 to point 6. The line from point 6 to point 1 represents the enthalpy increase of the refrigerant in the evaporator as the saturated liquid is evaporated.
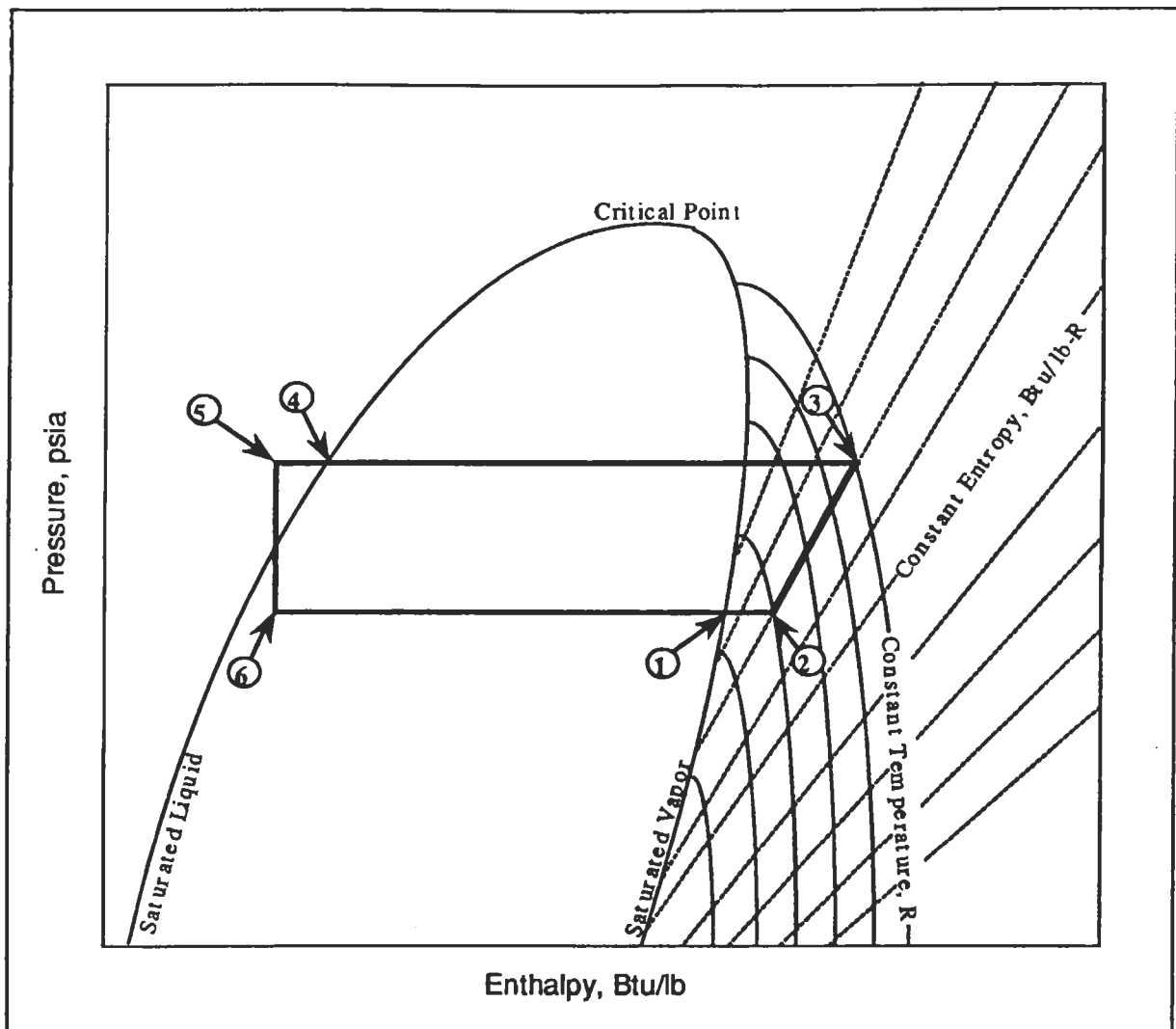

Figure 2. An Enthalpy-Pressure Diagram illustrating the refrigeration cycle shown in Figure 1. Isentropic compression and zero piping pressure drops are assumed.

REFRIGERATION CYCLE CALCULATIONS

Based on the design conditions given, the evaporator temperature (T_1) will be set at -4°F. This provides a temperature difference of 9°F between the required freezer design temperature of 5°F and the evaporator temperature. It also allows for operation at the extreme freezer design temperature of 0°F. The ambient design temperature is 90°F with the extreme going to 110°F. A condenser temperature (T_4) of 115°F is selected. In the interchanger, a superheating of the saturated vapor (stream 2) of 14°F is assumed. Two methods to calculate the performance of the cycle for different refrigerants are used. The first is a graphical technique which is based on the Enthalpy Pressure Diagram of a refrigerant (a comprehensive collection of diagrams for various refrigerants can be found in the ASHRAE Handbook[3]). The second utilizes tables of saturated refrigerant liquid and vapor properties, and an expression for the specific heat of the refrigerant. Both techniques are explained through the sample calculations in **Appendix D**.

RESULTS AND DISCUSSION OF REFRIGERATION CYCLE CALCULATIONS

Figure 3 combines the cycles of the various refrigerants on a single diagram. Looking at the different cycles, two of R-12's virtues immediately become apparent. It has the lowest compression ratio and operating pressure range of all the refrigerants considered. Note how R-717, ammonia, operates at the highest pressure range and has the largest compression enthalpy change, which leads to a high compressor exit temperature.

Table 5 shows various performance criteria for the candidate refrigerants. The first parameter, the Coefficient of Performance (COP), is given for the required design conditions. As can be seen, only R-290 has an equivalent (1% higher) performance to R-12. R-152a is the next highest, with AZ-20 and AZ-50 having the lowest COP. R-134a's COP is about 7% lower than that of R-12, which translates into a lower energy efficiency.

Table 5. Performance properties of candidate refrigerants.

Refrigerant	COP	RE	$RE_v \times N_v$	T_3(°F)
R-12	3	45	16.2	150
R-134a	2.8	55	13.1	140
R-290	3.03	106	23.4	140
R-717	2.76	448	28.4	340
R-152a	2.95	94	14.3	169
AZ-20	2.6	72	40.4	220
AZ-50	2.54	44	25.5	148

Source: *ASHRAE Handbook*, Inch-Pound Edition. Atlanta, GA: American Society of Heating, Refrigerating and Air-Conditioning Engineers. 1989.

Key: COP = Coefficient of Performance; RE = Refrigeration Effect; $RE_v \times N_v$ = Product of Volumetric Refrigeration Effect and Volumetric Efficiency; T_3 = Compressor Discharge Temperature

The Refrigeration Effect (RE) is the amount of heat removed by a unit mass of refrigerant. Conversely, it determines the mass of refrigerant needed to remove a given quantity of heat. R-717 has the highest RE. It is ten times that of R-12, which means only one tenth the mass of R-717 is needed to achieve the same refrigeration effect as that of R-12. A more important quantity is the Volumetric Refrigeration Effect (RE_v) combined with the compressor Volumetric Efficiency (N_v). These two quantities determine the size of the compressor that should be used. The product of RE_v, the refrigeration effect per unit volume, and N_v, the ratio of the actual volume of gas pumped by the compressor to the compressor displacement, determines the refrigeration effect a given compressor will produce with different refrigerants. Thus, the higher $RE_v \times N_v$ is for a given refrigerant, the smaller the compressor displacement and the compressor itself will be. **Table 5** also compares the various refrigerants on this basis. AZ-20 has the highest $RE_v \times N_v$, 2.5 times that of R-12; thus the compressor needed would be 40% smaller than that for R-12. R-134a would need a compressor with a displacement 1.25 times that of R-12. R-290 on the other hand would require a compressor that is 70% smaller than that for R-12. Another factor that needs to be
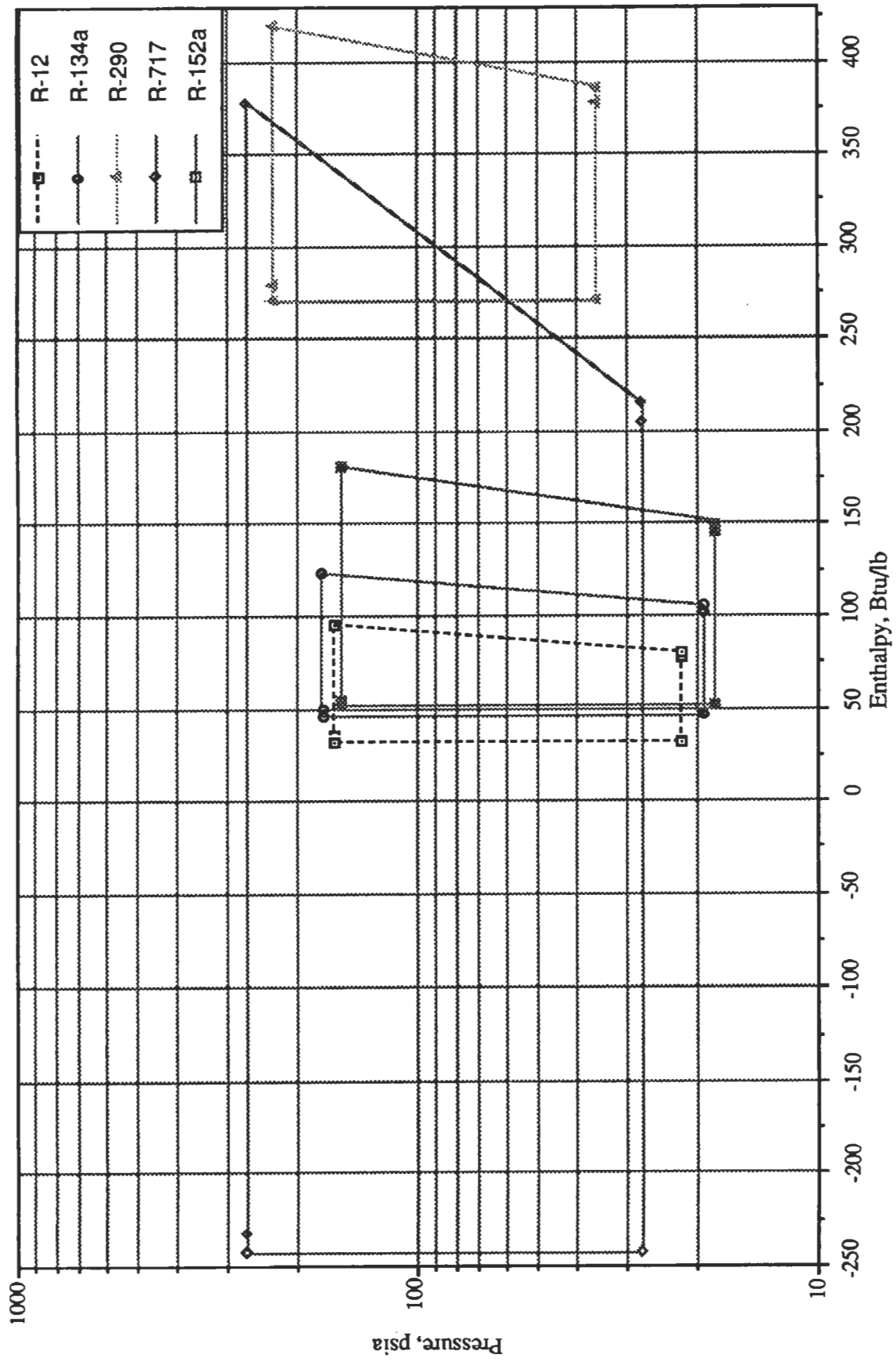
Figure 3. The refrigeration cycles of the various candidate refrigerants

considered is the compressor discharge temperature (T_3). Too high a discharge temperature could degrade both the compressor valves and the lubricating oil. As these data show, most of the refrigerants considered have a reasonable T_3, with the exception of R-717 (T_3 = 340°F) and to a lesser extent AZ-20 (T_3 = 169°F).

REFRIGERATION LOAD

Using the pull down time of 2 minutes given in the problem statement, the refrigeration system has to be able to reject 670 Btu/hour to the ambient air (see **Appendix D**). At the extreme design conditions, the pull down time will increase to 2.7 minutes, which is acceptable. Based on this load, the mass flow rate of each refrigerant can be calculated and is directly proportional to its RE.

HEAT EXCHANGER AREAS

Using the expressions supplied in the problem statement for the internal heat transfer coefficients (h) and overall heat transfer coefficients (U) of the desuperheater (desuperheat of the condenser), the condenser, and the evaporator, the various heat exchanger areas can be calculated (**Appendix D**). The results are shown in **Table 6**, in plain type for the

desuperheater, in bold type for the condenser, and in italics for the evaporator. The areas for the condenser and the evaporator do not vary much for the different refrigerants. The reason for this is that although the internal heat transfer coefficients vary, they are an order of magnitude higher than the external heat transfer coefficient, which depends only on the geometry of the exchanger and its fan. The controlling resistance is on the outside. Additionally, the temperature driving force and heat load in the evaporator is the same for all the refrigerants, and the heat load in the condenser does not vary much. In the desuperheater, the internal heat transfer coefficients are of the same order of magnitude as the external heat transfer coefficient and thus their variation influences the overall heat transfer coefficient. In the case of R-717, the internal heat transfer coefficient becomes the controlling resistance. It is interesting to note that even though R-717 has the lowest overall heat transfer coefficient and the highest heat transfer load in the desuperheater, it had the smallest exchanger area because of the large temperature difference driving force caused by the high compressor discharge temperature.

Table 6. Heat transfer parameters for candidate refrigerants

Refrigerant	R-12	R-134a	R-290	R-717	R-152a
Mass flow rate	14.96	12.21	6.32	1.496	7.1
(lb/hr)	**14.96**	**12.21**	**6.32**	**1.496**	**7.1**
	14.96	*12.21*	*6.32*	*1.496*	*7.1*
Internal Heat	13.6	13.8	15.4	4.5	9.1
Transfer Coeff. (h)	**848**	**845**	**961**	**572**	**672**
(BTU/hr-ft^2-°F)	*641*	*735*	*657*	*412*	*487*
Overall Heat 7.0	7.1	7.4	3.7	5.8	
Transfer Coeff. (U)	**12.5**	**12.5**	**12.5**	**12.4**	**12.5**
(BTU/hr-ft^2-°F)	*10.7*	*10.7*	*10.7*	*10.6*	*10.6*
Temperature	35	25	25	225	54
Difference (°F)	**25**	**25**	**25**	**25**	**25**
	9.0	*9.0*	*9.0*	*9.0*	*9.0*
Heat flow rate (Q)	89.8	91.9	91.4	223.9	112.0
(BTU/hr)	**805**	**817**	**800**	**688**	**785**
	670	*670*	*670*	*670*	*670*
Heat Exchange	.364	.520	.490	.280	.360
Area (ft^2)	**2.58**	**2.61**	**2.56**	**2.22**	**2.52**
	6.96	*6.95*	*6.96*	*7.00*	*7.00*

Source: *ASHRAE Handbook*, Inch-Pound Edition. Atlanta, GA: American Society of Heating, Refrigerating and Air-Conditioning Engineers. 1989.

Key: desuperheater (regular), **condenser (bold)**, and *evaporator (Italics)*

ENERGY CONSUMPTION

The energy consumption of the refrigerator is comprised of:

1. the electrical energy input to the compressor to drive the refrigeration cycle and remove heat from the interior of the refrigerator. This heat is gained by conduction through the walls and gaskets of the refrigerator, and is also generated internally by the anti-sweat heater.

2. electrical energy input to the evaporator and condenser fans, and to the anti-sweat heater.

According to the 1993 DOE energy efficiency standard for the size and type of refrigerator under consideration, the maximum allowable energy consumption is 578.6 KW-hour/year (see **Appendix D**). To determine how best to meet this criterion with the insulation materials available and the chosen candidate refrigerants, two useful figures have been prepared. **Figure 4** (see **Appendix D** for calculations) shows the heat gain of the refrigerator as a function of the fresh food section insulation thickness for different insulation R values. The heat gain shown includes that which is gained by the freezer. The freezer insulation thickness was calculated so that the heat gain per unit insulation volume in the fresh food section is equal to that of the freezer. This provides for optimum use of the insulation. **Figure 5** (see **Appendix D** for calculations) shows the total annual energy consumption of the refrigerator as a function of the total heat load that the refrigeration cycle has to reject to the atmosphere. This dependence
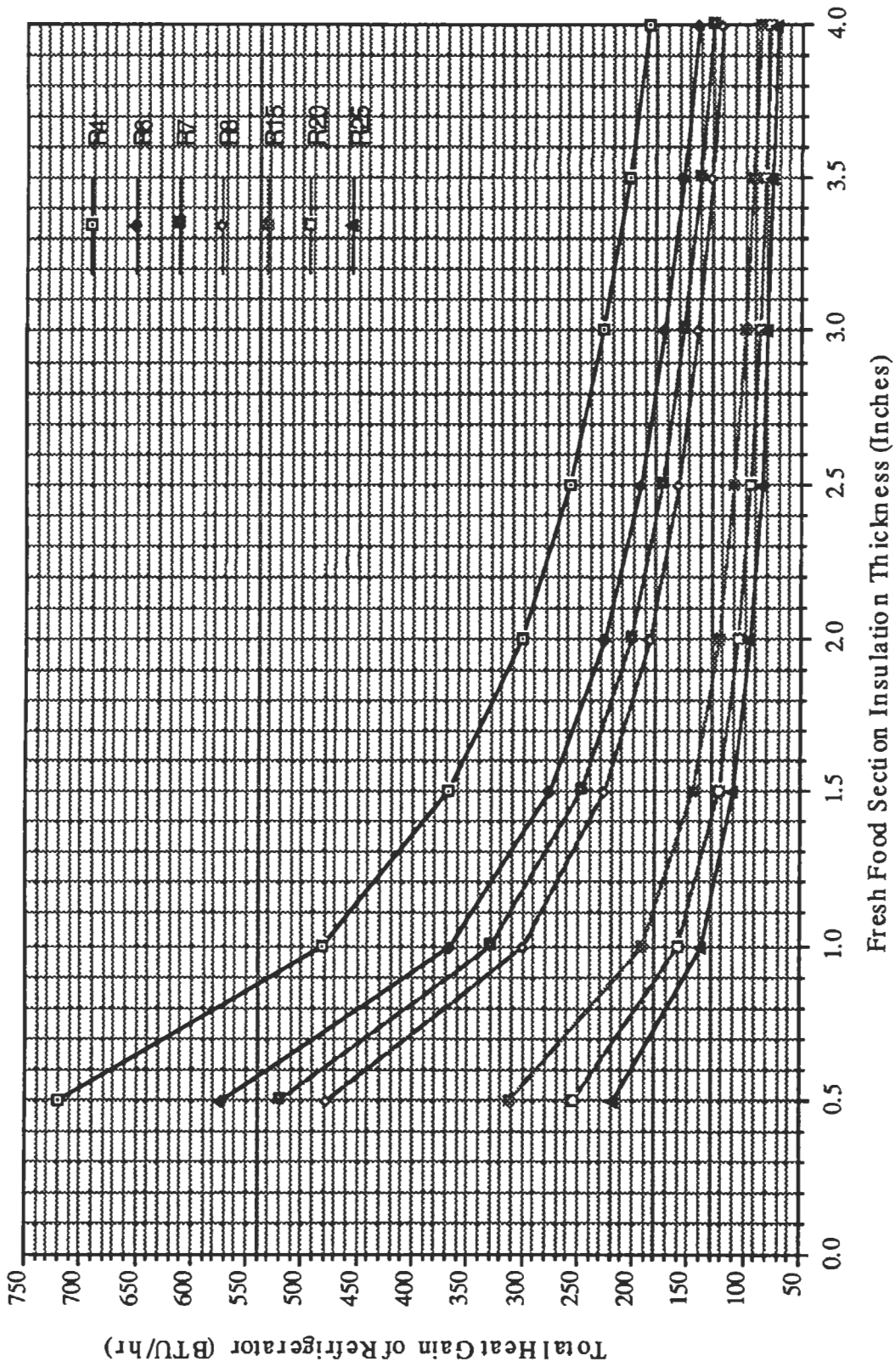
Figure 4. Heat flow into the refrigerator vs. insulation thickness for various insulation R-values.

Freezer insulation thickness was calculated to provide equal heat gain per unit insulation volume in the freezer and fresh food section.
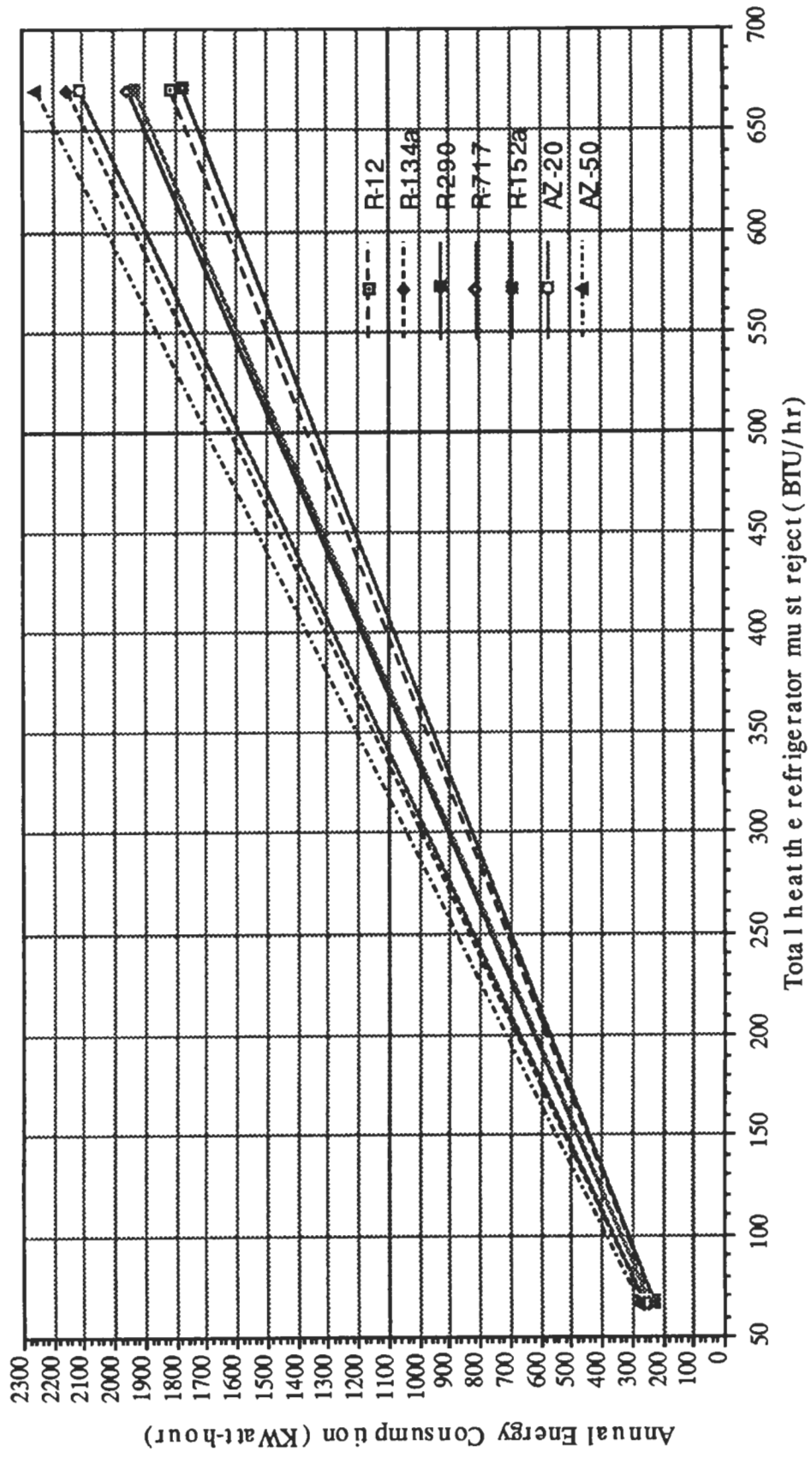
Figure 5. Total heat (including anti-sweat heater) to be rejected vs. Annual energy consumption.

Total heat the refrigerator must reject (BTU/hr)

Annual Energy Consumption (KWatt-hour)

R-12
R-134a
R-290
R-717
R-152a
AZ-20
AZ-50
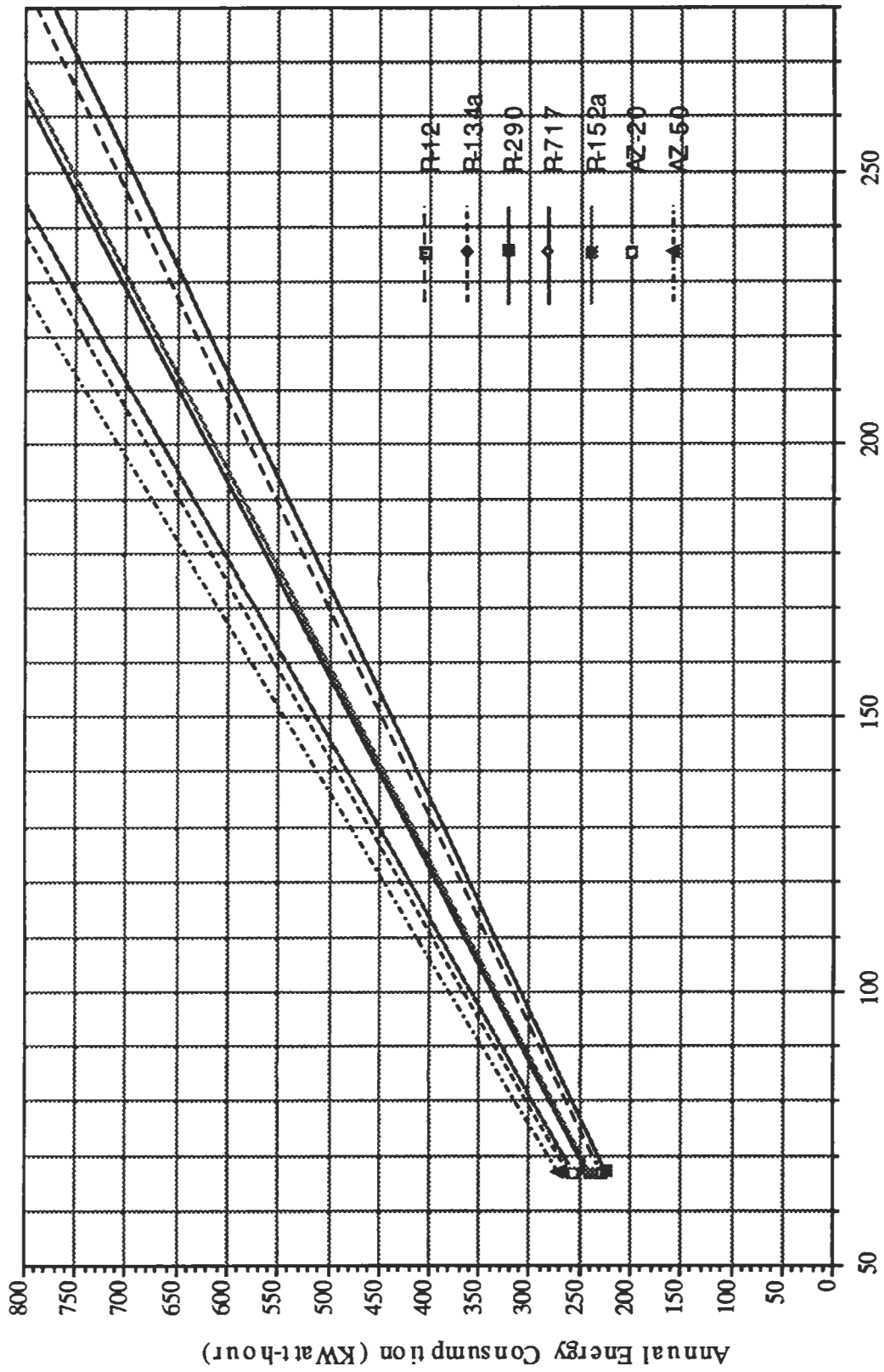
Figure 6. Total heat (including anti-sweat heater) to be rejected vs. Annual energy consumption. This is a close-up view of the lower end of **Figure 5**.
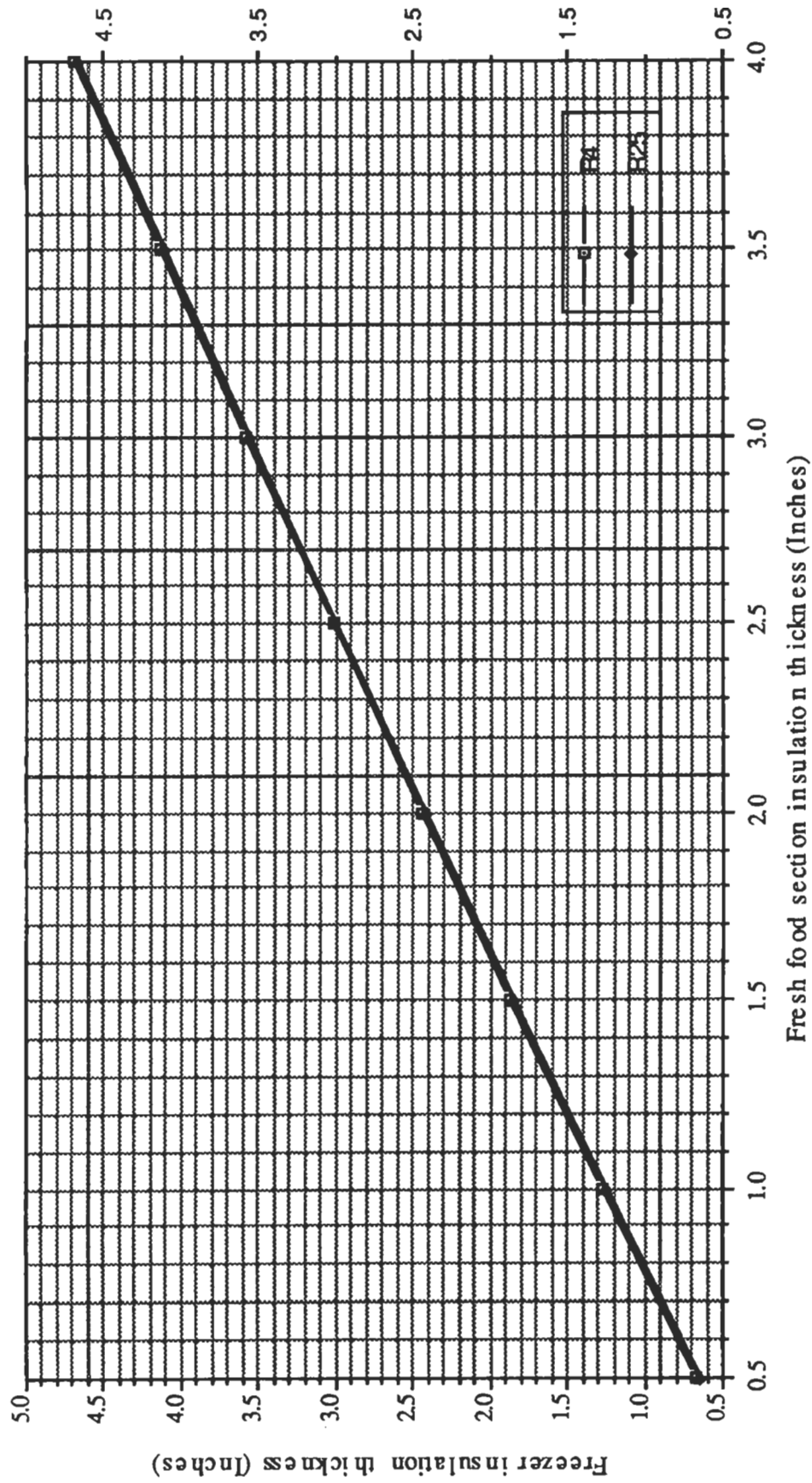
Figure 7. Fresh food section insulation thickness and corresponding freezer insulation thickness.

is shown for the different candidate refrigerants.

In the old refrigerator design which uses R-11 blown urethane foam with an insulation value of R7, the fresh food section insulation thickness was 1.5 inches (1.85 inches for the freezer). From **Figure 4**, this corresponds to a heat gain of 250 Btu/hour. Adding on the 19.5 Btu/hour generated internally by the anti-sweat heater gives a total of 269.5 Btu/hour that the refrigeration cycle must remove from the interior of the refrigerator. Using this number and the line for R-12 in **Figure 5** results in the energy consumption of our old design: 780 KW-hour/year. To meet the 1993 DOE standard, the energy consumption must be dropped 26%, down to 578 KW-hour/year. If we were still able to use R-12, the total heat the refrigerator is able to reject using R-12 and the allowable energy consumption would be 200 Btu/hour (using **Figure 6**, which provides more detail of the lower end of **Figure 5**). Accounting for the anti-sweat heater (heat gain through insulation must be less than 180.5 Btu/hour) and using **Figure 4**, we would need a 2.4 inch thick R7 insulation in the fresh food section (2.9 inches in the freezer by using **Figure 7**). If we were to use R-134a, we would need an R7 insulation thickness of 3.3 inches in the fresh food section (3.4 inches in the freezer from **Figure 7**). Since R-11 is a CFC, the foam made by using it will no longer be available. An alternate blowing agent under consideration is carbon dioxide. It will, however, provide a lower insulation value of R6 which translates to a fresh food section insulation thickness of 3.8 inches and a freezer insulation thickness of 4.5 inches. One can calculate that this change would reduce the fresh food section by over one third, and the freezer volume by over a half.

These are, therefore, unacceptable and impractical insulation thicknesses. In the short term, use must be made of HCFC blown foams which are approximately 7% less effective than the CFC blown foams. An alternative longer term solution would be the use of vacuum insulation which provides insulating R values of 20-30. Using an R25 insulator with R-134a would require a fresh food insulation thickness of 0.95 inches (1.2 inches in the freezer) to meet the 1993 DOE standard. Using R-290 and the allowable energy consumption dictated by the 1993 DOE standard, the total heat the refrigerator is able to reject to the atmosphere is 207 Btu/hour. Of that load, 19.5 Btu/hour is generated by the anti-sweat heater, which leaves a maximum load of 187.5 Btu/hour that the insulation must handle. Using the carbon dioxide blown foam (R6), a fresh food insulation thickness of 2.6 inches (3.1 inches in the freezer) needs to be used.

From the above analysis, the most energy efficient CFC-free refrigerator could be realized by using R-290 and vacuum insulation. Assuming an R25 vacuum insulation and the original design's fresh food section insulation of 1.5 inches (1.85 inches for the freezer), the heat gain through the insulation will be 110 Btu/hour. The total refrigeration load (including anti-sweat heater) is therefore 129.5 Btu/hour. The refrigerator's energy consumption using R-290 would be 380 Kwatt-hour/year, 34% less than the 1993 DOE standard. If regulatory barriers to using R-290 (propane) cannot be overcome, then R-134a has to be used. With R-134a, the energy consumption would go up to 430 Kwatt-hour/year, 26% less than the 1993 DOE standard. In conclusion, the key to an energy efficient refrigerator which does not use CFCs lies in developing insulation technology.

REFERENCES

[1] "Genetron-AZ-20, Technical Bulletin." Morristown, NJ: Allied Signal — Fluorocarbons, 1993.

[2] "Genetron-AZ-50, Technical Bulletin." Morristown, NJ: Allied Signal — Fluorocarbons, 1993.

[3] "Fundamentals," Chapter 17 in *ASHRAE Handbook*, Inch-Pound Edition. Atlanta, GA: American Society of Heating, Refrigeration and Air-Conditioning Engineers. 1989.

Appendices

APPENDIX A

Manufacturer-supplied information on regulations regarding the phase out of chlorofluoro-carbons and hydrochlorofluorocarbons.

The Revised Montreal Protocol
Copenhagen, November 1992

ICI Americas Inc.

The Revised Montreal Protocol Copenhagen, November 1992

The Montreal Protocol was revised and considerably strengthened at a meeting of the United Nations Environment Program on November 25, 1992. The changes should come into force by January 1, 1994. Some changes affect the timetable for controls on the CFCs, halons, 1,1,1-trichloroethane (methyl chloroform) and carbon tetrachloride (CTC) already in the Protocol; these are mandatory for all countries that are already Parties to the Protocol. Other changes bring in new substances, such as the hydrochlorofluorocarbons (HCFCs), the hydrobromofluorocarbons (HBFCs) and methyl bromide. At the meeting, the Parties also passed an amendment which limits the areas of application of the HCFCs.

ICI welcomes this agreement which greatly strengthens the London Amendments of 1990. Although there are areas where individual countries should be able to go faster and further than the new timetables, we believe this was the most stringent agreement which could have achieved the support of all the Parties.

All of these substances are to be phased out completely, with the exception of methyl bromide, but the timetables differ. The details are given below in the tables; dates refer to January 1 of the year stated; all freezes and reductions are based on 1986 levels for the CFCs and halons, 1989 for Other CFCs, CTC and 1,1,1-trichloroethane, and 1991 for methyl bromide. HCFC controls begin with a consumption freeze in 1996. This is known as the HCFC cap.

	"Original CFCs"[1]	"Other CFCs"[2]	Halons	CTC	Methyl Chloroform
Freeze	*	–	*	–	1993
-20%	–	1993	–	–	–
-50%	–	–	–	–	1994
-75%	1994	1994	–	–	–
-85%	–	–	–	1995	–
-100%	1996	1996	1994	1996	1996

*The freeze for the original CFCs and Halons is already in force.

[1] "Original CFCs" are those controlled in the 1987 Montreal Protocol – CFCs 11, 12, 113, 114 and 115

[2] "Other CFCs" are all fully halogenated 1, 2 and 3 carbon CFCs, other than the five "original CFCs"

	HCFCs*	HBFCs	Methyl Bromide
Freeze[1]	1996	–	1995
-35%	2004	–	–
-65%	2010	–	–
-90%	2015	–	–
-99.5%	2020	–	–
-100%	2030	1996	–

*The HCFCs controlled are all of the 1, 2 and 3 carbon compounds containing hydrogen, chlorine and fluorine.

[1] The levels of the freeze and percentage reductions are based on the total ozone depletion potential (ODP) weighted tonnage of HCFC consumption in 1989, plus 3.1% of the ODP tonnage of CFC consumption in 1989. ("Consumption" = production + imports – exports; "production" excludes material used as a feedstock)

KLEA

The Revised Montreal Protocol Copenhagen, November 1992

Destruction Technologies

Five destruction techniques were approved for situations where reclaim of allowances may take place (probably not reclaimable in the US). All five are incineration techniques and no overall minimum percentage efficiency of destruction was defined. Basic standards were suggested and a Code of Good Housekeeping was adopted.

Definition of Production: Inadvertent Production, etc.

The definition of "production" (i.e., reportable and subject to allowances) was revised to exclude trace impurities generated during manufacture, remaining after feedstock use or released during manufacturing processes. Parties are urged to minimize these releases by various techniques.

KLEA

ICI

Fluorochemicals

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The following is a concise overview of the regulatory actions in the United States that are impacting the use of chlorofluorocarbons (CFCs), halons, methyl chloroform, and carbon tetrachloride and their alternatives. For more detailed information, please contact Du Pont on 1-800-441-9442.

THE MONTREAL PROTOCOL

Following several years of negotiations, an international agreement regulating the production and use of CFCs, halons, methyl chloroform, and carbon tetrachloride entered into force in mid 1989. Known as the Montreal Protocol, this landmark agreement initially required a production and consumption freeze. It currently calls for a stepwise reduction of CFCs, halons, methyl chloroform, and carbon tetrachloride in developed countries that will result in a complete phaseout of production and consumption by the year 2000 (2005 for methyl chloroform). Developing countries have been given an additional 10 years to complete the transition to new technologies.

Parties to the Protocol will be meeting in Copenhagen, Denmark in November of 1992 to renegotiate current agreements. The revised Protocol will likely result in accelerated phaseout of CFCs, halons, methyl chloroform, and carbon tetrachloride, and include a binding resolution for HCFC phaseout.

The Protocol is a joint effort of governments, scientists, industry, and environmental groups. Coordinated by the United Nations Environment Programme (UNEP), it has been ratified by about 80 sovereign nations, representing greater than 90 percent of the world's current CFC consumption.

Trade sanctions will enter into force on January 1, 1993 against those "non-party" nations, that are not signatories to the Protocol.

CLEAN AIR ACT

On November 15, 1990, the new Clean Air Act (CAA) was signed into law. This legislation includes a section entitled Stratospheric Ozone Protection (Title VI), which contains extraordinarily comprehensive regulations for the production and use of Class I compounds (CFCs, halons, carbon tetrachloride, methyl chloroform), Class II compounds (HCFCs), and HCFC/HFC substitutes. These regulations, which will be phased in over the next several years, will affect every industry that currently uses chlorinated and brominated substances that impact stratospheric ozone.

Table 1 shows how the CAA has accelerated the phaseout for Class I compounds in the United States versus the current Montreal Protocol.

TABLE 1
Phaseout Schedules for CFC Production

	Fraction of 1986 Production			
	1990 Montreal Protocol	Possible 1992 Montreal Protocol	1990 U.S. Clean Air Act	Estimate of EPA Regulations
1992	1.0	1.0	0.80	0.50
1993	1.0	1.0	0.75	0.50
1994	1.0	0.25*	0.65	0.40†
1995	0.50	0.25*	0.50	0.25
1996	0.50	0	0.40	0
1997	0.15	0	0.15	0
1998	0.15	0	0.15	0
1999	0.15	0	0.15	0
Equiv. years	4.45	2.50	3.55	1.65

* Range of proposals that have some support are 0.20 to 0.30

†EPA regulations must meet or exceed 1992 Montreal Protocol.

The major provisions of the CAA include:

- Phaseout schedules
- Mandates for recycling, recovery of refrigerants in auto air conditioning and stationary refrigeration equipment
- Bans on "nonessential" products
- Dictates concerning safe alternatives
- Labeling for containers of and products containing or made with Class I or Class II compounds.

The regulations of the CAA are stringent, but achievable. Support of alternatives and cooperation by all groups (industry, environmentalists, and regulators) is necessary to meet these stringent requirements.

PHASEOUT SCHEDULES

The current Clean Air Act Amendments call for a phaseout of Class I substances by January 1, 2000. On February 11, 1992, President Bush called for an acceleration of the phaseout of Class I compounds to year-end 1995, with limited exemptions for essential uses, which have not yet been defined.

The EPA must implement this, as well as interim cuts, by rulemaking. As of this printing, the rulemaking has not been finalized, the exemptions have not been identified, and interim cuts have not been established.

The existing Clean Air Act phaseout and reductions (until rulemaking is final) are:

- CFCs and halons (Class I substances) will be phased out in steps until total phaseout on January 1, 2000 (Table 1).

- Carbon tetrachloride (Class I) will undergo stepwise phaseout to 85 percent of non-CFC feedstock use by January 1, 1995, and total phaseout by January 1, 2000.
- Methyl chloroform (Class I) will undergo stepwise phaseout to 50 percent of 1989 levels by January 1, 1996, and total phaseout by January 1, 2002.
- HCFCs (Class II substances) are regulated as follows:
  - Production frozen and use limited to refrigeration equipment on January 1, 2015.
  - Use in new refrigeration equipment is allowed until January 1, 2020.
  - Only service of in-place refrigeration equipment is allowed after January 1, 2020, and a total HCFC production ban becomes effective on January 1, 2030.
- EPA can accelerate phaseout schedules for Class I and II compounds if they deem it necessary for health or environmental reasons, if technology is feasible, or if required by the Montreal Protocol.
- EPA may grant small and limited exceptions for Class I and Class II production for a few essential safety, medical, or national security uses.

RECYCLING
- The venting of CFC and HCFC refrigerants during service, maintenance, repair, or disposal of appliances and industrial process refrigeration is illegal as of July 1, 1992.

MOBILE AIR CONDITIONING
- CFC and HCFC refrigerants for mobile air conditioning must be captured and recycled by certified personnel after January 1, 1992. There will be a one-year delay for persons servicing less than 100 cars per year. In addition, sales of small cans (less than 30 lbs) will be restricted to certified personnel.
- HFCs, for mobile air conditioning will have the same regulations by November 12, 1995.

NONESSENTIAL PRODUCTS
- CFCs will be banned for nonessential products, such as party streamers, noise horns and noncommercial cleaning fluids, and others identified in rulemaking by EPA, yet to be confirmed (aerosols, plastic packaging, possibly hand-held fire extinguishers) by November 15, 1992.
- Bans for HCFC use in aerosols and foams will begin January 1, 1994. Some safety and medical aerosol products and foams used for insulation are exempted.

WARNING LABELS
- Warning labels must be used for Class I compounds (CFCs, halons, carbon tetrachloride, and methyl chloroform) on all transport containers and all products containing these substances by May 15, 1993. Products made with these substances must also have warning labels if suitable alternatives exist.

- Warning labels must be used for HCFCs on all transport containers by May 15, 1993. All products containing or made with these substances that have suitable alternatives must also have labels by January 1, 2015.
- Warning labels cannot be a liability defense.

SAFE ALTERNATIVES
- The EPA will publish lists of safe alternatives for Class I and II substances and enact regulations that will maximize the use of safe alternatives in federal agencies beginning November 15, 1992.
- Alternatives for Class I and II products must reduce risk to health and the environment.

PREEMPTION
- State and local preemption is very limited, with a two year preemption over laws only governing appliance design.

CFC/HALON EXCISE TAXES
Table 2 shows current tax rate for compounds covered by the Montreal Protocol.

TABLE 2
Base Tax, $/ODP pound, 1990–1992

Year	Prev. Listed Compounds	Newly Listed Compounds
1990	1.37	–
1991	1.37	1.37
1992	1.67	1.37

Prev. Listed Compounds	ODP	Newly Listed Compounds	ODP
CFC-11	1.0	Carbon tetrachloride	1.1
CFC-12	1.0	Methyl chloroform (1,1,1-trichloroethane)	0.1
CFC-113	0.8		
CFC-114	1.0	CFC-13	1.0
CFC-115	0.6	CFC-111	1.0
Halon 1211	3.0	CFC-112	1.0
Halon 1301	10.0	CFC-211	1.0
Halon 2402	6.0	CFC-212	1.0
		CFC-213	1.0
		CFC-214	1.0
		CFC-215	1.0
		CFC-216	1.0
		CFC-217	1.0

TABLE 3
Base Tax, $/ODP pound, 1993–1995
All Listed Compounds

Effective Date	$/ODP pound	Methyl Chloroform
1993	3.35	2.11
1994	4.35	4.35
1995	5.35	5.35

Notes:

1. Reduced rate exemptions still exist for CFCs and Halons used as fire extinguishers and in insulating foams.

2. There are newly implemented reduced rates for medical sterilants and metered dose inhalers (MDIs). Base tax rate for sterilants is $1.67/ODP pound for 1993 only. Base tax for MDIs is $1.67/pound beginning in 1993.

FLOOR STOCKS TAXES

The Floor Stock Tax, which took effect January 1, 1990, is imposed on any company that has Class I compounds for sale or for use in further manufacturing. The tax rate is the incremental difference between the tax from the previous year to the current year, and is applied to year-end inventory. This tax is applied to amounts of 400 lb (181 kg) or greater per IRS employer identification number. In many cases, it does not apply to mixtures or to refrigerant used for service of the owner's systems. Mixtures must contain the ingredient as a contributor to accomplishing the purpose. Exemptions also apply to rigid foam insulation, feedstock, recycled CFCs, halons, and final products (not for resale) except if the chemicals are directly emitted.

For further information, contact the IRS.

HALON AND RIGID FOAM TAXES

Effective January 1, 1991, a new tax was imposed on Halon 1301, and CFC-11 and CFC-12 used for rigid foam. This tax is assessed at approximately $0.25 per pound through 1993. In 1994, normal base taxes apply. Floor stocks rules apply. New exemption certificates must be filed, if required.

IMPORTED PRODUCTS

The new tax rate applies to all imported products on the IRS list. There are three calculation methods used to determine the tax rate for these products: actual use data, IRS list or one percent of the product's value. A *de minimus* exemption is possible for some products.

SUMMARY

The Montreal Protocol, the Clean Air Act, and new CFC/halon taxes will all have tremendous impact on companies producing and using these products. Although some of the regulations may seem stringent, they provide us with an achievable plan:

Du Pont is prepared to support customer efforts in meeting these new regulations. The Company has dedicated resources to the safe and rapid introduction of acceptable alternatives to CFCs and to the retrofit of existing CFC equipment for their use. Du Pont has also initiated recovery, reclamation, and conservation programs to further assist customers in meeting the provisions of these new regulations.

TABLE 4

Definitions:

Class I substances are those which significantly cause or contribute to harming the ozone layer and have an Ozone Depletion Potential (ODP) greater than, or equal to, 0.2. These substances, which include all isomers, are separated into five groups:

Group I   – CFC-11, 12, 113, 114, 115

Group II – Halon 1211, 1301, 2402

Group III – Other CFCs with one, two or three carbon atoms

Group IV – Carbon tetrachloride

Group V – Methyl chloroform (except the 1,1,2 isomer)

Class II substances are those which are known, or may be reasonably anticipated, to cause or contribute to harmful effects on the ozone layer. These substances include all isomers of HCFCs having one, two or three carbon atoms.

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DUPONT

APPENDIX B

Manufacturer supplied information on:

• Thermodynamic data for Genetron 134a.

• Thermodynamic data for Genetron AZ-20

• Thermodynamic data for Genetron AZ-50

• Physical Property data for KLEA 134a

• The toxicology of KLEA 134a

• Lubricants for R-134a

• Refrigerator compressors

TECHNICAL BULLETIN

genetron®
134a

genetron® 134a

(Tetrafluoroethane)

GENETRON® 134a, a non-ozone-depleting compound, is the refrigerant of choice to replace CFC-12 in numerous air conditioning and cooling applications. It replaces CFC-122 in automobile air conditioning, residential, commercial and industrial refrigeration and in certain centrifugal chiller applications.

AlliedSignal offers the Genetron® Refrigerant Reclamation program through its Genetron Wholesalers. Now, service contractors and building owners have a safe, cost-effective means of complying with the Clean Air Act no-venting laws enacted July 1, 1992. For more information, call your Genetron Wholesaler. For the name of the nearest Genetron Wholesaler, call 1-800-522-8001.

Physical Properties:

Chemical formula	CF_3CH_2F
Molecular weight	102.03
Boiling point at 1 atm.	-15.1°F (-26.2°C)
Critical temperature	214.0°F (101.1°C)
Critical pressure, psia	589.9
Critical density, lb./cu. ft.	31.97
Liquid density at 80°F (26.7°C), lb./cu. ft.	75.0
Heat of vaporization at boiling point, Btu/lb.°F	92.4
Specific heat of liquid at 80°F (26.7°C), Btu/lb. °F	0.341
Specific heat of vapor at constant pressure (1 atm.) and 80°F (26.7°C), Btu/lb. °F	0.204
*Flammable range, % volume in air	None
Ozone depletion potential	0
Greenhouse warming potential (estimate)	0.285

Comparative Cycle Performance:

Evaporator temperature = 20°F
Condenser temperature = 110°F
Suction superheat = 30°F
Subcooling = 10°F
Compressor isentropic efficiency = 65%

genetron®	12	22	134a
Evaporator pressure, psig	21.0	43.0	18.5
Condenser pressure, psig	136.4	226.3	146.4
Compression ratio	4.23	4.17	4.86
Compressor discharge temperature, °F	188.1	227.0	178.3
Coefficient of performance	2.90	2.79	2.83
Refrigerant circulation per ton, lb./min.	3.80	2.78	3.00
Compressor displacement per ton, cfm	4.51	2.82	4.55
Liquid flow per ton, cu. in./min.	83.2	67.4	71.7
Latent heat at evaporator temp., Btu/lb.	66.5	90.6	86.9
Net refrigeration effect, Btu/lb.	52.7	72.0	66.7

*Flame limits measured using ASTM E-681 with electrically activated kitchen match ignition source per ASHRAE Standard 34.

genetron 134a Thermodynamic Table

Temp. (°F)	Pressure (psia)	Liquid Density (lb/ft³)	Vapor Volume (ft³/lb)	Enthalpy H_{liq} (Btu/lb)	Enthalpy ΔH_{vap} (Btu/lb)
-20	12.95	86.466	3.4174	5.71	93.10
-18	13.63	86.260	3.2551	6.30	92.81
-16	14.35	86.054	3.1019	6.88	92.52
-14	15.09	85.847	2.9574	7.47	92.23
-12	15.87	85.639	2.8209	8.06	91.93
-10	16.67	85.431	2.6919	8.65	91.64
-8	17.51	85.222	2.5699	9.24	91.34
-6	18.38	85.012	2.4546	9.83	91.04
-4	19.29	84.801	2.3454	10.43	90.74
-2	20.23	84.589	2.2420	11.03	90.43
0	21.20	84.377	2.1440	11.63	90.12
2	22.22	84.163	2.0512	12.23	89.81
4	23.27	83.949	1.9632	12.84	89.49
6	24.35	83.734	1.8797	13.44	89.18
8	25.48	83.518	1.8004	14.05	88.86
10	26.65	83.301	1.7251	14.66	88.53
12	27.86	83.084	1.6536	15.27	88.21
14	29.11	82.865	1.5856	15.89	87.87
16	30.41	82.645	1.5210	16.50	87.55
18	31.75	82.425	1.4595	17.12	87.21
20	33.14	82.203	1.4010	17.74	86.88
22	34.57	81.980	1.3452	18.36	86.54
24	36.05	81.757	1.2922	18.99	86.19
26	37.58	81.532	1.2416	19.61	85.85
28	39.16	81.306	1.1934	20.24	85.50
30	40.79	81.079	1.1474	20.87	85.15
32	42.47	80.851	1.1035	21.50	84.79
34	44.21	80.622	1.0617	22.14	84.43
36	45.99	80.392	1.0217	22.77	84.08
38	47.84	80.160	.9835	23.41	83.71
40	49.74	79.928	.9470	24.05	83.34
42	51.70	79.694	.9122	24.69	82.97
44	53.71	79.458	.8788	25.34	82.59
46	55.79	79.222	.8469	25.98	82.22
48	57.93	78.984	.8164	26.63	81.84
50	60.13	78.745	.7871	27.28	81.46
52	62.39	78.504	.7591	27.93	81.07
54	64.71	78.262	.7323	28.58	80.69
56	67.11	78.019	.7066	29.24	80.29
58	69.57	77.774	.6820	29.90	79.89
60	72.09	77.527	.6584	30.56	79.49
62	74.69	77.279	.6357	31.22	79.09
64	77.36	77.030	.6140	31.88	78.69
66	80.09	76.778	.5931	32.55	78.27
68	82.90	76.525	.5731	33.22	77.86
70	85.79	76.271	.5538	33.89	77.44
72	88.75	76.014	.5353	34.56	77.02
74	91.79	75.756	.5175	35.24	76.59
76	94.90	75.496	.5004	35.91	76.17
78	98.09	75.234	.4840	36.59	75.73
80	101.37	74.971	.4682	37.27	75.30
82	104.73	74.705	.4530	37.96	74.85
84	108.16	74.437	.4383	38.65	74.40
86	111.69	74.167	.4242	39.33	73.96
88	115.30	73.895	.4106	40.03	73.49
90	118.99	73.621	.3975	40.72	73.04
92	122.78	73.344	.3849	41.42	72.57
94	126.65	73.065	.3728	42.12	72.10
96	130.62	72.784	.3610	42.82	71.62
98	134.68	72.500	.3497	43.52	71.15

genetron 134a Thermodynamic Table (continued)

Enthalpy H_{vap} (Btu/lb)	Entropy S_{liq} (Btu/lb °F)	Entropy S_{vap} (Btu/lb °F)
98.81	.0133	.2250
99.11	.0146	.2247
99.40	.0159	.2244
99.70	.0172	.2242
99.99	.0185	.2239
100.29	.0198	.2236
100.58	.0212	.2234
100.87	.0225	.2231
101.17	.0238	.2229
101.46	.0251	.2227
101.75	.0264	.2224
102.04	.0277	.2222
102.33	.0290	.2220
102.62	.0303	.2218
102.91	.0316	.2216
103.19	.0329	.2214
103.48	.0342	.2212
103.76	.0355	.2210
104.05	.0368	.2208
104.33	.0381	.2206
104.62	.0393	.2205
104.90	.0406	.2203
105.18	.0419	.2201
105.46	.0432	.2200
105.74	.0445	.2198
106.02	.0458	.2196
106.29	.0470	.2195
106.57	.0483	.2194
106.85	.0496	.2192
107.12	.0509	.2191
107.39	.0521	.2189
107.66	.0534	.2188
107.93	.0547	.2187
108.20	.0560	.2186
108.47	.0572	.2184
108.74	.0585	.2183
109.00	.0598	.2182
109.27	.0610	.2181
109.53	.0623	.2180
109.79	.0636	.2179
110.05	.0648	.2178
110.31	.0661	.2177
110.57	.0673	.2176
110.82	.0686	.2175
111.08	.0698	.2174
111.33	.0711	.2173
111.58	.0724	.2172
111.83	.0736	.2171
112.08	.0749	.2170
112.32	.0761	.2170
112.57	.0774	.2169
112.81	.0786	.2168
113.05	.0799	.2167
113.29	.0811	.2166
113.52	.0824	.2166
113.76	.0836	.2165
113.99	.0849	.2164
114.22	.0861	.2163
114.44	.0873	.2162
114.67	.0886	.2162

Temp. (°F)	Pressure (psia)	Liquid Density (lb/ft3)	Vapor Volume (ft3/lb)	Enthalpy H_{liq} (Btu/lb)	Enthalpy ΔH_{vap} (Btu/lb)	Enthalpy H_{vap} (Btu/lb)	Entropy S_{liq} (Btu/lb °F)	Entropy S_{vap} (Btu/lb °F)
100	138.83	72.213	.3388	44.23	70.66	114.89	.0898	.2161
102	143.07	71.924	.3283	44.94	70.17	115.11	.0911	.2160
104	147.42	71.632	.3181	45.65	69.68	115.33	.0923	.2159
106	151.86	71.338	.3083	46.37	69.17	115.54	.0936	.2159
108	156.40	71.040	.2988	47.09	68.66	115.75	.0948	.2158
110	161.04	70.740	.2896	47.81	68.15	115.96	.0961	.2157
112	165.79	70.436	.2807	48.54	67.63	116.17	.0973	.2156
114	170.64	70.129	.2722	49.26	67.11	116.37	.0986	.2156
116	175.59	69.819	.2639	50.00	66.57	116.57	.0998	.2155
118	180.65	69.506	.2559	50.73	66.03	116.76	.1011	.2154
120	185.82	69.189	.2481	51.47	65.49	116.96	.1023	.2153
122	191.11	68.868	.2406	52.21	64.94	117.15	.1036	.2152
124	196.50	68.543	.2333	52.96	64.37	117.33	.1048	.2151
126	202.00	68.215	.2263	53.71	63.80	117.51	.1061	.2150
128	207.62	67.882	.2195	54.46	63.23	117.69	.1074	.2150
130	213.36	67.545	.2129	55.22	62.64	117.86	.1086	.2149
132	219.22	67.203	.2065	55.98	62.05	118.03	.1099	.2148
134	225.19	66.857	.2003	56.75	61.44	118.19	.1112	.2147
136	231.29	66.506	.1942	57.52	60.83	118.35	.1124	.2145
138	237.51	66.151	.1884	58.30	60.21	118.51	.1137	.2144
140	243.86	65.789	.1827	59.08	59.58	118.66	.1150	.2143
142	250.33	65.422	.1772	59.86	58.94	118.80	.1162	.2142
144	256.94	65.050	.1719	60.65	58.29	118.94	.1175	.2141
146	263.67	64.671	.1667	61.45	57.62	119.07	.1188	.2139
148	270.54	64.286	.1616	62.25	56.94	119.19	.1201	.2138
150	277.54	63.895	.1567	63.06	56.25	119.31	.1214	.2137
152	284.67	63.496	.1519	63.87	55.55	119.42	.1227	.2135
154	291.95	63.090	.1473	64.70	54.82	119.52	.1240	.2133
156	299.37	62.676	.1428	65.52	54.10	119.62	.1253	.2132
158	306.93	62.254	.1384	66.36	53.35	119.71	.1266	.2130
160	314.64	61.823	.1341	67.20	52.58	119.78	.1279	.2128

genetron-134a Thermodynamic Formulas

T_c=213.980 °F P_c=589.871 psia ρ_c=31.9702 lb./cu.ft. T_b=-15.08 °F   MWt.=102.030

Vapor pressure correlated as:

$$\ln(P_{vap}/\text{psia}) = A + \frac{B}{T} + CT + DT^2 + \left(\frac{E\,(F-T)}{T}\right) \ln\,(F-T); \; T \text{ Rankine}$$

A=22.98993635 B=-7243.87672 C=-0.013362956 D=0.692966E-05 E=.1995548 F=674.72514

Liquid density correlated as:

$$\rho(\text{lb./cu.ft.}) = \rho_c + \sum_{i=1}^{4} Di(1 - \frac{T}{T_c})^{i/3}$$

D_1=51.1669818 D_2=63.8999897 D_3=-72.213814 D_4=49.3004419

ρ_c =31.9702477

Ideal gas heat capacity correlated as:

$$C_p^0(\text{Btu/lb. R}) = C_1 + C_2 + C_2T + C_3T^2 + \frac{C_5}{T}; T \text{ Rankine}$$

C_1=0.0012557213 C_2=0.00043742894 C_3=-0.1487126E-06 C_5=6.802105688

Martin-Hou PVT Equation:

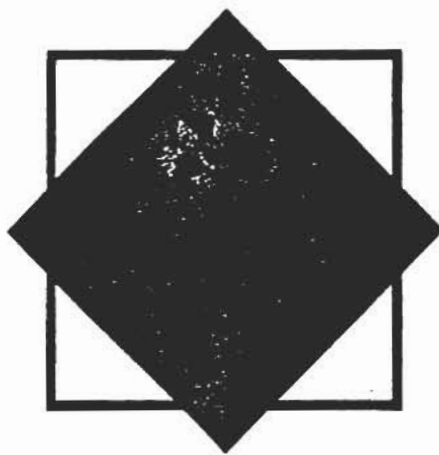$$P = \frac{RT}{(v-b)} + \sum_{i=2}^{5} \frac{A_i + B_i T + C_i e^{(-KT_r)}}{(v-b)^i}$$

P (psia), v (cu.ft./lb.), T(R), $T_r = T/T_c$

R = 0.105180
b = 0.005535126747
K = 0.5474999905E+01

i	A_i	B_i	C_i
2	-4.447445323	.002352000740	-131.4300642
3	.08630832505	-.296165168E-04	3.856548532
4	-.001001713054	0	0
5	-.106369059E-05	.107907448E-07	-.000313783768

TECHNICAL BULLETIN

genetron®

AZ-20

AZ-20 AZEOTROPE
(Difluoromethane/Pentafluoroethane)

AZ-20 is a non-ozone-depleting, non-segregating azeotropic mixture of HFC-32 and HFC-125. It has been primarily designed to replace HCFC-22 in residential air conditioning applications.

AlliedSignal offers the Genetron® Refrigerant Reclamation program through its Genetron Wholesalers. Now, service contractors and building owners have a safe, cost-effective means of complying with the Clean Air Act no-venting laws enacted July 1, 1992. For more information, call your Genetron Wholesaler. For the name of the nearest Genetron Wholesaler, call 1-800-522-8001.

Physical Properties:

Chemical formula	CH_2F_2/CF_3CF_2H
Molecular weight...........................	67.26
Boiling point at 1 atm	$-62.5°F$
Critical temperature.......................	163.8°F
Critical pressure, psia	733.2
Critical density, lb./cu. ft.	29.9
Liquid density at 80°F (26.7°C), lb./cu. ft.	64.4
Heat of vaporization at boiling point, Btu/lb.	123.3
Specific heat of liquid at 80°F (26.7°C), Btu/lb. °F	0.405
Specific heat of vapor at constant pressure (1 atm) and 80°F (26.7°C), Btu/lb. °F	0.203
***Flammable range, % volume in air	Nonflammable
Ozone depletion potential	0

Comparative Cycle Performance:

Evaporator temperature = 45°F
Condenser temperature = 120°F
Suction superheat = 10°F
Subcooling = 10°F
Compressor isentropic efficiency = 65%

genetron®	AZ-20**	22	502	125	32
Evaporator pressure, psig ...	132.3	76.0	88.7	106.7	132.9
Condenser pressure, psig ...	420.9	259.9	282.7	345.4	429.7
Compression ratio.........	2.96	3.03	2.88	2.97	3.01
Compressor discharge temperature, °F	186.1	192.4	158.3	143.4	222.8
Coefficient of performance ..	3.37	3.62	3.40	3.05	3.41
Refrigerant circulation per ton, lb./min.	2.70	2.95	4.61	5.74	1.92
Compressor displacement per ton, cfm	1.25	1.84	1.89	1.82	1.16
Liquid flow per ton, cu. in./min.	79.2	73.2	113.6	149.6	60.6
Latent heat at evaporator temp., Btu/lb.............	98.6	85.7	60.8	55.3	131.3
Net refrigeration effect, Btu/lb................	74.2	67.8	43.4	34.8	104.3

*U.S. Patent #4,978,467
**Azeotrope consisting of 60% HFC-32 and 40% HFC-125. Preliminary information based on estimated properties.
***Flame limits measures using ASTM E-681 with electrically activated kitchen match ignition source per ASHRAE Standard 34.

AZ-20 Thermodynamic Table

Temp. (°F)	Pressure (psia)	Liquid Density (lb/ft³)	Vapor Volume (ft³/1b)	Enthalpy H_{liq} (Btu/lb)	Enthalpy ΔH_{vap} (Btu/lb)
-20	42.20	78.00	1.5216	6.34	115.17
-18	44.11	77.78	1.4585	6.98	114.75
-16	46.08	77.55	1.3986	7.63	114.31
-14	48.13	77.32	1.3415	8.28	113.87
-12	50.24	77.09	1.2873	8.93	113.43
-10	52.43	76.85	1.2356	9.59	112.98
-8	54.68	76.62	1.1864	10.25	112.53
-6	57.02	76.39	1.1396	10.91	122.07
-4	59.43	76.15	1.0949	11.57	111.61
-2	61.91	75.91	1.0523	12.23	111.15
0	64.48	75.67	1.0117	12.90	110.68
2	67.12	75.43	0.9729	13.57	110.20
4	69.85	75.19	0.9359	14.25	109.71
6	72.67	74.94	0.9006	14.92	109.23
8	75.57	74.70	0.8668	15.60	108.73
10	78.56	74.45	0.8345	16.28	108.23
12	81.64	74.20	0.8037	16.96	107.73
14	84.81	73.95	0.7742	17.65	107.22
16	88.07	73.70	0.7459	18.34	106.71
18	91.43	73.44	0.7189	19.03	106.19
20	94.88	73.19	0.6930	19.72	105.67
22	98.44	72.93	0.6682	20.42	105.13
24	102.09	72.67	0.6445	21.12	104.59
26	105.85	72.41	0.6217	21.82	104.05
28	109.71	72.15	0.5998	22.53	103.50
30	113.68	71.88	0.5789	23.24	102.94
32	117.76	71.62	0.5588	23.95	102.38
34	121.94	71.35	0.5395	24.67	101.81
36	126.24	71.07	0.5210	25.39	101.23
38	130.65	70.80	0.5032	26.11	100.65
40	135.18	70.53	0.4861	26.84	100.06
42	139.82	70.25	0.4697	27.57	99.46
44	144.59	69.97	0.4539	28.30	98.86
46	149.47	69.68	0.4387	29.04	98.24
48	154.48	69.40	0.4241	29.78	97.63
50	159.62	69.11	0.4101	30.52	97.00
52	164.88	68.82	0.3965	31.27	96.63
54	170.28	68.53	0.3835	32.03	95.72
56	175.80	68.23	0.3710	32.79	95.07
58	181.46	67.93	0.3589	33.55	94.41
60	187.26	67.63	0.3472	34.32	93.74
62	193.19	67.33	0.3360	35.09	93.06
64	199.27	67.02	0.3252	35.86	92.38
66	205.48	66.71	0.3147	36.65	91.68
68	211.85	66.39	0.3046	37.43	90.98
70	218.35	66.08	0.2949	38.22	90.26
72	225.01	65.75	0.2855	39.02	89.53
74	231.82	65.43	0.2765	39.83	88.79
76	238.78	65.10	0.2677	40.64	88.04
78	245.90	64.77	0.2592	41.45	87.29
80	253.18	64.43	0.2510	42.27	86.52
82	260.61	64.09	0.2431	43.10	85.73
84	268.21	63.75	0.2355	43.94	84.93
86	275.98	63.40	0.2281	44.78	84.13
88	283.91	63.04	0.2209	45.63	83.30
90	292.01	62.68	0.2140	46.49	82.47
92	300.29	62.32	0.2073	47.36	81.61
94	308.73	61.95	0.2008	48.24	80.74
96	317.36	61.57	0.1945	49.12	79.86
98	326.16	61.19	0.1884	50.02	78.96

AZ-20 Thermodynamic Table (continued)

Enthalpy H_{vap} (Btu/lb)	Entropy S_{liq} (Btu/lb °F)	Entropy S_{vap} (Btu/lb °F)	Temp. (°F)	Pressure (psia)	Liquid Density (lb/ft³)	Vapor Volume (ft³/lb)	Enthalpy H_{liq} (Btu/lb)	Enthalpy ΔH_{vap} (Btu/lb)	Enthalpy H_{vap} (Btu/lb)	Entropy S_{liq} (Btu/lb °F)	Entropy S_{vap} (Btu/lb °F)
121.51	0.0147	0.2766	100	335.15	60.80	0.1825	50.92	78.05	128.97	0.1022	0.2417
121.73	0.0161	0.2759	102	344.32	60.41	0.1767	51.84	77.11	128.95	0.1038	0.2411
121.94	0.0176	0.2752	104	353.68	60.00	0.1712	52.76	76.16	128.92	0.1054	0.2405
122.15	0.0190	0.2746	106	363.23	59.60	0.1657	53.70	75.19	128.89	0.1070	0.2399
122.36	0.0205	0.2739	108	372.97	59.18	0.1605	54.65	74.19	128.84	0.1086	0.2393
122.57	0.0219	0.2732	110	382.90	58.75	0.1554	55.61	73.18	128.79	0.1103	0.2387
122.78	0.0234	0.2725	112	393.03	58.32	0.1504	56.59	72.14	128.73	0.1119	0.2381
122.98	0.0248	0.2719	114	403.37	57.88	0.1456	57.59	71.07	128.66	0.1136	0.2375
123.18	0.0263	0.2712	116	413.90	57.43	0.1409	58.59	69.99	128.58	0.1153	0.2369
123.38	0.0277	0.2706	118	424.64	56.96	0.1364	59.62	68.87	128.49	0.1170	0.2362
123.58	0.0292	0.2699	120	435.59	56.49	0.1319	60.66	67.73	128.39	0.1187	0.2356
123.77	0.0306	0.2693	122	446.75	56.01	0.1276	61.73	66.55	128.28	0.1205	0.2349
123.96	0.0320	0.2687	124	458.12	55.51	0.1234	62.18	65.34	128.15	0.1223	0.2343
124.15	0.0335	0.2680	126	469.71	54.99	0.1193	63.92	64.10	128.02	0.1241	0.2336
124.33	0.0349	0.2674	128	481.52	54.47	0.1153	65.05	62.82	127.87	0.1260	0.2329
124.51	0.0364	0.2668	130	493.56	53.92	0.1113	66.21	61.49	127.70	0.1279	0.2322
124.69	0.0378	0.2662	132	505.81	53.36	0.1075	67.41	60.11	127.52	0.1299	0.2315
124.87	0.0392	0.2656	134	518.30	52.78	0.1037	68.63	58.69	127.32	0.1318	0.2307
125.05	0.0407	0.2650	136	531.02	52.17	0.1000	69.89	57.21	127.10	0.1339	0.2299
125.22	0.0421	0.2644	138	543.97	51.54	0.0964	71.20	55.67	126.87	0.1360	0.2291
125.39	0.0435	0.2638	140	557.16	50.88	0.0929	72.55	54.06	126.61	0.1382	0.2283
125.55	0.0450	0.2632	142	570.59	50.19	0.0893	73.95	52.37	126.32	0.1404	0.2275
125.71	0.0464	0.2626	144	584.26	49.47	0.0859	75.42	50.59	126.01	0.1428	0.2266
125.87	0.0478	0.2621	146	598.18	48.69	0.0824	76.97	48.69	125.66	0.1453	0.2257
126.03	0.0493	0.2615	148	612.36	47.87	0.0790	78.60	46.68	125.28	0.1479	0.2247
126.18	0.0507	0.2609	150	626.78	46.98	0.0755	80.33	44.52	124.85	0.1506	0.2236
126.33	0.0521	0.2603	152	641.46	46.02	0.0702	82.21	42.15	124.36	0.1536	0.2225
126.48	0.0535	0.2598	154	656.40	44.95	0.0685	84.26	39.54	123.80	0.1568	0.2213
126.62	0.0550	0.2592	156	671.61	43.74	0.0648	86.55	36.60	123.15	0.1605	0.2199
126.76	0.0564	0.2587	158	687.08	42.33	0.0608	89.20	33.16	122.36	0.1646	0.2183
126.90	0.0578	0.2581	160	702.82	40.57	0.0564	92.46	28.87	121.33	0.1698	0.2164
127.03	0.0593	0.2575									
127.16	0.0607	0.2570									
127.28	0.0621	0.2564									
127.41	0.0636	0.2559									
127.52	0.0650	0.2553									
127.64	0.0665	0.2548									
127.75	0.0679	0.2543									
127.86	0.0693	0.2537									
127.96	0.0708	0.2532									
128.06	0.0722	0.2526									
128.15	0.0737	0.2521									
128.24	0.0751	0.2515									
128.33	0.0766	0.2510									
128.41	0.0781	0.2505									
128.48	0.0795	0.2499									
128.55	0.0810	0.2494									
128.62	0.0825	0.2489									
128.68	0.0839	0.2483									
128.74	0.0854	0.2478									
128.79	0.0869	0.2472									
128.83	0.0884	0.2467									
128.87	0.0899	0.2461									
128.91	0.0914	0.2456									
128.93	0.0929	0.2450									
128.96	0.0945	0.2445									
128.97	0.0960	0.2439									
128.98	0.0975	0.2434									
128.98	0.0991	0.2428									
128.98	0.1006	0.2422									

AZ-20 Thermodynamic Formulas

T_c=163.76 °F P_c=733.159 psia ρ_c=29.9468 lb./cu.ft. T_b=−62.482 °F MWL=67.264 kg/kmol

Vapor pressure correlated as:

$$\ln(P_{vapor}) = A + \frac{B}{T} + CT + DT^2 + \left(\frac{E\,(F-T)}{T}\right) \ln (F-T)$$

where: P_{vapor} is in psia, T is in °R

A=0.1901947141E+02 B=−0.521653680E+04 C=−0.107701611E−01 D=0.6843454187E−05 E=0 F=0

Liquid density correlated as:

$$\rho = \rho_c + \sum_{(i=1)}^{4} Di(1 - \frac{T}{T_c})^{i/3}$$

where: ρ are in lb/ft.³, T & T_c are in °R

D_1=0.5481930643E+02 D_2=0.1580236741E+02 D_3=0.2124438410E+02 D_4=0.8594503424E+01
ρ_c =0.2994684605E+02

Ideal gas heat capacity correlated as:

$$C_p^o = \sum_{(i=1)}^{4} C_{pi}T^{(i-1)} + \frac{C_{p5}}{T}$$

where: C_p^o is in Btu/lb °R, T is in °R

C_1=0.9112455865E−01 C_2=0.2149035804E−03 C_3=0.2836463297E−07 C_4=0 C_5=0
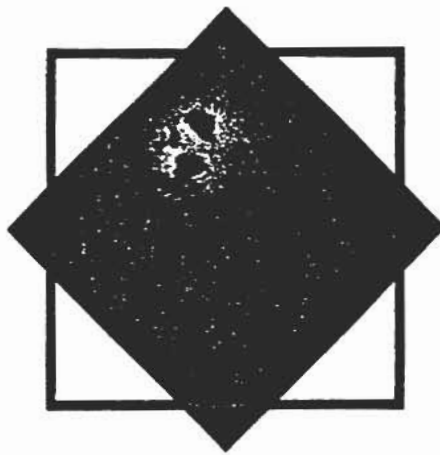
Martin-Hou PVT Equation of State:

$$P = \frac{RT}{(v-b)} + \sum_{i=2}^{5} \frac{A_i + B_i T + C_i e^{(-K \frac{T}{T_c})}}{(v-b)^i}$$

where: P is in psia, v is in ft³./lb. and T is in °R

R = 0.159542 b = 0.5152850271E−02 K = 0.5474999905E+01

i	A_i	B_i	C_i
2	−0.7172855554E+01	0.4167135193E−02	−0.1996216073E+03
3	0.3331429759E+00	−0.3420353982E−03	0.6618079798E+01
4	−0.2020678245E−02	0	0
5	−0.3390760734E−04	0.7745573305E−07	−0.7822029984E−03

TECHNICAL BULLETIN

genetron®

AZ-50

AZ-50 AZEOTROPE
(Pentafluoroethane/Trifluoroethane)

AZ-50 is a non-ozone-depleting, non-segregating azeotropic mixture of HFC-125 and HFC-143a. Initially, it has been designed to replace R-502 in low-temperature commercial refrigeration applications such as supermarket freezer and display cases.

AlliedSignal offers the Genetron® Refrigerant Reclamation program through its Genetron Wholesalers. Now, service contractors and building owners have a safe, cost-effective means of complying with the Clean Air Act no-venting law enacted July 1, 1992. For more information, call your Genetron Wholesaler. For the name of the nearest Genetron Wholesaler, call 1-800-522-8001.

Physical Properties:

Chemical Formula	CHF_2CF_3/CH_3CF_3
Molecular Weight	97.1
Boiling Point @ 1 Atm (°F)	-50.5
Critical Temperature (°F)	160.3
Critical Pressure (Psia)	538.5
Critical density (lb./cu. ft.)	30.6
Saturated Liquid Density @ 80°F (lb./cu. ft.)	64.1
Heat of Vaporization at Boiling Point (B/lb)	89.7
Specific Heat of Liquid @ 80°F (B/lb °F)	0.36
Specific Heat of Vapor @ Constant Pressure	0.21
(Cp @ 80°F and 1 Atm, B/lb °F)	
Flammable range, % volume in air	Nonflammable
(based on ASHRAE Standard 34 with match ignition)	
Ozone Depletion Potential	0

Comparative Cycle Performance

Evaporator temperature: -25°F
Condenser temperature: 100°F
Return gas @ 65°F
Degrees subcooling: 10°F
Compressor isentropic efficiency: 65%

genetron®	AZ-50	502	22
Evaporator pressure, psig	13.5	12.1	7.4
Condenser pressure, psig	238.6	216.2	195.9
Compression ratio	8.98	8.62	9.52
Compressor discharge temperature, °F	248.2	268.7	352.3
Coefficient of performance	1.65	1.67	1.68
Refrigerant circulation per ton, lb./min.	3.23	3.66	2.52
Compressor displacement per ton, cfm	6.41	6.67	7.24
Liquid flow per ton, cu. in./min.	89.6	85.8	59.8
Latent heat at evaporator temp., Btu/lb.	84.0	71.4	98.1
Net refrigeration effect, Btu/lb.	61.9	54.6	79.5

AZ-50 Thermodynamic Table

Temp. (°F)	Pressure (Psia)	Density (lb/ft³)	Vapor Volume (ft³/lb)	H_{fg} (B/lb)	Enthalpy ΔH_{ap} (B/lb)
-40	19.48	79.91	2.2450	0.00	87.35
-38	20.51	79.69	2.1381	0.73	86.90
-36	21.58	79.47	2.0374	1.46	86.45
-34	22.69	79.24	1.9426	2.19	86.00
-32	23.84	79.02	1.8531	2.92	85.55
-30	25.04	78.80	1.7687	3.65	85.09
-28	26.28	78.57	1.6890	4.37	84.65
-26	27.56	78.34	1.6136	5.10	84.20
-24	28.90	78.12	1.5424	5.83	83.74
-22	30.28	77.89	1.4751	6.55	83.30
-20	31.71	77.66	1.4113	7.28	82.84
-18	33.19	77.43	1.3509	8.00	82.39
-16	34.73	77.20	1.2937	8.72	81.94
-14	36.31	76.96	1.2395	9.44	81.50
-12	37.95	76.73	1.1880	10.16	81.05
-10	39.64	76.49	1.1391	10.88	80.59
-8	41.39	76.26	1.0927	11.60	80.14
-6	43.19	76.02	1.0487	12.32	79.69
-4	45.06	75.78	1.0067	13.04	79.24
-2	46.98	75.54	0.9669	13.75	78.79
0	48.96	75.30	0.9289	14.47	78.33
2	51.00	75.06	0.8928	15.18	77.89
4	53.11	74.81	0.8584	15.89	77.44
6	55.27	74.57	0.8256	16.60	76.99
8	57.51	74.32	0.7943	17.31	76.54
10	59.81	74.07	0.7645	18.02	76.09
12	62.17	73.82	0.7360	18.73	75.64
14	64.60	73.57	0.7088	19.44	75.18
16	67.11	73.31	0.6828	20.14	74.74
18	69.68	73.06	0.6580	20.85	74.28
20	72.32	72.80	0.6342	21.55	73.83
22	75.04	72.55	0.6115	22.25	73.39
24	77.83	72.29	0.5898	22.95	72.94
26	80.70	72.02	0.5690	23.66	72.47
28	83.65	71.76	0.5491	24.36	72.02
30	86.67	71.49	0.5300	25.05	71.58
32	89.77	71.23	0.5116	25.75	71.12
34	92.95	70.96	0.4941	26.45	70.67
36	96.22	70.69	0.4772	27.15	70.21
38	99.56	70.41	0.4611	27.84	69.76
40	103.00	70.14	0.4455	28.54	69.30
42	106.51	69.86	0.4306	29.24	68.83
44	110.12	69.58	0.4163	29.93	68.38
46	113.82	69.30	0.4025	30.63	67.91
48	117.60	69.01	0.3893	31.32	67.45
50	121.48	68.72	0.3766	32.01	66.99
52	125.45	68.43	0.3643	32.71	66.52
54	129.52	68.14	0.3525	33.40	66.06
56	133.68	67.85	0.3411	34.10	65.58
58	137.94	67.55	0.3302	34.79	65.11
60	142.30	67.25	0.3196	35.49	64.63
62	146.76	66.94	0.3094	36.19	64.15
64	151.33	66.64	0.2996	36.88	63.68
66	156.00	66.33	0.2901	37.58	63.19
68	160.77	66.01	0.2810	38.28	62.70
70	165.66	65.69	0.2722	38.98	62.21
72	170.65	65.37	0.2637	39.69	61.70
74	175.76	65.05	0.2554	40.39	61.21
76	180.98	64.72	0.2475	41.10	60.70
78	186.32	64.39	0.2398	41.81	60.18
80	191.77	64.05	0.2323	42.52	59.67
82	197.35	63.71	0.2251	43.23	59.15
84	203.04	63.36	0.2181	43.95	58.62
86	208.86	63.01	0.2114	44.67	58.08
88	214.81	62.66	0.2049	45.40	57.53
90	220.89	62.30	0.1985	46.13	56.98
92	227.09	61.93	0.1924	46.86	56.42
94	233.43	61.56	0.1864	47.60	55.85
96	239.90	61.18	0.1807	48.35	55.26
98	246.51	60.80	0.1750	49.10	54.67

AZ-50 Thermodynamic Table (continued)

H_{vap} (B/lb)	S_{liq} (B/lb °F)	S_{vap} (B/lb °F)
87.35	0.0000	0.2082
87.63	0.0018	0.2078
87.91	0.0035	0.2075
88.19	0.0052	0.2072
88.47	0.0069	0.2069
88.74	0.0086	0.2066
89.02	0.0103	0.2064
89.30	0.0119	0.2061
89.57	0.0136	0.2058
89.85	0.0153	0.2056
90.12	0.0169	0.2053
90.39	0.0185	0.2051
90.66	0.0202	0.2049
90.94	0.0218	0.2046
91.21	0.0234	0.2044
91.47	0.0250	0.2042
91.74	0.0266	0.2040
92.01	0.0281	0.2038
92.28	0.0297	0.2036
92.54	0.0313	0.2034
92.80	0.0328	0.2032
93.07	0.0343	0.2031
93.33	0.0359	0.2029
93.59	0.0374	0.2027
93.85	0.0389	0.2026
94.11	0.0404	0.2024
94.37	0.0419	0.2023
94.62	0.0434	0.2021
94.88	0.0448	0.2020
95.13	0.0463	0.2018
95.38	0.0478	0.2017
95.64	0.0492	0.2016
95.89	0.0507	0.2014
96.13	0.0521	0.2013
96.38	0.0535	0.2012
96.63	0.0549	0.2011
96.87	0.0563	0.2010
97.12	0.0577	0.2009
97.36	0.0591	0.2008
97.60	0.0605	0.2007
97.84	0.0619	0.2006
98.07	0.0633	0.2005
98.31	0.0646	0.2004
98.54	0.0660	0.2003
98.77	0.0673	0.2002
99.00	0.0687	0.2001
99.23	0.0700	0.2000
99.46	0.0713	0.1999
99.68	0.0727	0.1999
99.90	0.0740	0.1998
100.12	0.0753	0.1997
100.34	0.0766	0.1996
100.56	0.0779	0.1995
100.77	0.0792	0.1995
100.98	0.0806	0.1994
101.19	0.0819	0.1993
101.39	0.0831	0.1992
101.60	0.0844	0.1991
101.80	0.0857	0.1991
101.99	0.0870	0.1990
102.19	0.0883	0.1989
102.38	0.0896	0.1988
102.57	0.0909	0.1987
102.75	0.0922	0.1986
102.93	0.0935	0.1985
103.11	0.0948	0.1985
103.28	0.0961	0.1984
103.45	0.0974	0.1983
103.61	0.0987	0.1982
103.77	0.1000	0.1981

Temp. (°F)	Pressure (Psia)	Density (lb/ft³)	Vapor Volume (ft³/lb)	H_{liq} (B/lb)	Enthalpy ΔH_{vap} (B/lb)	H_{vap} (B/lb)	Entropy S_{liq} (B/lb °F)	S_{vap} (B/lb °F)
100	253.27	60.41	0.1696	49.86	54.07	103.93	0.1013	0.1980
102	260.16	60.01	0.1643	50.62	53.46	104.08	0.1027	0.1978
104	267.20	59.61	0.1592	51.40	52.82	104.22	0.1040	0.1977
106	274.39	59.19	0.1542	52.18	52.18	104.36	0.1054	0.1976
108	281.73	58.77	0.1493	52.97	51.53	104.50	0.1067	0.1975
110	289.22	58.34	0.1446	53.77	50.86	104.63	0.1081	0.1973
112	296.87	57.90	0.1400	54.59	50.16	104.75	0.1095	0.1972
114	304.68	57.45	0.1355	55.41	49.45	104.86	0.1109	0.1971
116	312.65	56.99	0.1311	56.25	48.72	104.97	0.1123	0.1969
118	320.78	56.52	0.1269	57.10	47.97	105.07	0.1137	0.1967
120	329.09	56.03	0.1227	57.97	47.19	105.16	0.1152	0.1966
122	337.56	55.53	0.1186	58.86	46.38	105.24	0.1166	0.1964
124	346.21	55.02	0.1147	59.77	45.55	105.32	0.1181	0.1962
126	355.04	54.49	0.1108	60.69	44.69	105.38	0.1197	0.1960
128	364.06	53.94	0.1070	61.65	43.78	105.43	0.1212	0.1958
130	373.25	53.37	0.1032	62.62	42.85	105.47	0.1229	0.1955
132	382.64	52.78	0.0996	63.63	41.87	105.50	0.1245	0.1953
134	392.22	52.17	0.0960	64.67	40.84	105.51	0.1262	0.1950
136	402.00	51.53	0.0924	65.75	39.76	105.51	0.1280	0.1947
138	411.98	50.86	0.0889	66.87	38.61	105.48	0.1298	0.1944
140	422.17	50.15	0.0854	68.05	37.39	105.44	0.1317	0.1940
142	432.56	49.40	0.0820	69.28	36.10	105.38	0.1337	0.1937
144	443.17	48.60	0.0786	70.58	34.71	105.29	0.1358	0.1933
146	454.00	47.73	0.0751	71.96	33.20	105.16	0.1380	0.1928
148	465.05	46.80	0.0716	73.45	31.55	105.00	0.1404	0.1923
150	476.33	45.77	0.0681	75.07	29.72	104.79	0.1429	0.1917
152	487.84	44.61	0.0644	76.87	27.64	104.51	0.1458	0.1910
154	499.58	43.26	0.0605	78.93	25.22	104.15	0.1491	0.1902
156	511.58	41.62	0.0563	81.40	22.25	103.65	0.1530	0.1892
158	523.81	39.42	0.0511	84.69	18.20	102.89	0.1583	0.1877
160	536.31	35.06	0.0423	91.31	9.93	101.24	0.1689	0.1849

AZ-50 Thermodynamic Formulas

$T_c = 160.340\ °F$ $P_c = 538.454\ psia$ $\rho_c = 30.5897\ lb./cu.ft.$ $T_b = -50.548\ °F$ $MWt = 97.146$

Experimental vapor pressure correlated as:

$$\ln(P_{vap}/psia) = A + \frac{B}{T} + CT + DT^2 + \left(\frac{E\,(F-T)}{T}\right)\ln(F-T);\ T\ Rankine$$

$A = 0.3222381141E{+}02$ $B = -0.7569295200E{+}04$ $C = -0.3634877222E{-}01$
$D = 0.2291767446E{-}04$ $E = 0.0000000000E{+}00$ $F = 0.0000000000E{+}00$

Experimental ideal gas heat capacity correlated as:

$$C_p^0\ (Btu/lb.\ R) = C_1 + C_2T + C_3T^2 + C_4T^3 + C_5/T;\ T\ Rankine$$

$C_1 = 0.2728881192E{-}01$ $C_2 = 0.3987582461E{-}03$ $C_3 = -0.1296346452E{-}06$
$C_4 = 0.0000000000E{+}00$ $C_5 = 0.2399596951E{+}01$

Experimental liquid density correlated as:

$$\rho\ (lb./cu.ft.) = \rho_c + \sum_{i=1}^{4} D_i (1 - T_r)^{i/3}$$

$D_1 = 0.5204001872E{+}02$ $D_2 = 0.3273414337E{+}02$ $D_3 = -0.2118057886E{+}02$ $D_4 = 0.2272687143E{+}02$
$\rho_c = 0.3058967743E{+}02$

$$P = \frac{RT}{(v-b)} + \sum_{i=2}^{5} \frac{A_i + B_i T + C_i\, e^{(-KT_r)}}{(v-b)^i}$$

$P\ (psia),\ v\ (cu.ft./lb.),\ T\ (R),\ T_r = T/T_c$

$R = 0.110468$ $b = 0.5799337186E{-}02$ $K = 0.5474999905E{+}01$

i	A_i	B_i	C_i
2	-0.4663913843E+01	0.2619347200E-02	-0.1084958628E+03
3	0.2025098298E+00	-0.2053479471E-03	0.3402951257E+01
4	-0.1149541921E-02	0.0000000000E+00	0.0000000000E+00
5	-0.1726310882E-04	0.3979151156E-07	-0.3509789950E-03

TECHNICAL BULLETIN

genetron®

123

genetron® Understanding Needs. Finding Solutions.

AlliedSignal

genetron® 123

(Dichlorotrifluoroethane)

GENETRON® 123 is a very low-ozone-depleting compound that serves as a replacement to CFC-11 in centrifugal chillers.

AlliedSignal offers the Genetron® Refrigerant Reclamation program through its Genetron Wholesalers. Now, service contractors and building owners have a safe, cost-effective means of complying with the Clean Air Act no-venting laws enacted July 1, 1992. For more information, call your Genetron Wholesaler. For the name of the nearest Genetron Wholesaler, call 1-800-522-8001.

Physical Properties:

Chemical Formula	$CHCl_2CF_3$
Molecular Weight	152.9
Boiling Point @ 1 Atm (°F)	82.2
Critical Temperature (°F)	363
Critical Pressure (Psia)	540
Critical Density (lb./cu. ft.)	34.5
Saturated Liquid Density @ 86°F (lb./cu. ft.)	90.4
Heat of Vaporization at Boiling Point (B/lb)	72.9
Specific Heat of Liquid @ 86°F (B/lb °F)	0.21
Specific Heat of Vapor @ Constant Pressure	0.17
(Cp @ 86°F and 1 Atm, B/lb °F)	
Flammable range, % volume in air	Nonflammable
(based on ASHRAE Standard 34 with match ignition)	
Ozone Depletion Potential	0.016

Comparative Cycle Performance

Evaporator temperature: 35°F
Condenser temperature: 105°F
Degrees superheat @ evaporator: 0°F
Degrees subcooling: 0°F
Compressor isentropic efficiency: 75%

genetron®	123	11
Evaporator pressure, in Hg	19.5	17.2
Condenser pressure, psig	8.1	10.9
Compression ratio	4.47	4.06
Compressor discharge temperature, °F	122.8	144.0
Coefficient of performance	4.63	4.72
Refrigerant circulation per ton, lb./min.	3.29	3.01
Compressor displacement per ton, cfm	21.78	18.20
Liquid flow per ton, cu. in./min.	64.1	57.9
Latent heat at evaporator temp., Btu/lb.	76.9	81.0
Net refrigeration effect, Btu/lb.	60.7	66.4

genetron® 123 Thermodynamic Table

Temp. (°F)	Pressure (Psia)	Density (lb/ft³)	Vapor Volume (ft³/lb)	H_{liq} (B/lb)	Enthalpy ΔH_{vap} (B/lb)
0	2.00	97.71	15.9382	8.16	79.40
2	2.12	97.54	15.0945	8.58	79.26
4	2.25	97.38	14.3033	9.00	79.12
6	2.38	97.22	13.5607	9.43	78.97
8	2.52	97.05	12.8634	9.85	78.84
10	2.66	96.89	12.2082	10.27	78.70
12	2.82	96.72	11.5923	10.70	78.55
14	2.97	96.56	11.0131	11.13	78.41
16	3.14	96.39	10.4679	11.56	78.26
18	3.31	96.23	9.9545	11.99	78.11
20	3.50	96.06	9.4709	12.42	77.97
22	3.69	95.90	9.0150	12.85	77.82
24	3.88	95.73	8.5851	13.28	77.68
26	4.09	95.56	8.1794	13.72	77.52
28	4.31	95.40	7.7964	14.15	77.38
30	4.53	95.23	7.4347	14.59	77.22
32	4.77	95.06	7.0929	15.03	77.07
34	5.01	94.90	6.7697	15.46	76.92
36	5.26	94.73	6.4639	15.90	76.77
38	5.53	94.56	6.1746	16.35	76.61
40	5.80	94.39	5.9007	16.79	76.45
42	6.09	94.22	5.6412	17.23	76.30
44	6.39	94.05	5.3953	17.68	76.13
46	6.70	93.88	5.1622	18.12	75.98
48	7.02	93.71	4.9411	18.57	75.82
50	7.35	93.54	4.7312	19.02	75.65
52	7.70	93.37	4.5320	19.47	75.49
54	8.06	93.20	4.3429	19.92	75.33
56	8.43	93.03	4.1631	20.37	75.16
58	8.82	92.86	3.9922	20.82	75.00
60	9.22	92.69	3.8297	21.28	74.83
62	9.63	92.51	3.6752	21.73	74.66
64	10.06	92.34	3.5281	22.19	74.49
66	10.51	92.17	3.3880	22.64	74.33
68	10.97	91.99	3.2546	23.10	74.16
70	11.44	91.82	3.1275	23.56	73.98
72	11.94	91.65	3.0064	24.02	73.81
74	12.44	91.47	2.8908	24.49	73.63
76	12.97	91.30	2.7806	24.95	73.45
78	13.52	91.12	2.6755	25.41	73.28
80	14.08	90.94	2.5751	25.88	73.10
82	14.66	90.77	2.4793	26.35	72.91
84	15.26	90.59	2.3877	26.81	72.74
86	15.87	90.41	2.3002	27.28	72.55
88	16.51	90.24	2.2165	27.75	72.37
90	17.17	90.06	2.1366	28.22	72.19
92	17.85	89.88	2.0600	28.70	71.99
94	18.55	89.70	1.9868	29.17	71.81
96	19.27	89.52	1.9167	29.64	71.62
98	20.01	89.34	1.8496	30.12	71.43
100	20.77	89.16	1.7853	30.59	71.24
102	21.56	88.98	1.7237	31.07	71.05
104	22.37	88.79	1.6647	31.55	70.85
106	23.20	88.61	1.6081	32.03	70.66
108	24.06	88.43	1.5538	32.51	70.46
110	24.94	88.25	1.5017	32.99	70.27
112	25.85	88.06	1.4518	33.48	70.06
114	26.78	87.88	1.4038	33.96	69.86
116	27.74	87.69	1.3577	34.45	69.66
118	28.72	87.51	1.3135	34.93	69.46
120	29.74	87.32	1.2710	35.42	69.25

H_{vap} (B/lb)	S_{liq} (B/lb °F)	S_{vap} (B/lb °F)
87.56	0.0186	0.1913
87.84	0.0195	0.1911
88.12	0.0204	0.1910
88.40	0.0213	0.1909
88.69	0.0222	0.1908
88.97	0.0231	0.1907
89.25	0.0240	0.1905
89.54	0.0249	0.1904
89.82	0.0258	0.1903
90.10	0.0267	0.1903
90.39	0.0276	0.1902
90.67	0.0285	0.1901
90.96	0.0294	0.1900
91.24	0.0303	0.1899
91.53	0.0312	0.1899
91.81	0.0321	0.1898
92.10	0.0330	0.1897
92.38	0.0339	0.1897
92.67	0.0348	0.1896
92.96	0.0356	0.1896
93.24	0.0365	0.1895
93.53	0.0374	0.1895
93.81	0.0383	0.1895
94.10	0.0392	0.1894
94.39	0.0401	0.1894
94.67	0.0409	0.1894
94.96	0.0418	0.1894
95.25	0.0427	0.1894
95.53	0.0436	0.1893
95.82	0.0445	0.1893
96.11	0.0453	0.1893
96.39	0.0462	0.1893
96.68	0.0471	0.1893
96.97	0.0479	0.1893
97.26	0.0488	0.1893
97.54	0.0497	0.1893
97.83	0.0505	0.1894
98.12	0.0514	0.1894
98.40	0.0523	0.1894
98.69	0.0531	0.1894
98.98	0.0540	0.1894
99.26	0.0549	0.1895
99.55	0.0557	0.1895
99.83	0.0566	0.1895
100.12	0.0574	0.1896
100.41	0.0583	0.1896
100.69	0.0591	0.1897
100.98	0.0600	0.1897
101.26	0.0609	0.1897
101.55	0.0617	0.1898
101.83	0.0626	0.1898
102.12	0.0634	0.1899
102.40	0.0643	0.1899
102.69	0.0651	0.1900
102.97	0.0659	0.1901
103.26	0.0668	0.1901
103.54	0.0676	0.1902
103.82	0.0685	0.1903
104.11	0.0693	0.1903
104.39	0.0702	0.1904
104.67	0.0710	0.1905

Temp. (°F)	Pressure (Psia)	Density (lb/ft³)	Vapor Volume (ft³/lb)	H_{liq} (B/lb)	Enthalpy ΔH_{vap} (B/lb)	H_{vap} (B/lb)	Entropy S_{liq} (B/lb °F)	S_{vap} (B/lb °F)
122	30.78	87.13	1.2301	35.91	69.05	104.96	0.0718	0.1905
124	31.84	86.94	1.1909	36.40	68.84	105.24	0.0727	0.1906
126	32.94	86.76	1.1531	36.89	68.63	105.52	0.0735	0.1907
128	34.06	86.57	1.1168	37.38	68.42	105.80	0.0743	0.1908
130	35.21	86.38	1.0818	37.88	68.20	106.08	0.0752	0.1908
132	36.40	86.19	1.0482	38.37	67.99	106.36	0.0760	0.1909
134	37.61	86.00	1.0158	38.87	67.78	106.65	0.0768	0.1910
136	38.85	85.81	0.9847	39.36	67.57	106.93	0.0777	0.1911
138	40.13	85.61	0.9547	39.86	67.35	107.21	0.0785	0.1912
140	41.44	85.42	0.9257	40.36	67.12	107.48	0.0793	0.1913
142	42.78	85.23	0.8979	40.86	66.90	107.76	0.0802	0.1914
144	44.15	85.03	0.8710	41.36	66.68	108.04	0.0810	0.1914
146	45.56	84.84	0.8451	41.86	66.46	108.32	0.0818	0.1915
148	47.00	84.64	0.8201	42.37	66.23	108.60	0.0826	0.1916
150	48.47	84.44	0.7960	42.87	66.01	108.88	0.0835	0.1917
152	49.98	84.25	0.7728	43.38	65.77	109.15	0.0843	0.1918
154	51.53	84.05	0.7504	43.88	65.55	109.43	0.0851	0.1919
156	53.11	83.85	0.7287	44.39	65.32	109.71	0.0859	0.1920
158	54.73	83.65	0.7078	44.90	65.08	109.98	0.0867	0.1921
160	56.38	83.45	0.6876	45.41	64.85	110.26	0.0876	0.1922

genetron® 123 Thermodynamic Formulas

T_c =363.200 °F P_c =533.097 psia ρ_c =34.5257 lb./cu.ft. T_b =82.166 °F $MWt.$ =152.930

Experimental vapor pressure correlated as:

$$\ln(P_{vap}/\text{psia}) = A + \frac{B}{T} + CT + DT^2 + \frac{E(F-T)}{T} \ln(F-T); \ T \text{ Rankine}$$

A=0.2135167313E+02 B=-0.7580945477E+04 C=-0.1151736692E-01
D=0.5341983248E-05 E=0.0000000000E+00 F=0.0000000000E+00

Experimental ideal gas heat capacity correlated as:

$$C_p^0 \text{ (Btu/lb. R)} = C_1 + C_2T + C_3T^2 + C_4T^3 + C_5/T; \ T \text{ Rankine}$$

C_1=0.3627324125E-01 C_2=0.2963321983E-03 C_3=-0.1222965602E-06
C_4=0.0000000000E+00 C_5=0.0000000000E+00

Experimental liquid density correlated as:

$$\rho \text{ (lb./cu.ft.)} = \rho_c + \sum_{i=1}^{4} D_i(1-T_r)^{i/3}$$

D_1=0.5473153636E+02 D_2=0.6881690823E+02 D_3=-0.9265622670E+02 D_4=0.6699838557E+02
ρ_c =0.3452572608E+02

Estimated Martin-Hou coefficients used:

$$P = \frac{RT}{(v-b)} + \sum_{i=2}^{5} \frac{A_i + B_iT + C_i e^{(-KT_r)}}{(v-b)^i}$$

P (psia), v (cu.ft./lb.), T (R), $T_r = T/T_c$

R =0.070173 b =0.5778313758E-02 K =0.5474999905E+01

i	A_i	B_i	C_i
2	-0.3461174842E+01	0.1482683303E-02	-0.6375783935E+02
3	0.1271057059E+00	-0.9675560464E-04	0.1712913479E+01
4	-0.5983292209E-03	0.0000000000E+00	0.0000000000E+00
5	-0.7744198272E-05	0.1325431541E-07	-0.1261428005E-03
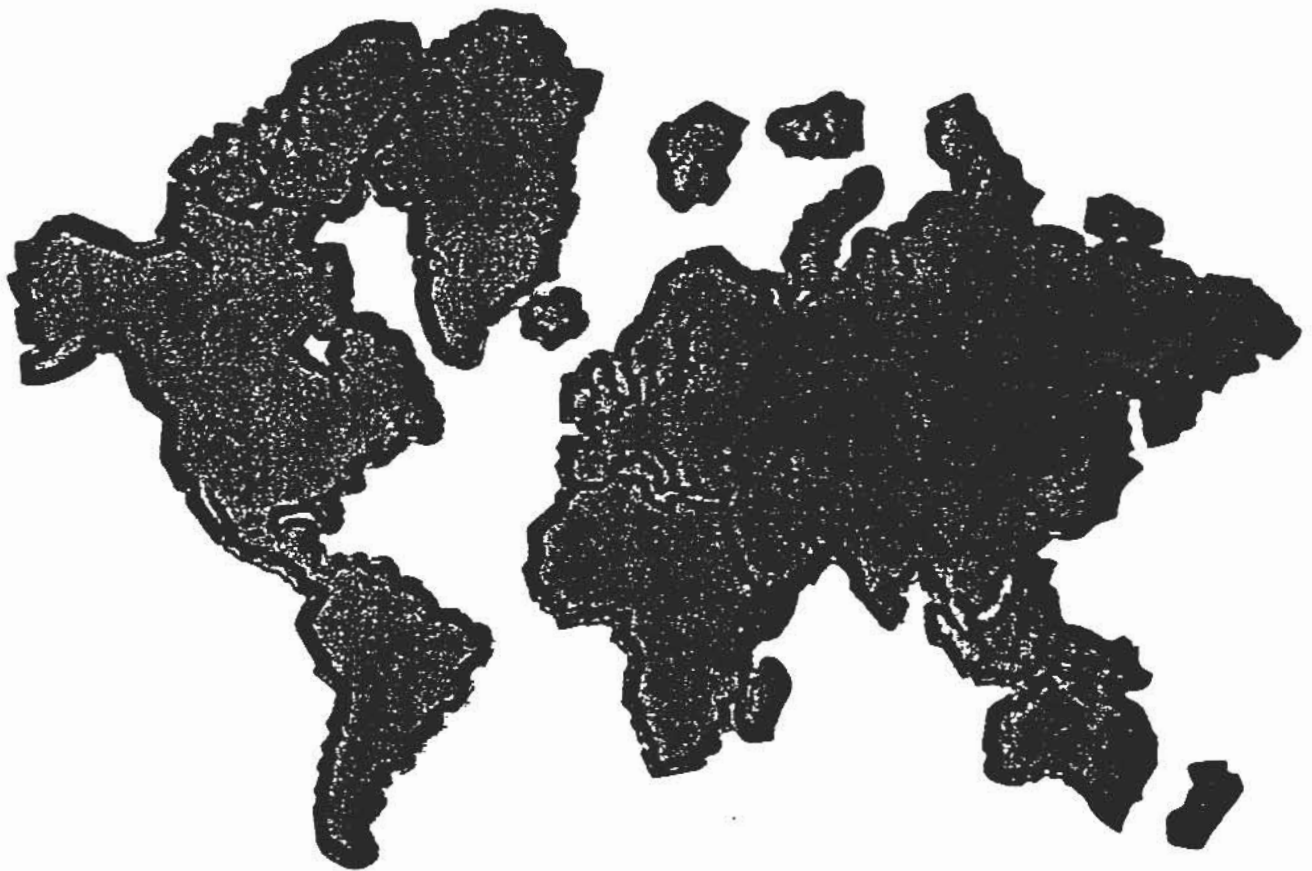
For more information contact your AlliedSignal Fluorcarbon representative.

PRODUCT INFORMATION

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800-631-8138

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genetron® Understanding Needs. Finding Solutions.

△AlliedSignal

KLEA® 134a
Fact Sheet

KLEA® 134a Is
1,1,1,2-Tetrafluoroethane CF_3CH_2F

KLEA® 134a has been thoroughly tested in a wide range of toxicological studies. The results have shown that KLEA 134a possesses extremely low toxicity and that it will be at least as safe in use as the materials that it replaces. ICI recommends an occupational exposure limit of 1000ppm as an 8 hour time-weighted average. For information on the properties and safe handling of KLEA 134a, please refer to the Material Safety Data Sheet supplied with the product or available upon request. Further details of toxicity tests and their results will also be given, if required.

The data presented here represents a combination of measurements and estimation. ICI Chemicals & Polymers Limited does not guarantee its accuracy and reserves the right to update the information in future in the light of the best available knowledge at the time.

The contents of this note are given in good faith but without any liability attaching to ICI Chemicals & Polymers Limited and it is the user's responsibility to satisfy himself that the product is entirely suitable for his purpose. Freedom from rights must not be assumed.

Physical Property Data
For KLEA® 134a*

PROPERTY		UNITS	VALUE
Molecular Weight			102.03
Boiling Point	(14.7 psia)	°F	−15.2
Melting Point		°F	−162.4
Critical Temperature		°F	213.8
Critical Pressure		psia	588.1
Critical Density		lb/cu ft	31.785
Acentric Factor			0.3256
Trouton's Constant		Btu/lb.R	0.2098
Density (liquid)	(70°F)	lb/cu ft	76.255
Density (sat vapor) at normal boiling point		lb/cu ft	0.3287
Coeff. Vol. Therm. Exp. (liquid)	(40°F - 80°F)	1/(°F)	0.001605
Specific Heat (liquid)	(70°F)	Btu/lb.R	0.336
Specific Heat (ideal gas)	(70°F)	Btu/lb.R	0.197
Latent Heat Vaporization	(70°F)	Btu/lb	77.735
Surface Tension		lb_f/in	4-911E-5
Thermal Conductivity (liquid)	(70°F)	Btu/ft.h.R	0.05007
Thermal Conductivity (sat vapor)	(70°F)	Btu/ft.h.R	7.737E-03
Vapor Pressure	(70°F)	psia	85.43
Viscosity (liquid)	(70°F)	lb/ft.h	0.531
Viscosity (vapor)	(70°F)	lb/ft.h	0.0335
Solubility in Water	(68°F, 14.7 psia)	% wt	0.0773

*Properties specified for 99.98 wt % R-134a
Standard States
Enthalpy (−40°F, liquid = 0 Btu/lb.)
Entropy (−40°F, liquid = 0 Btu/lb.R)
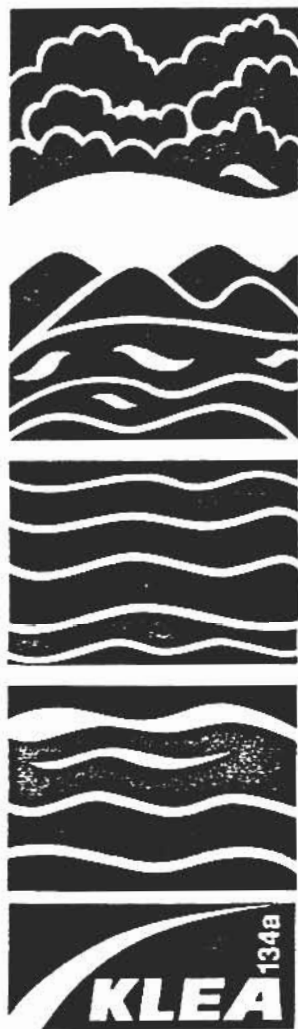
ICI is playing a leading role in the development of alternative fluorocarbons to replace the fully halogenated CFCs. A family of alternatives is being marketed by ICI under the trade name of KLEA®.

The Toxicology of KLEA® 134a

THE TOXICOLOGY OF KLEA® 134a

The first of the new ICI compounds is KLEA® 134a which is a direct replacement for CFC 12 in its major applications, especially air-conditioning and refrigeration. KLEA 134a also has potential as a replacement for CFC 11 as an insulation foam blowing agent. HFC 134a contains no chlorine and is thus ozone friendly.

The results of an extensive range of toxicity tests on KLEA 134a are now available. These tests have included a balanced package of whole animal and *in vitro* mutagenicity tests and acute, sub-chronic (90-day) and developmental toxicity inhalation studies. The findings have been subjected to a demanding review both by ICI's own team of product safety specialists and by the Program for Alternative Fluorocarbon Toxicity Testing (PAFT), an industry consortium of which ICI is a founding member. The results of the studies sponsored by PAFT were made public at the International CFC and Halon Alternatives Conference, Washington, D.C., September 29-October 1, 1992. Based on the PAFT findings and the evaluation of ICI's own experts, it has been concluded that KLEA 134a displays extremely low mammalian toxicity. A comparison of the toxicological properties of CFC 12 and HFC 134a prepared by PAFT is shown in the Appendix. US EPA, as a preliminary announcement under the Safe Alternative Policy section of the Clean Air Act Amendments of 1990, have stated that HFC 134a is acceptable for refrigeration and air-conditioning applications.

Over many years of widespread use, CFC 12 has proven to be outstandingly safe. Based on the toxicological evidence, KLEA 134a is expected to display similarly favorable properties.

On the basis of the above analysis, ICI is able to recommend the use of KLEA 134a in domestic, commercial and industrial refrigeration and air-conditioning applications and in all non-medical uses currently under consideration.

Mobile Air-Conditioning

The refrigerant is contained in equipment in the engine compartment, so that even substantial leakage should not result in significant exposure to the user in the passenger compartment. Exposures which might occur would be similar to those experienced for CFC 12. CFC 12 has a long-established safety record in mobile air-conditioning and the toxicological data indicate that this will continue to apply to KLEA 134a.

Refrigeration

For the commercial use of refrigeration equipment, the greatest possible exposure to KLEA 134a would follow accidental or deliberate damage to the enclosed system. A single short-lived exposure would occur. The results of the toxicity tests lead to the expectation that this event would be no more hazardous for KLEA 134a than for even the safest refrigerant gas it will replace.

The
Toxicology
of
KLEA® 134a
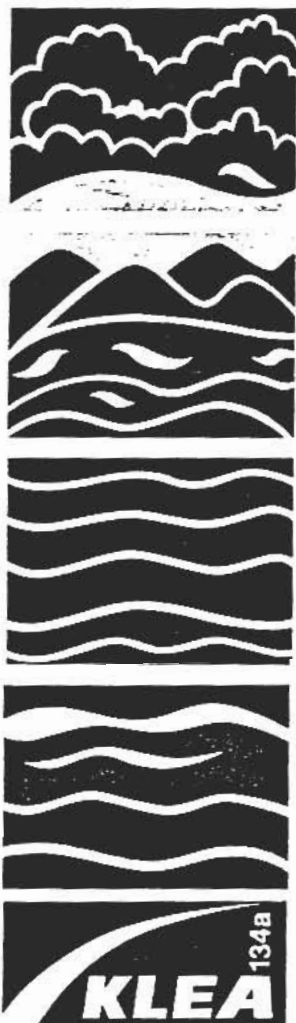
In domestic applications, the extremely unlikely circumstance of the full KLEA 134a refrigerant charge being released instantaneously would give rise to the highest conceivable exposure levels. The short-lived maximum concentration would be little or no greater than the levels identified by ICI as acceptable for long-term occupational exposures. With minimal ventilation, atmospheric concentrations would rapidly fall to low levels.

Industrial Handling

During use of KLEA 134a in the process of manufacture of refrigeration or air-conditioning equipment, ICI recommends an occupational exposure limit of 1,000 ppm as an eight-hour time-weighted average. This is the highest exposure limit set by regulatory authorities for any gas.*

Other Applications

The results of the toxicity tests indicate that KLEA 134a is acceptable for all currently proposed non-medical applications including insulation foam blowing. At this time, the specialized testing for use of KLEA 134a in medical aerosols is continuing.

Conclusion

KLEA 134a has already been more extensively tested and evaluated than many other chemicals currently in industrial use, and it exhibits a very low level of toxicological activity. Based upon these favorable toxicological properties, it is expected that the safety in use of KLEA 134a will equal or exceed that of CFC 12 which has an excellent record in air-conditioning and refrigeration. ICI's product safety team, on the basis of the extensive toxicity data, recommends the use of KLEA 134a with an occupational exposure limit of 1,000 ppm as an eight-hour time-weighted average.

*Full details for safe handling of KLEA 134a are described in the Material Safety Data Sheet (MSDS) supplied with the product.

APPENDIX

HFC 134a - Comparison With CFC 12

HFC 134a (1,1,1,2-tetrafluoroethane) is being studied as part of the PAFT* I program sector, which began in December 1987. HFC 134a is similar to CFC 12 in that it has low chemical reactivity and a high degree of stability. Both chemicals are gases.

The Toxicology of KLEA® 134a

	Boiling Point
CFC 12	-29.8°C
HFC 134a	-26.5°C

Acute Toxicity Studies
(short-term exposures to high concentrations, such as accidental leakages)

Both HFC 134a and CFC 12 are practically non-toxic by the inhalation route. The 4-hour LC_{50} for HFC 134a is greater than 500,000 ppm, and for CFC 12 it is 760,000 ppm. As with other halogenated hydrocarbons, CFC 12 and HFC 134a can, at high dose levels, sensitize the heart to adrenaline. For CFC 12, the threshold level for cardiac sensitization is 50,000 ppm, while for HFC 134a it is 75,000 ppm.

Genotoxicity Studies
(effects on genetic material; an early screen for possible carcinogenic activity)

Both HFC 134a and CFC 12 are inactive in Ames assays and in *in vitro* clastogenicity studies, both with and without metabolic activation. The genotoxicity of CFC 12 and HFC 134a *in vivo* has been investigated in both a rat cytogenetics study and in a mouse dominant lethal test at levels up to 50,000 ppm. Neither compound affected fertility or caused a mutagenic effect. Recent work has also shown HFC 134a to be negative in an *in vivo* mouse micronucleus assay and in a rat liver unscheduled DNA synthesis study. Thus, neither material has been shown to represent a significant genotoxic hazard.

Sub-Chronic Toxicity Studies
(repeated exposure to determine any overall toxicological effect)

In sub-chronic inhalation toxicity studies, HFC 134a showed no significant signs of toxicity even at an exposure level of 50,000 ppm, the highest level tested. In a similar study with CFC 12, no significant signs of toxicity were seen at an exposure level of 10,000 ppm, also the highest level tested. In addition, both compounds apparently undergo only minimal levels of metabolism in rodents.

Developmental Toxicity (Teratology) Studies
(assessment of the potential for causing birth defects)

Several teratology studies have been conducted with HFC 134a. No developmental effects were seen in inhalation studies with either pregnant rats or rabbits at levels as high as 300,000 ppm and 40,000 ppm, respectively. CFC 12 has also been shown not to cause developmental effects when tested in rats and rabbits at exposure levels up to 50,000 ppm.

*PAFT: Program for Alternative Fluorocarbon Toxicity Testing — an industry consortium undertaking the toxicity testing of potential replacements for CFCs.

Statement Released by PAFT November 1992

The Toxicology of KLEA® 134a

Chronic Toxicity/Carcinogenicity Studies
(lifetime testing to assess late-in-life toxicity or increased incidence of cancer)

HFC 134a did not cause chronic toxicity nor a carcinogenic response in an oral study when administered to groups of rats for 1 year at a dose level of 300 mg/kg/day followed by lifetime observation. Data from a chronic inhalation toxicity/carcinogenicity study is currently being evaluated. In this study rats were exposed to levels of up to 50,000 ppm of HFC 134a for 6 hours per day, 5 days per week for 2 years. The in-life phase of the study was completed in November 1991 and showed no significant adverse effects. Pathological examination has identified microscopic, benign tumors of the testes in male rats exposed to 50,000 ppm. No increases in numbers of tumors of any type were seen at 10,000 ppm exposures. The benign testes tumors occurred only near the end of the study, and were not life-threatening to the rats. The toxicological no-effect level is considered to be 10,000 ppm. CFC 12 also was evaluated in a lifetime inhalation toxicity study. In this study, groups of rats and mice were exposed to levels of up to 5,000 ppm for 6 hours per day for 2 years. Again, no significant signs of toxicity and no evidence of carcinogenicity were seen.

Summary
The PAFT I program on HFC 134a is now complete with the exception of the final reporting (expected early in 1993) of the chronic study, for which both the in-life phase and the pathology are now complete. All results available to date show that HFC 134a will be at least as safe in use as CFC 12. The results indicate that the product is safe for use in domestic, commercial, and industrial refrigeration and air conditioning applications, and in all other currently proposed industrial applications, provided that the recommended normal hygiene practices are observed. Most PAFT companies have set an occupational exposure limit for HFC 134a of 1,000 ppm (eight-hour time-weighted average). Some applications, such as medical aerosols, will require further testing.

The
Toxicology
of
KLEA® 134a
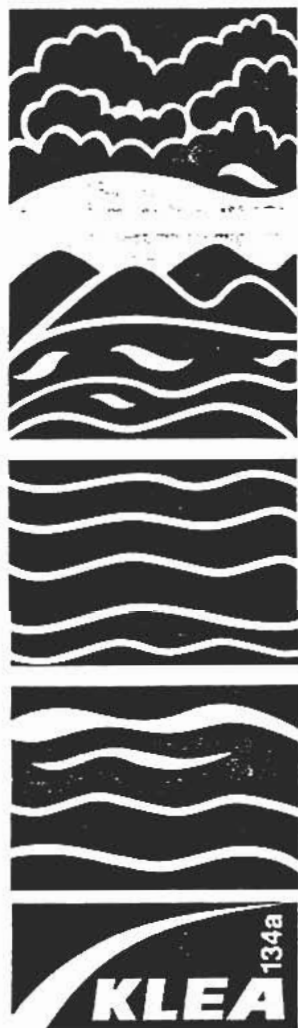
FOR FURTHER INFORMATION, CONTACT:

Fluorochemicals

ICI Americas Inc.
Chemicals and Polymers Group
Fluorochemicals Business
Wilmington, DE 19897
Telephone: (302) 886-4344

R-134a UPDATE

R-134a Lubrication Needs Being Addressed by Industry

Since R-134a is not miscible with the traditional mineral oils used as lubricants in R-12 systems, new lubricants are needed to ensure the efficiency and long term reliability of R-134a systems.

To meet this need, lubricant suppliers have been exploring and evaluating several classes of synthetic lubricants, foremost among which are polyalkaline glycols (PAGs) and polyolesters.

Polyalkaline Glycols

PAGs were the first class of synthetic lubricants to be considered for use with R-134a. As a class they showed acceptable lubricity, miscibility with R-134a, and were commercially available during the early stages of R-134a product development. As a matter of fact, they were the only choice available to the automobile industry for testing as they developed their 134a automotive air conditioning systems.

But according to Jim Lavelle, Alternative Refrigerants Product Manager at Elf Atochem North America, the industry's largest producer of CFC alternatives, several general problems exist with PAGs.

The PAGs will absorb water to quite high levels, which can lead to chemical breakdown inside the refrigeration system. This is especially a problem in situations where the product is expected to last many years, such as in appliances. Another problem is that PAGs may not be compatible with residual R-12, other chlorinated materials and mineral oils. This means that PAGs could not be used to retrofit old systems without cleaning out all remaining refrigerant and oil first. All of which led the industry to explore the esters.

Polyolester Lubricants

Esters solve the problems of PAGs. They are a different family of synthetic lubricants, heretofore used successfully as aviation lubricants. Lavelle reports that Elf Atochem, through its sister company, Elf Lubricants, has done extensive testing of a polyolester lubricant for use with R-134a, and that results are very positive. The Elf Atochem polyolester lubricant (which is being marketed under the brand name Planetelf ACD) offers a solution to the problem of water absorption and chlorine compatibility. Lavelle says it exhibits excellent chemical stability, good miscibility with R-134a, and outstanding lubricating qualities. In addition the Planetelf ACD polyolester lubricant is compatible with most materials used in refrigeration systems, and inhibits wear on the various parts inside the compressor.

In marketing Planetelf ACD lubricants, Elf Atochem will target them for use in broad R-134a refrigeration/air conditioning applications in both retrofit and new equipment.

An Understanding of R-134a Behavior With Refrigeration Oils

Refrigeration systems contain mixtures of oil and refrigerant. The separation into two layers depends on several factors: temperature; chemical nature of the oil and refrigerant; and relative concentration of the two components.

Generally, in domestic refrigeration and automotive air conditioning, complete miscibility between oil and refrigerant is required for: improving the return of oil to the compressor during the functioning of the installation; avoiding accumulation of viscous oil in low temperature parts of the installation (evaporator, expansion valve . . .); and avoiding clogging of capillary tubes.

The following chart summarizes the behavior of HFC 134a with oils. HFC 134a demonstrates very poor miscibility with the traditional mineral or synthetic oils but demonstrates excellent miscibility with both PAG and polyolester oils.

	HFC 134a
Naphthenic Mineral Oils	(–)
Paraffinic Mineral Oils	(–)
Alkylbenzene Synthetic Oils	(–)
Polyalkylene Glycol Synthetic Oils	(+)
Polyolester Synthetic Oils	(+)

(+) Good miscibility
(–) Poor miscibility

Quart Packaging Avoids Waste, Saves Costs

Users of ester lubricants in R-134a systems are finding that unused lubricant in opened 1-gallon containers becomes contaminated quickly, as it absorbs water and moisture-borne contaminants to the point where it cannot be used. This has proven costly as well as wasteful. Elf Atochem is packaging its Planetelf ACD refrigeration lubricants in 1-quart containers, and marketing 4-packs of these quarts in 1-gallon cartons. Use of the quart, where a quart, or less, of the lubricant is required, preserves the integrity of the remainder of the gallon, and makes for a more efficient purchase.

Results of PAFT Toxicity Tests on R-134a Released

The Program for Alternative Fluorocarbon Toxicity Testing (PAFT) notified the EPA on September 10, 1992 of results from the long-term toxicology testing of HFC 134a. The news is good. The tests confirmed the low toxicity of HFC 134a as a CFC alternative.

In virtually all studies on HFC 134a there were no effects as a result of exposure to the refrigerant. The only exception was in the lifetime study where at an extremely high exposure level (50 times current occupational exposure limits for HFC 134a) there was an increase in benign testicular tumors in rats. This was seen at the very end of the study and there was no effect on life expectancy. No such effect was observed at lower exposure levels. The tests appear to indicate that there are no negative implications in the use of HFC 134a in domestic, commercial and industrial refrigeration applications.

elf atochem

ATO

An Introduction To

EMKARATE RL

Refrigeration Lubricants

Chemicals & Polymers

ICI-
Meeting
The Need

The Montreal Protocol has set time scales for the phase-out of CFCs. ICI has played, and will continue to play, a leading role in assisting the refrigeration and air-conditioning industries in the transition to alternative refrigerants.

As a result of this commitment, ICI has developed its range of Emkarate® RL refrigeration lubricants for use with CFC alternatives. Designed specifically to meet the rigorous performance demands of the refrigeration industry, Emkarate RL refrigeration lubricants have evolved with over five years of intimate collaborative research with major refrigeration compressor and system manufacturers around the world.

ICI offers the global refrigeration industry world-scale production and distribution of approved refrigerants and lubricants fully complemented by a worldwide technical resource. Ongoing research and technical service activities are supported regionally by dedicated facilities in Europe, the United States and Japan.

All experimental data contained in this literature has been generated for Emkarate RL refrigeration lubricants in combination with KLEA™ 134a, the first commercially available alternative refrigerant intended as a replacement for CFC 12. Similar physical and performance data will be generated for other alternative refrigerants as they become available.

ICI is fully committed and will continue to meet the needs of the refrigeration industry in order to facilitate the transition from CFCs.

Typical Properties of Emkarate RL Refrigeration Lubricants — Application Guide

	Viscosity @ 40°C	Viscosity @ 100°C	Viscosity @ -30°C	Viscosity Index	Pour Point	Density @ 15°C	Flash Point (COC)	Acid Value	Color	Water Content	Miscibility 10% Lube in 134a High °C	Miscibility Low °C	Hermetic	Automotive	Industrial/Commercial
Units	All Measurements in cSt			No Units	°C	g/ml	°C	mgKOH/g	Hazen	ppm	High °C	Low °C			
Test Method	D445			D2270	D97	D1298	D92	In House	D1209	E203	In House				
Product Code															
Emkarate RL 7H	7.8	2.2		144	-61	0.918	188	0.02	125	50	>80	-34	●		
Emkarate RL 15 S	14.0	3.5	308	132	-65	0.961	205	0.02	80	50	>80	-37	●		
Emkarate RL 22 H	18.9	4.2	505	128	-52	0.998	240	0.02	100	50	>80	-44	●	● ●	
Emkarate RL 32 S	32.0	5.6	2310	114	-46	0.973	250	0.02	110	50	>80	-5	●	● ● ●	
Emkarate RL 46 S	48.0	7.3	3649	111	-40	0.972	275	0.02	125	50	>80	-5		●	● ● ● ● ● ●
Emkarate RL 68 S	74.1	10.1	7358	118	-35	0.978	245	0.02	<250	50	>80	-3		● ●	● ● ● ● ● ●
Emkarate RL 100 S	100	12.7		122	-36	0.979	260	0.02	125	50	75	+6			● ● ● ● ●
Emkarate RL 150 S	134	15.0	–	114	-37	0.978	265	0.02	80	50	67	+9			● ● ●
Emkarate RL 220 H	230	19.5		96	-15	0.978	280	0.02	175	50		-16			● ●

Water content of Emkarate RL refrigeration lubricants is typically less than 100 ppm. Resistivity of Emkarate RL refrigeration lubricants is typically > 1.0 Ωcm x 10¹³. Dielectric strength of Emkarate RL refrigeration lubricants is typically > 25KV. Test methods are ASTM standard methods unless otherwise stated.

Viscosity

Emkarate RL refrigeration lubricants are available in a range of viscosities to suit most refrigeration applications. The higher VI of these lubricants compared to traditional mineral oils minimizes energy consumption at low temperatures, maximizes lubrication at elevated temperatures and in some cases enables users to choose a lubricant of a lower ISO grade.

Viscosity Comparison
Mineral Oil vs Emkarate RL Refrigeration Lubricants
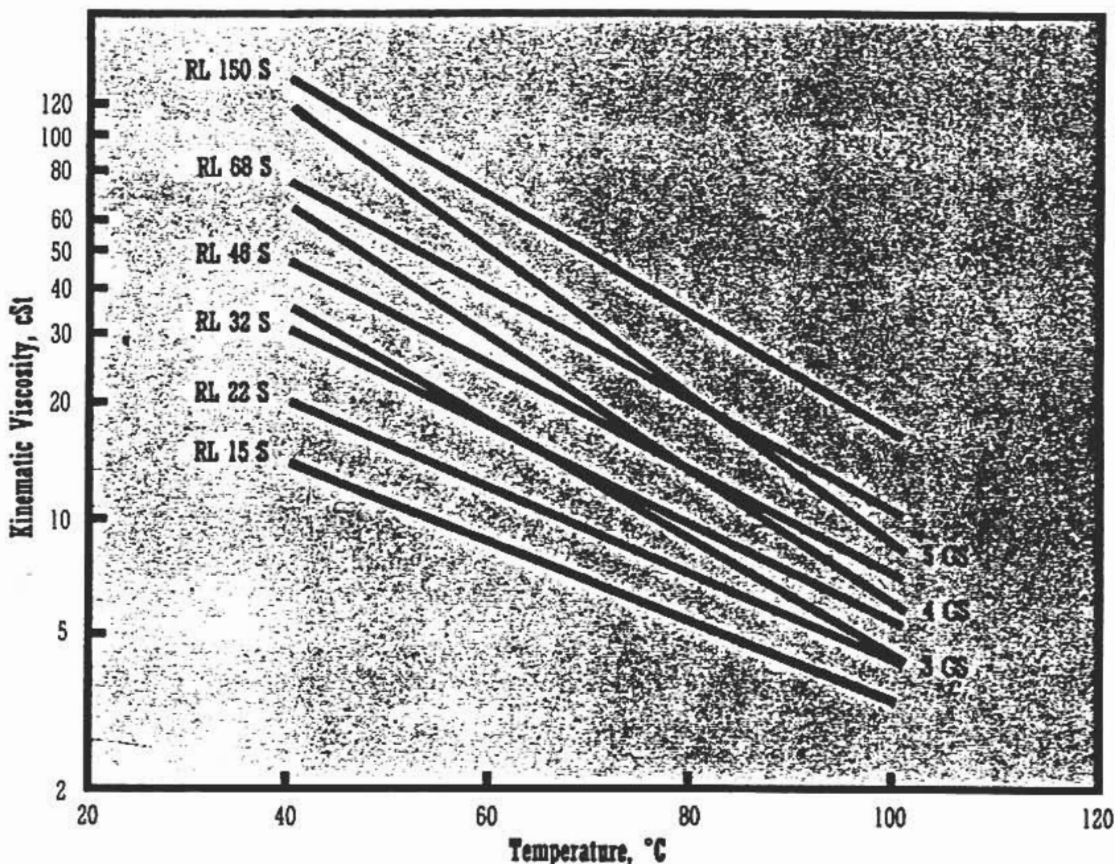
Retrofitting

Retrofit involves the conversion of a refrigeration system from CFC-12 to HFC-134a. Depending on the specific system geometry, compressor, performance required, instrumentation and materials of construction, the retrofit can be as easy as a mere refrigerant change or involve significant hardware modifications. When retrofitting a system from CFC-12 to HFC-134a, the system manufacturer should be consulted for specific recommendations.

In most cases, however, a retrofit to HFC-134a will require a change to an ester lubricant to assure proper oil transport, stability, lubrication and heat transfer performance. Emkarate RL refrigeration lubricants have undergone significant laboratory and field retrofits in conjunction with many of the world's leading refrigeration system manufacturers. As a result of this extensive experience, Emkarate RL synthetic esters have demonstrated superior stability, lubricity and performance characteristics in a wide array of systems. Additionally, ICI has developed a simple set of guidelines for retrofitting to HFC-134a, including detailed lubricant flushing procedures.

If you would like more information on converting your CFC-12 system to HFC-134a, contact ICI or Virginia KMP at the locations shown overleaf.

Customer Service and Distribution

Emkarate RL refrigeration lubricants may be ordered from any one of ICI's Sales Offices which are located around the world. The local office will also be happy to assist with any other queries regarding these products. Specialist technical assistance is also available.

Emkarate RL refrigeration lubricants can be supplied in road tank wagons or non-returnable 200 kg net mild steel drums. Smaller volumes are available from ICI's authorized distributor of Emkarate RL refrigeration lubricants, Virginia KMP.

ICI continues to develop lubricants for use with alternative refrigerants and to meet the needs of the refrigeration industry. For information on these products, please contact your local Sales Office in the first instance.

Classification

Emkarate RL refrigeration lubricants are not classed as hazardous under international transport regulations, i.e., UN, IMO, RID, ADR and ICAO/IATA.

Handling and Storage

Emkarate RL refrigeration lubricants have high flash points. They are expected to have a very low systemic toxicity. Nevertheless, good industrial practices should be used when handling these products. There should be good ventilation of working areas, and contact with the skin and eyes should be avoided by the use of protective gloves and goggles.

Any splashes in the eye(s) should be removed by washing with plenty of water; splashes on the skin should be removed by washing with soap and water.

Emkarate RL refrigeration lubricants are hygroscopic and suitable precautions should be taken to prevent moisture absorption from the air.

Emkarate RL refrigeration lubricants may be stored in mild steel tanks or drums.

If this material is redistributed or reformulated for sale, details of the methods for safe handling should be passed to all customers.

Toxicology

Information on toxicology, environmental matters and safety in handling is continually updated by ICI. This information is provided in Material Safety Data Sheets supplied with the products and separately upon request.

Enquiries should be directed to your nearest Sales Office.

Customers are urged to ensure that the product is entirely suitable for their own purposes. ICI can frequently help in these technical decisions if required.

ICI Sales Offices

ICI Americas Inc.
(Head Office) Wilmington
DE 19897 USA
Tel: (302) 886-3000
Telex: 4945649 ICI UI
Fax: (302) 886-2972

ICI Chemical Products
PO Box 90, Wilton
Middlesbrough
Cleveland TS6 8JE UK
Tel: Eston Grange (0642) 454144
Telex: 587461 ICI WWC G
Fax: (0642) 432444

Distributed by

Virginia KMP Corp.
4100 Platinum Way
Dallas, TX 75237 USA
Tel: (214) 330-7731
Toll Free: 1-800-473-5074
Fax: 1-800-877-8567

Virginia Refrigeration Ltd.
121 Blackfriars Road
Southsea
Portsmouth
Hants PO5 4NL UK
England
Tel: 0705 756365
Fax: 0705 294349

Virginia/KMP

POLYOL ESTER REFRIGERATION OIL

THE PREFERRED LUBRICANT FOR CFC, HCFC and HFC REFRIGERANT APPLICATIONS
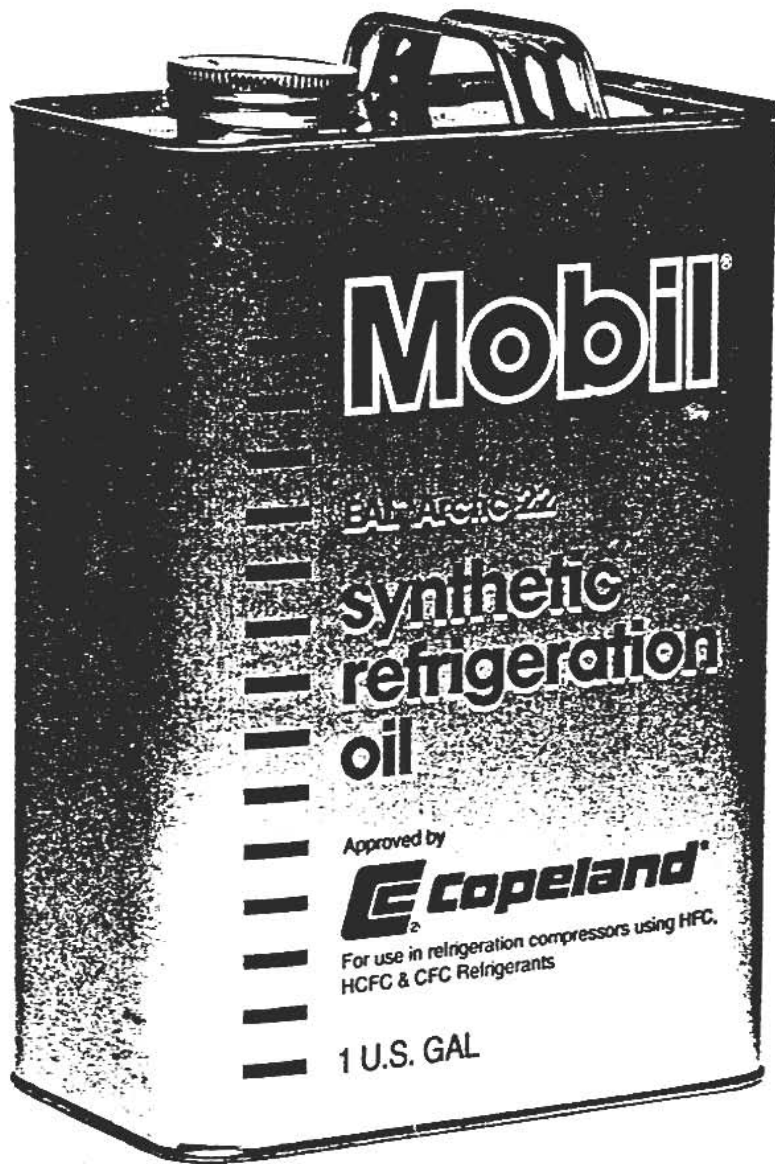
Mobil EAL Arctic 22, available from Copeland, is a synthetic lubricant formulated with selected polyol ester base stocks and additives which provide lubricity, stability and resistance to corrosion. It is part of the Mobil EAL (Environmental Awareness Lubricants) family specifically developed to lubricate refrigeration compressors and system components where HFCs are used. HFCs are ozone-friendly, chlorine-free, non-CFC refrigerants which are replacing CFCs in the world refrigeration market. The performance of Mobil EAL Arctic 22 has been well documented with HFCs in refrigeration systems. It exhibits the desired miscibility at critical temperatures, low viscosity loss, as well as stability for long system life.

Mobil EAL Arctic 22 has also been found to be completely compatible with CFC refrigerants as well as the new

Mobil®

HCFC interim "blends." This allows the lubricant to be used with virtually any traditional refrigerant. This "backward compatibility" and the superior lubricity of Mobil EAL Arctic 22 makes it the preferred lubricant for most applications.

Typical Characteristics

Physical characteristics listed in the table below are typical and may vary slightly.

Advantages

Mobil EAL Arctic 22 will provide the following benefits:

- Long lubricant life

- Controlled miscibility
- Outstanding protection against compressor wear
- High system efficiency
- Excellent low-temperature fluidity
- Excellent high-temperature stability

Health and Safety

Based on available toxicological information, these products produce no adverse effects on health when properly handled and used. No special precautions are suggested beyond attention to good personal hygiene, including laundering oil-soaked clothing and washing skin-contact areas with soap and water. Material Safety Data Bulletins are available from Copeland Corporation by calling 1-513-498-3558.

MOBIL EAL ARCTIC 22 CHARACTERISTICS

Falex Pin / V-Block Test, ASTM D 3233, lb	
Four-Ball Wear Test	
• ASTM D 4172, 20kg / 1200 rpm / 60°C / 1 hr, scar. dia., mm	0.72
Copper Corrosion, ASTM D 130, 24 hr at 100°C (212°F)	1A
Color, ASTM D 1500	0.5
Water Content, ppm	<100

Copeland®

1675 W. Campbell Road • Sidney, Ohio 45365-0669

Open-ended Problem • 72
April 1994

Castrol
icematic
sw

SYNTHETIC ESTER REFRIGERATION LUBRICANTS

Castrol have supplied compressor lubricants to the refrigeration industry since the 1940's. The original products, for use with ammonia and carbon dioxide refrigerant gases, were mineral oils. With the advent of chlorofluorocarbons (CFC) refrigerants in the late 1940's, naphthenic crude oils of low wax content were used to yield low floc point oils. As compressor design evolved, increasing the demands on the lubricant with higher discharge temperatures and speeds, new synthetic lubricants were introduced to the market place. During the 1970's Castrol introduced alkyl benzene and, in the 1980's, polyalphaolefin lubricants into their product range under the Icematic branding.

Since the advent of new non-ozone depleting hydrofluorocarbon (HFC) refrigerants, which are not compatible with normal oils, Castrol's chemists and engineers have been working to develop a new series of HFC miscible lubricants. The development thrust has been in close cooperation with leading compressor manufacturers and refrigerant gas suppliers, primarily using the first commercialised non-ozone depleting refrigerant HFC R134a, and has culminated in the Castrol Icematic SW series of refrigeration compressor lubricants.

Castrol are ideally placed to support the international refrigeration industry. Castrol companies are located in 39 countries worldwide and a distribution network is in place in a further 100 countries.

The Requirements for New Compressor Lubricants

Conventional compressor lubricants are miscible with CFC and hydrochlorofluorocarbon (HCFC) refrigerants but are immiscible with the non-ozone depleting HFC alternatives. Use of a conventional immiscible lubricant in conjunction with HFC R134a adversely affects the efficiency of the refrigeration unit. Oil plugs separate from the liquefied refrigerant within the condensor of the system, impeding the flow and causing spluttering as they pass through the restrictor (capillary tube or expansion valve) into the evaporator. Once transferred to the evaporator of the system, the immiscible oil settles at the bottom of the tubes causing further impedance to the gas flow and reducing heat transfer efficiency. In a severe case, lack of oil return to the compressor can promote component wear and eventual failure through lubricant starvation.

CIRCUIT DIAGRAM

COMPRESSOR

EVAPORATOR

Low Temperature/Low Pressure

CONDENSER

High Temperature/High Pressure

EXPANSION VALVE

EVAPORATOR

GASEOUS REFRIGERANT/IMMISCIBLE OIL

CONDENSER

LIQUID REFRIGERANT/IMMISCIBLE OIL
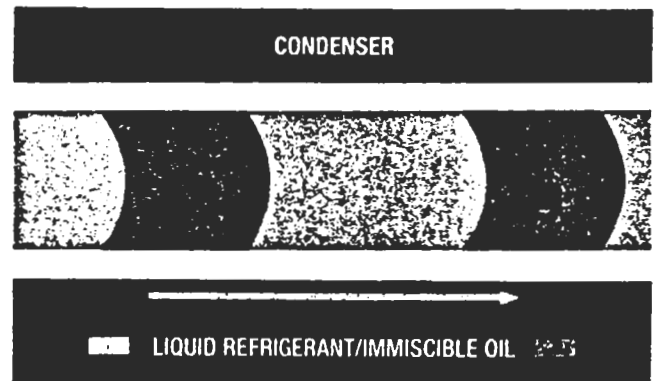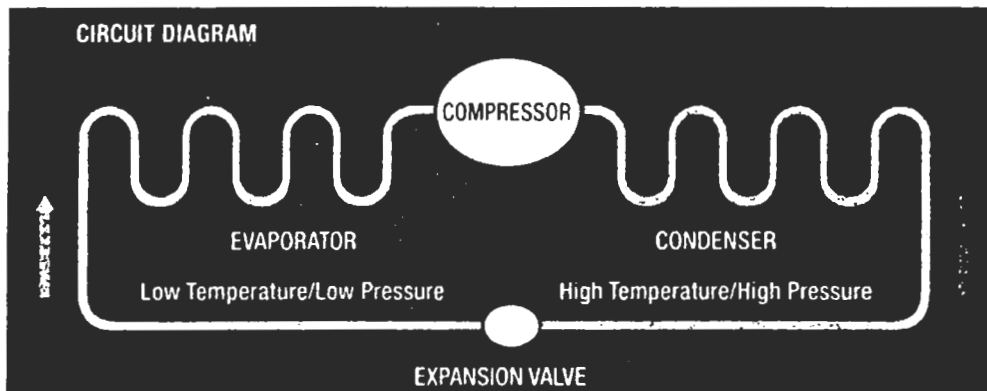
Product Technology

The Castrol Icematic SW series of lubricants is based on synthetic polyol esters and formulated with additives. The polyol ester base stocks are new components which have been specifically synthesised by Castrol for the purpose of compressor lubrication in conjunction with HFC refrigerants. Polyol ester based lubricants are the most stable within the diverse ester family of products.

Aviation Gas Turbine Lubrication is one of the most severe applications that any lubricant must endure. The polyol ester based lubricants have to withstand the temperature range of below -40°C to above +250°C where all mineral oils and many synthetic products would fail.

Castrol is a major supplier of synthetic aviation lubricants and has drawn on this experience in developing the Castrol Icematic SW range.

The Use of Additives

Castrol have formulated a low treat, ashless and highly effective additive package into the Icematic SW product range to provide important performance benefits over non-formulated products. These benefits include :

- No copper plating of compressor components
- Increased hydrolytic stability
- Reduced compressor wear

Typical Physical Properties and Compressor Applications

Castrol Icematic SW refrigeration lubricants have excellent physical properties. The Viscosity Index (the relationship between the lubricant's viscosity at 40°C and 100°C) is typically above 100, which is significantly higher than conventional naphthenic mineral oils which are usually less than 20. This means that the

Icematic SW lubricants are thinner at low temperatures, reducing compressor power consumption and improving oil return from the evaporator. Conversely, at high operating temperatures they retain their viscosity better than naphthenic oils, providing a strong boundary film lubrication to the compressor, even under the most arduous temperatures experienced during high ambient temperature conditions. The lubricants have similar pour points and considerably higher flash points than comparable viscosity naphthenic oils.

TEST	METHOD	CASTROL ICEMATIC						
		SW10	SW22	SW32	SW68	SW100	SW150	SW220
Kinematic Viscosity, 40°C (cSt)	ASTM D445	10	22	32	68	100	150	220
Kinematic Viscosity, 100°C (cSt)	ASTM D445	2.7	4.7	5.7	8.8	11.4	15.1	19.3
Pour Point (°C)	ASTM D97	<-60	-60	-54	-39	-30	-29	-26
Closed Flash Point (°C)	ASTM D93	190	230	245	250	255	260	290
Total acid number (mgKOH/g)	ASTM D664	0.15	0.15	0.15	0.15	0.15	0.15	0.15
Hydroxyl number (mgKOH/g)		2	2	2	2	2	2	2
Water (ppm)		50	50	50	50	50	50	50
Colour	Apha	300	300	300	300	300	300	300

Application : Domestic/Industrial/Air Conditioning

Compressor Type:

	SW10	SW22	SW32	SW68	SW100	SW150	SW220
Reciprocating - Hermetic	■	■					
Reciprocating - Semi-hermetic		■	■	■			
Rotary		■	■	■			
Scroll			■				
Centrifugal			■				
Screw				■	■	■	■

Application : Mobile/Automotive Air Conditioning

Compressor Type:

	SW10	SW22	SW32	SW68	SW100	SW150	SW220
Reciprocating - Semi-Hermetic			■		■		
Rotary Vane					■		
Swash Plate					■	■	■

Miscibility with Refrigerant Gases

Castrol Icematic SW lubricants utilise new polyol ester basestocks that have been specifically synthesised to provide excellent miscibility with HFC R134a over a wide temperature range. The products have been tested with many refrigerant gases and found to be miscible with most CFC, HCFC and HFC R134a.

The lower critical solution temperature (the point above which all proportions of oil and refrigerant are miscible) for Icematic SW products with HFC R134a are comparable to, and in many cases better than, naphthenic oils with CFC R12.

REFRIGERANT TYPE	LOWER CRITICAL SOLUTION TEMPERATURE (°C)				
	R12 CFC	R502 CFC/HCFC	R22 HCFC	R134a HFC	R125 HFC
Icematic SW 10	<-50	<-50	<-50	<-50	<-50
Icematic SW 22	<-50	<-50	<-50	-39	<-50
Icematic SW 32	<-50	<-50	<-50	<-50	<-50
Icematic SW 68	<-50	<-50	<-50	-40	<-50
Icematic SW 100	<-50	<-50	<-50	-33	●
Icematic SW 150	<-50	<-50	<-50	-28	●
Icematic SW 220	<-50	<-50	<-50	-22	●

● Higher viscosity products are not suitable for this refrigerant when used in isolation due to floc formation above the miscibility temperature of the oil and refrigerant mixture.

Castrol Icematic SW Miscibility curves with HFC 134a

Note: Icematic SW10 and SW32 miscible at all proportions to <- 60°C
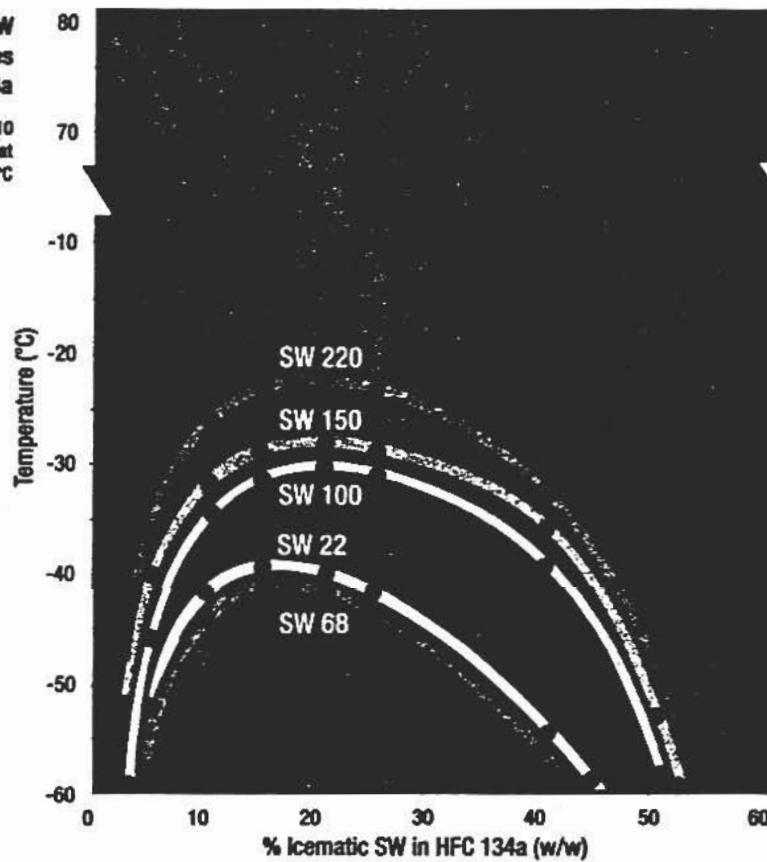
heated to 95°C and bubbled with oxygen (30cc/min), with copper and steel catalysts immersed in the emulsion. The Icematic SW products are very resistant to hydrolysis. The acidity rise was restricted to 7mgKOH/g after completion of the 1000 hour test period. The non-formulated base stock suffered a rapid rise in acidity to 160mgKOH/g after only 500 hours when the test was terminated. The true level of breakdown is likely to be considerably higher than this, considering that many of the acid breakdown products are volatile.

Hygroscopicity

Castrol Icematic SW lubricants are more hygroscopic than naphthenic mineral oils. They saturate at approximately 1000ppm from atmospheric moisture, compared to about 100ppm for mineral oils. The Icematic SW lubricants are considerably less hygroscopic than polyalkylene glycol lubricants (the first generation of oils developed for use with HFC R134a) which saturate in excess of 1% water (10,000 ppm).

Due to the hygroscopic nature of the Castrol Icematic SW lubricants, care should be taken in handling the product to minimise atmospheric exposure and retain its dried (typically 50ppm water) condition.

Water Ingress Into Refrigeration Lubricants. Open vessel, agitated, 70% RH.

Polyalkylene Glycol

Castrol Icematic SW

Naphthenic Oil

Water (%)

Time (Hours)
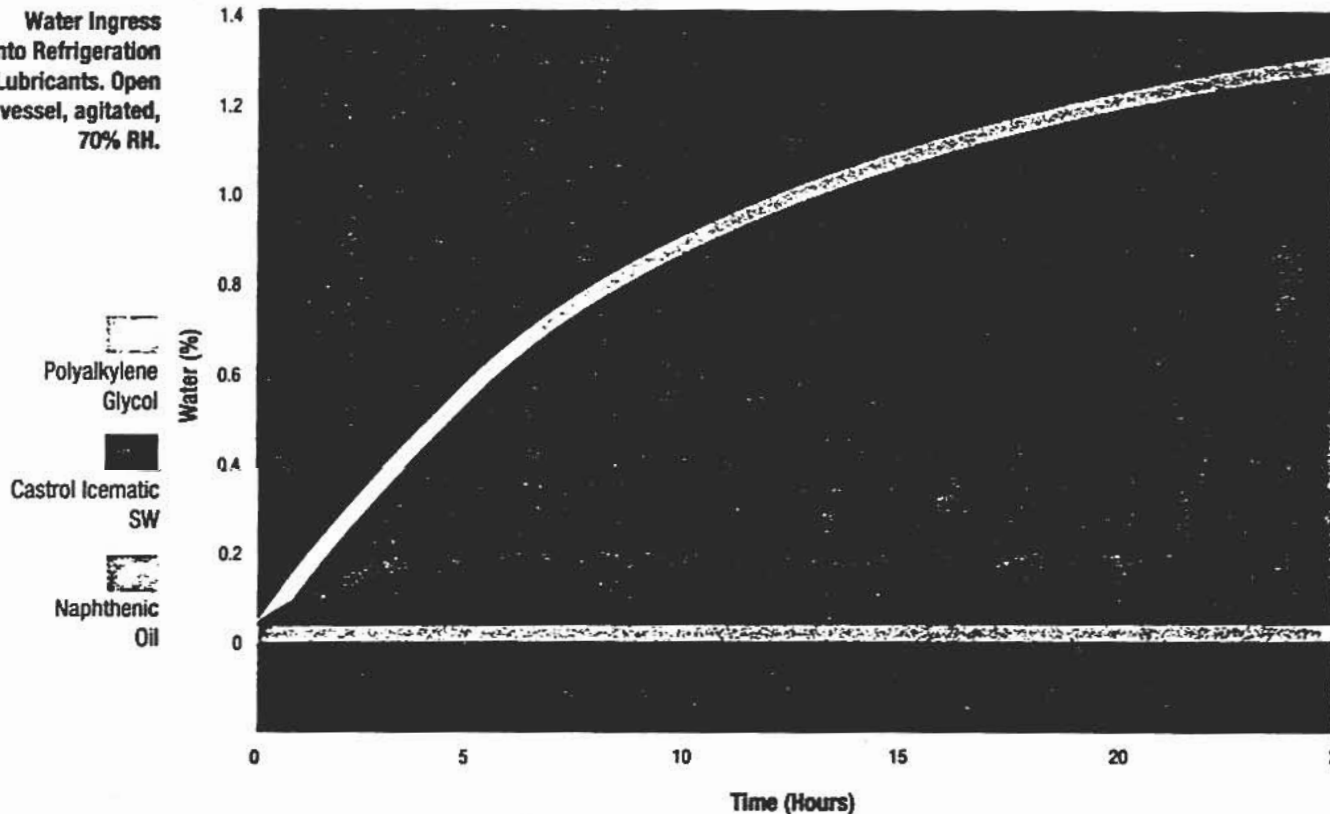
Thermal Stability

The Ashrae 97 sealed tube stability test was developed to assess the thermal stability of refrigerant gas. The refrigeration industry has adopted the test in a modified form to assess the stability of lubricants with the new HFC refrigerants. Castrol Icematic SW lubricants have been extensively screened in this test, in which metal specimens typical of compressor construction have been introduced, both in-house and by compressor manufacturers. The metal specimens suffer no degradation after the 2 week test at 175°C, in conjunction with a variety of refrigerants. The lubricant retained its original colour and acidity levels at, or slightly below, that of the unused oil.

Ashrae 97 Sealed Tube Stability Test 175°C/14 Days

CASTROL ICEMATIC	REFRIGERANT	METAL SPECIMEN CONDITION		
		Copper	Steel	Aluminium
SW 68	HFC R134a	Bright	Bright	Clean
SW 68	CFC R12	Bright	Bright	Clean
' SW 68	HCFC R22	Bright	Bright	Clean

Hydrolytic Stability

The hydrolytic stability of any ester based lubricant is essential for the long term durability of the refrigeration system, particularly as relatively high water levels can be present within the working unit.

The Icematic SW lubricants have been subjected to the TOST thermal and oxidative stability test, where an oil/water emulsion is
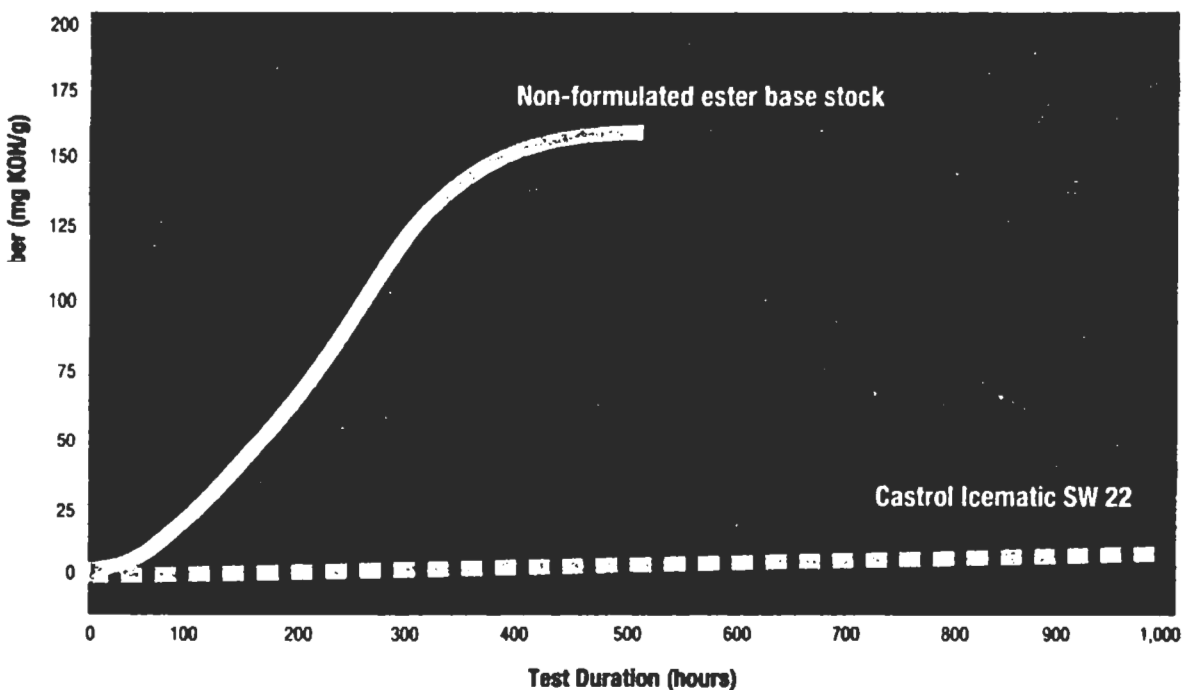
Hydrolytic Stability of Castrol Icematic SW lubricants TOST ASTM D 943

Pet Insulation Compatibility

PET insulation material is used in insulating the motor winding of compressors and is in contact with both the refrigerant gas and lubricant. Tests have shown that the Castrol Icematic SW lubricants perform similarly to naphthenic oils on PET elasticity and tensile strength. HFC R134a can reduce the elasticity of PET insulation. The manufacturers are developing low oligomer content variants to minimise this effect.

TEST OIL/REFRIGERANT (conditions: 60°/1000 hours)	TENSILE STRENGTH (MPa)	ELONGATION (EB%)
CFC 12	145	65.7
HFC 134a	138	43.0
Icematic SW 22	138	53.5
Icematic SW 22+ CFC 12	139	53.7
Icematic SW 22+ HFC 134a	139	47.0
ISO VG 32 Naphthenic oil	143	51.7

Wear Tests

Chlorinated additives have for many years been used to provide reduced wear on internal working components. Similarly, chlorinated refrigerants have provided excellent anti-wear properties in refrigeration compressors. With the elimination of chlorine in non-ozone depleting refrigerants, this wear protection is no longer available and has to be borne by the lubricant alone. HFC refrigerants are indeed detrimental to component wear, thinning the lubricant whilst providing no wear protection.

The Castrol Icematic SW lubricants, thanks to their balanced formulation of additives and base stock, possess excellent anti-wear performance characteristics, giving negligible steel wear in bench tests. This has been successfully translated into excellent compressor results of low wear in rapid life tests and in the field.

Falex - Rotating Pin on Loaded Vee Blocks

TEST OIL	REFRIGERANT	SEIZURE LOAD (lb)	WEAR 30 min/400lb (mg)
ISO VG 100 naphthenic	None	750	30
Non-formulated ester	None	1,150	16
Icematic SW 68	None	1,800	2
ISO VG 100 naphthenic	CFC R12	900	6
Non-formulated ester	CFC R12	1,400	4
Icematic SW 68	CFC R12	2,300	<1
Non-formulated ester	HFC R134a	1,500	36
Icematic SW 68	HFC R134a	2,400	<1

FZG gear test results have been generated for centrifugal compressor manufacturers where the gear box can share the same lubricant as the compressor.

FZG Gear Test Results

TEST OIL	FAILURE LOAD STAGE
Icematic SW 32	10
Icematic SW 68	12

Elastomers

Elastomer compatibility of the lubricant and refrigerant is critical for open compressor systems. The shaft seal 'O' ring should swell slightly under the effect of lubricant and refrigerant to ensure an effective seal against the positive pressure within the compressor. Should the seal suffer excessive shrinkage or swelling, leakage will occur and the refrigerant charge will be lost from the system.

Castrol recommends that only compressors fitted with Nitrile Butadiene (NBR), Hydrogenated Nitrile Butadiene (HNBR) or Neoprene rubbers be used with HFC R134a and Icematic SW lubricants. This is particularly important when converting an existing CFC or HCFC system to HFC R134a, to ensure that the shaft seal is compatible. If it is not a new one must be fitted.

Conversion of Existing CFC Refrigeration Plant to HFC Refrigerants

With the impending phase out of CFC refrigerants, users have only two options to adopt longer term alternative refrigerants :

1)   Switch to HCFC R22
2)   Convert existing equipment to HFC refrigerants

The initial preferred route (in the absence of a suitable conversion procedure to adopt HFC refrigerants) was to switch existing equipment using harmful CFC refrigerants to HCFC R22. This approach entails considerable cost to the user in terms of hardware, requiring at least a new compressor and condenser. HCFC refrigerants are ozone depleting, albeit less so than CFC's, and can therefore only be regarded as a short term solution.

In response to this, in 1990 Castrol successfully pioneered a unique procedure for converting existing CFC-12 refrigeration systems to the new non-ozone depleting HFC 134a. This is called the "Castrol Retrofill Procedure" and a patent has been applied for.

The "Castrol Retrofill Procedure" has proved to be the most practicable and economical approach and has been widely adopted by the refrigeration industry. Written procedure documents are available for several refrigeration applications.

The procedure entails the necessary removal of the original mineral lubricant, which is not compatible with R134a. The system is flushed of its original oil by draining the compressor and then recharging with Castrol Icematic SW. The system then runs normally with its original CFC-12 refrigerant. The "Castrol Retrofill Procedure" is environmentally friendly as it totally eliminates the need for the traditional CFC-11 solvent flush method of cleansing systems and utilises lubricants which are biodegradable.

During the flushing cycle, a fully compatible system is maintained as the Castrol Icematic SW lubricant is entirely miscible with the original oil and refrigerant. Through normal operation of the refrigeration cycle, the residual mineral oil retained within the system is returned to the compressor sump. The number of flushes and the time required for the flushing stage are dependant on the size and application of the original refrigeration equipment but has never exceeded three.

**Castrol Icematic SW Lubricants
"Castrol Retrofill Procedure"**
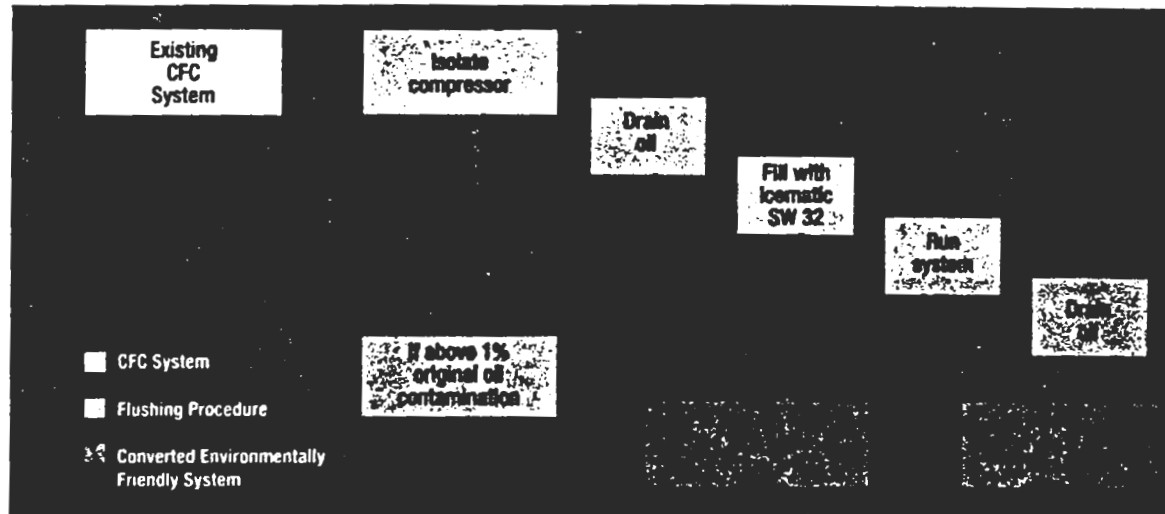
Through repeat oil changes with Castrol Icematic SW, the residual mineral oil contamination is reduced to the recommended level of below 1%. Residual mineral oil within the system will adversely affect unit efficiency as miscibility of the oil mixture and R134a at low temperatures is reduced correspondingly according to the level of contamination. The Castrol Icematic SW series has excellent low temperature miscibility with R134a and can therefore tolerate a higher level of contamination than competitor lubricants.

**Castrol Icematic SW
Refrigeration Oils
Miscibility of Icematic SW/
Naphthenic mixtures with
R134a
(10% Oil in R134a)**
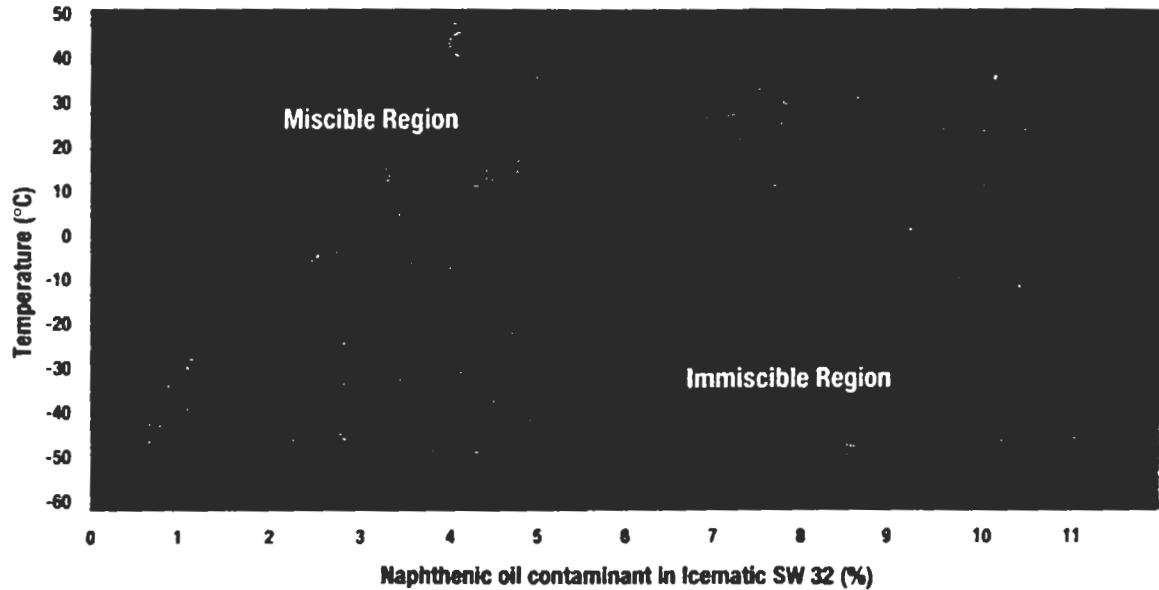
The formulated Castrol Icematic SW products have been extensively tested and have proved to be fully compatible with

chlorinated refrigerants and solvents. No copper plating problems have been experienced with Castrol Icematic SW. We therefore have no need to advise maximum level of contamination from chlorinated residues. Indeed, systems can undergo the oil changes now, as a preparation for conversion, and run with the original R12 until such time as R134a becomes readily available and similarly priced. This is expected to be in 1995.

Toxicity

The Castrol Icematic SW lubricants are listed on EINECS and TSCA registers and classified as being of no significant hazard. The products are despatched with a Material Safety Data Sheet (MSDS), which provides information on handling and what to do in case of spillage.

Castrol Customer Service

Castrol Icematic SW lubricants can be ordered from any Castrol company throughout the world. Representatives from your local Castrol company will be happy to provide advice and assistance relating to the Icematic SW series of refrigeration lubricants. Castrol companies offer a full support package to customers, including technical assistance and routine returned oil analysis.

Castrol is committed to serving the refrigeration industry with the highest quality compressor lubricants in order to ease the transition to environmentally friendly refrigerants. Cooperative agreements are in place with several refrigerant gas manufacturers, ensuring that Castrol remains at the forefront of refrigeration lubricant technology, and linking the development of new lubricants to new alternative refrigerants as they themselves are developed. These agreements enable Castrol to maintain an overview of the research and new developments in refrigerants across a wide cross-section of the chemical industry.

Distribution

Castrol companies will supply Icematic SW products directly to compressor manufacturers and designated distributors. The service sector will have access to the products through a variety of sources; Castrol, OEM and refrigerant gas manufacturer distribution networks, and larger independent wholesale companies.

Castrol Icematic SW lubricants can be supplied in bulk tankers, 55 gallon non-returnable mild steel drums, 5 gallon pails and 6 x 1 gallon cases.

AMERICOLD

R 12
(CFC)
HIGHEST EFFICIENCY
Hermetic Refrigeration Compressors
LOW BACK PRESSURE
APPLICATIONS

2340 SECOND AVENUE, N.W.
CULLMAN, ALABAMA 35055
FAX: (205) 739-0217   TEL: (205) 734-9160

"SSG" SERIES — HIGHER EFFICIENCY — R12

Model	Power Supply 115/60 or 100/50	Power Supply 220/240/50 or 230/60	Motor Type	Displacement Wt. CI/R	CC/R	Nominal Performance at Standard Rating Cond. † (With Air Over Compressor) BTU/HR	KCAL/HR	WATTS	AMPS	Cond. † BTU/WHR	Compressor Cooling ★	Oil Charge	Net Weight Less Oil LBS.	KG.	Compressor Height Dimension IN.	CM.
SSG106-1	X			.312	5.11	650	164	127	1.20	5.12	FAN		23	10.43	8.25	20.95
SSG107-1	X		Resistance Start (PTCR) Capacitor Run	.361	5.92	768	194	147	1.20	5.22	FAN	10.1 ozs. (300 cc)	23	10.43	8.25	20.95
SSG108-1	X			.401	6.57	867	218	164	1.35	5.28	FAN		23	10.43	8.25	20.95
SSG109-1	X			.427	6.99	943	238	176	1.57	5.36	FAN		23	10.43	8.25	20.95
SSG111-1	X			.500	8.19	1147	289	213	1.88	5.39	FAN		24	10.89	8.50	21.59
SSG112-1	X			.554	9.08	1274	314	240	2.18	5.32	FAN		24	10.89	8.50	21.59
SSG113-1	X			.597	9.78	1373	346	256	2.38	5.36	FAN		24	10.89	8.50	21.59

★ Also available with oil cooler tubes - designated as "200" series models, i.e. SSG106 becomes SSG206.
Compressor can be used with either FAN or STATIC CONDENSER.

"HG" SERIES — SUPER HIGH EFFICIENCY — R12

Model	Power Supply 115/60 or 100/50	Power Supply 220/240/50 or 230/60	Motor Type	Displacement Wt. CI/R	CC/R	Nominal Performance at Standard Rating Cond. † (With Air Over Compressor) BTU/HR	KCAL/HR	WATTS	AMPS	Cond. † BTU/WHR	Compressor Cooling ★	Oil Charge	Net Weight Less Oil LBS.	KG.	Compressor Height Dimension IN.	CM.
HG106-1	X			.277	4.50	621	156	116	1.01	5.35	FAN		23	10.43	8.25	20.95
HG107-1	X			.312	5.11	708	173	131	1.15	5.40	FAN		23	10.43	8.25	20.95
HG108-1	X		Resistance Start (PTCR) Capacitor Run	.361	5.92	848	213	154	1.33	5.50	FAN	10.1 ozs. (300 cc)	23	10.43	8.25	20.95
HG109-1	X			.401	6.57	947	238	169	1.43	5.53	FAN		23	10.43	8.25	20.95
HG110-1	X			.473	6.99	1029	259	187	1.70	5.51	FAN		24	10.89	8.50	21.59
HG111-1	X			.500	8.19	1193	300	215	1.90	5.55	FAN		24	10.89	8.50	21.59
HG112-1	X			.554	9.08	1322	333	239	2.10	5.53	FAN		24	10.89	8.50	21.59
HG113-1	X			.597	9.78	1426	359	259	2.30	5.51	FAN		24	10.89	8.50	21.59

Also available with oil cooler tubes - designated as "200" series models, i.e. HG106 becomes HG206.
Compressor can be used with either FAN or STATIC CONDENSER.

AVAILABLE LINE CONNECTIONS — I.D.†

Tube Set	Suction IN.	Suction MM	Discharge IN.	Discharge MM	Process IN.	Process MM	Oil Cooler IN.	Oil Cooler MM
"A"	.258	6.55	.258	6.55	.258	6.55	.188†	4.78
"B"	.320	8.13	.194	4.93	.258	6.55	.195	4.95

† Except Tube Set "A" Oil Cooler Dimension is O.D.
All connections are copper coated steel.

† STANDARD RATING CONDITIONS — ALL MODELS

Evaporating Refrigerant Temperature	-10F/-23.3C
Condensing Temperature	130F/54C
Ambient Temperature	90F/32C
Liquid Temperature at Expansion Device	90F/32C
Suction Gas Temperature	90F/32C

APPENDIX C

Project summary: *Life Cycle Design Guidance Manual:*
Environmental Requirements and the Product System

♻EPA   Project Summary

Life Cycle Design Manual: Environmental Requirements and the Product System

Gregory A. Keoleian and Dan Menerey

The U.S Environmental Protection Agency's (EPA) Risk Reduction Engineering Laboratory and the University of Michigan are cooperating in a project to reduce environmental impacts and health risks through product system design. The resulting framework for life cycle design is presented in *Life Cycle Design Manual: Environmental Requirements and the Product System*. Environmental requirements in life cycle design are chosen to minimize aggregate resource depletion, energy use, waste generation, and deleterious human and ecosystem health effects.

The manual adopts a systems-oriented approach based on the product life cycle. A product life cycle includes raw materials acquisition, bulk and engineered materials processing, manufacturing/assembly, use/service, retirement, and disposal. Design activities address the product system, which includes product, process, distribution, and management/information components.

Integrating environmental requirements into the earliest stages of design is a fundamental tenet of life cycle design. Concepts such as concurrent design, total quality management, cross-disciplinary teams, and total cost assessment are also essential elements of the framework. A multilayer requirements matrix is proposed to balance environmental, performance, cost, cultural, and legal requirements. The following design strategies for pollution prevention and resource conservation are presented: product life extension, material life extension, material selection, reduced material intensiveness, process

management, efficient distribution, and improved business management (which includes information provision). Environmental analysis tools for developing requirements and evaluating design alternatives are outlined.

This Project Summary was developed by the University of Michigan for the EPA's Risk Reduction Engineering Laboratory, Cincinnati, OH, to announce key findings of the research project that is fully documented in a separate report of the same title (see Project Report ordering information at back).

Overview

The purpose of the *Life Cycle Design Project* is to promote environmental impact and risk reduction through design. This project complements the EPA's *Life Cycle Assessment Project* which is developing guidelines for life cycle inventory analysis. The framework developed in this project guides designers to reduce aggregate impacts associated with their products. Successful low-impact designs must also satisfy performance, cost, cultural, and legal criteria.

Investigation of the design literature and interviews with 40 design professionals contributed to the development of a basic framework for life cycle design. The interviews were conducted to identify barriers and the information and tools needed to achieve environmental objectives. Life Cycle Design Demonstration Projects are being conducted with AT&T Bell Labs and Allied Signal to test the design framework.

A summary of the seven chapters contained in *Life Cycle Design Manual: Environ-*

mental Requirements and the Product System follows.

Chapter 1. Introduction

Most environmental impacts result from design decisions made long before manufacture or use. Yet environmental criteria often are not considered at the beginning of design when it is easiest to avoid impacts. As a result, many companies channel resources into fixing problems rather than preventing them.

In the past 15 yr, companies began to focus more on pollution prevention and resource conservation. Innovative firms are now adopting ambitious environmental policies in response to changing public perceptions. But translating these policies into successful action is a major challenge. Without proper support, environmental design programs may be launched without specific objectives, definitions, or principles.

Such practices demonstrate the need for a design framework that helps reduce total environmental impacts while satisfying other criteria. When design considers all stages of the life cycle from raw material acquisition to final disposal of residuals, the full consequences of product development can be understood and acted on.

Purpose

The manual seeks to:
- provide guidance on reducing impacts and health risks caused by product development
- encourage the inclusion of environmental requirements at the earliest stage of design rather than focusing on end-of-pipe solutions
- integrate environmental, performance, cost, cultural, and legal requirements in effective designs

Scope

Environmental requirements for product design are the main focus of the manual. In life cycle design, products are defined as systems that include the following components: product, process, distribution network (packaging and transportation), and management (including information provision). Life cycle design can be applied to:
- *improvements,* or minor modifications of existing products and processes;
- *new features* associated with developing the next generation of an existing product or process; and
- *innovations* characteristic of new designs.

No single design method or set of rules applies to all types of products. For that reason, the manual provides general guidelines rather than prescriptions. Designers should use the manual to develop tools best suited to their specific projects.

Audience

Each participant in product system development has an important role to play in achieving impact reduction. The manual is primarily targeted for the following decision makers:
- product designers
- industrial designers
- process design engineers
- packaging designers
- product development managers
- staff and managers in: accounting, marketing, distribution, corporate strategy, environmental health and safety, law, purchasing, and service

Chapter 2. Life Cycle Design Basics

Several key elements form the foundation of life cycle design. First, design takes a systems approach based on the life cycle framework. Every activity related to making and using products is included in design. As a result, the product is combined with processing, distribution, and management to form a single system for design. When the full consequences of development are identified, environmental goals can be better targeted.

The Life Cycle Framework

The term *product life cycle* has been applied to both business activities and material balance studies. In business use, a product life cycle begins with the first phases of design and proceeds through the end of production. Businesses track costs, estimate profits, and plan strategy based on this type of product life cycle.

In contrast, environmental inventory and impact analysis follows the physical system of a product. Such life cycle analyses track material and energy flows and transformations from raw materials acquisition to the ultimate fate of residuals.

Life cycle design combines the standard business use of a life cycle with the physical system. By taking a systems approach, life cycle design seeks to avoid the cross-media transfer of pollutants or the shifting of impacts from one life cycle stage to another.

Life Cycle Stages

The product life cycle can be organized into the following stages:
- raw material acquisition
- bulk material processing
- engineered materials production
- manufacturing/assembly
- use and service
- retirement
- disposal

A general flow diagram of the product life cycle is presented in Figure 1. The net effect of each product life cycle is the consumption of resources and the conversion of these resources into residuals which accumulate in the earth and biosphere.

Product System Components

Life cycle design addresses the entire product system, not just isolated components. This is the most logical way to reduce total environmental impacts. The product system can be decomposed into four primary components:
- product
- process
- distribution network
- management

The *product* component consists of all materials in the final product and includes all forms of these materials from acquisition to their ultimate fate. *Processing* transforms materials and energy into intermediary and final products. *Distribution* consists of packaging systems and transportation networks used to contain, protect, and transport items. *Management* responsibilities include administrative services, financial management, personnel, purchasing, marketing, customer services, and training and educational programs. The management component also develops information and conveys it to others.

The process, distribution and management/information components can be further classified into the following subcomponents: facility or plant, unit operations or process steps, equipment and tools, labor, secondary material inputs, and energy.

Goals of Life Cycle Design

The primary objective of life cycle design is to reduce total environmental impacts and health risks caused by product development and use. This objective can only be achieved in concert with other life cycle design goals. Life cycle design seeks to:
- conserve resources
- prevent pollution
- support environmental equity
- preserve diverse, sustainable ecosystems
- maintain long-term, viable economic systems

Resource conservation, pollution prevention, and the equitable distribution of resources and risks are essential to preserve the sustainable ecosystems that comprise the planet's life support system. For this reason, product systems must be developed that balance human resources, natural resources, and capital while preserving healthy ecosystems.

Chapter 3. The Development Process

Design actions translate life cycle goals into high-quality, low-impact products. As Figure 2 shows, product development is complex. Many elements in the diagram feed
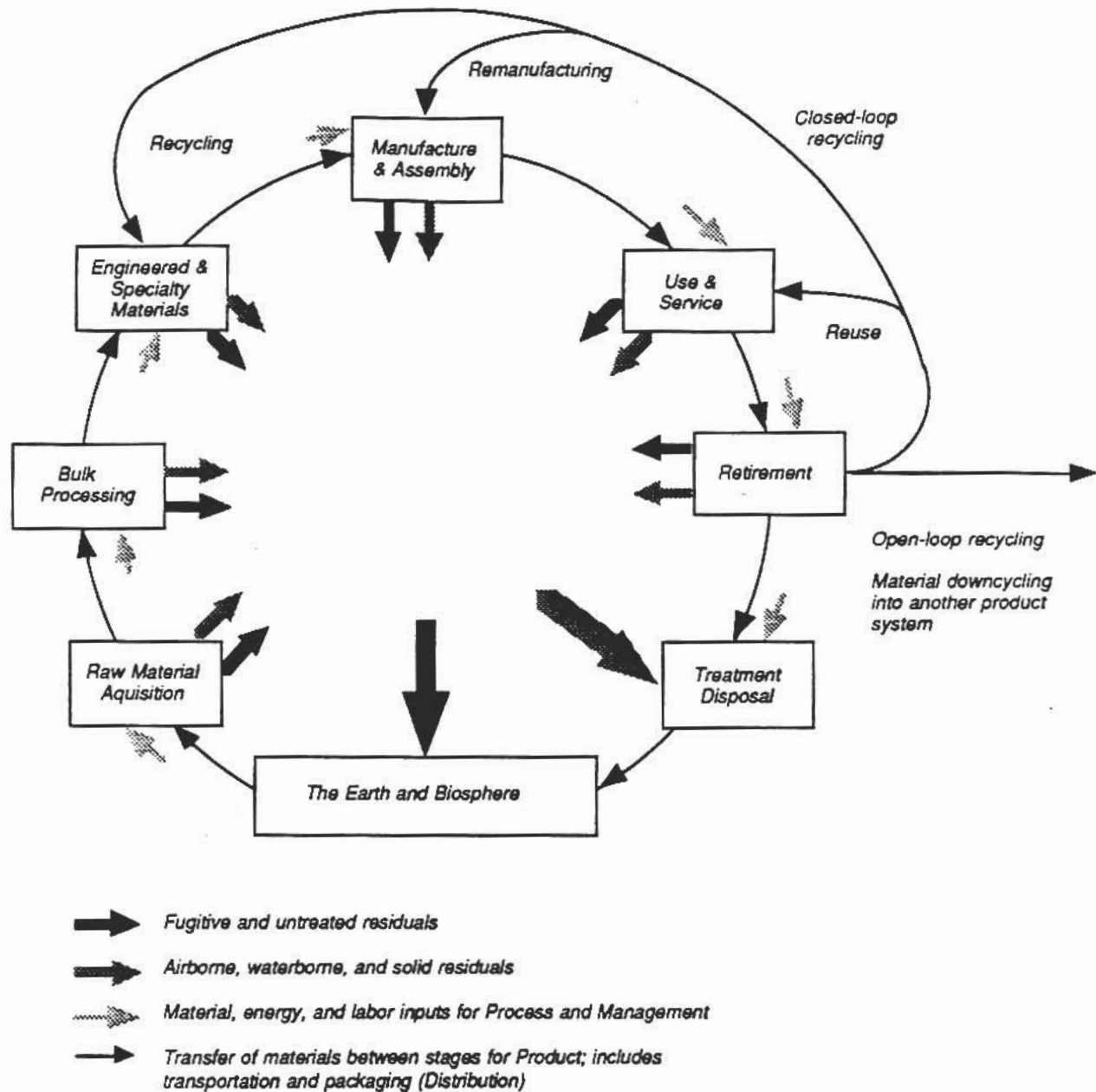
Figure 1. The product life cycle system.

back on others. This emphasizes the continual search for improved products.

Life cycle goals are located at the top to indicate their fundamental importance. Unless these goals are embraced by the entire development team, true life cycle design is impossible.

Management exerts a major influence on all phases of development. Both concurrent design and total quality management provide models for life cycle design. In addition, appropriate corporate policy, strategic planning, and measures of success are needed to support design projects.

Research and development discovers new approaches for reducing environmental impacts. The state of the environment provides a context for design. In life cycle design, current and future environmental needs are translated into appropriate designs.

A typical design project begins with a needs analysis, then proceeds through formulating requirements, conceptual design, preliminary design, detailed design, and implementation. During the needs analysis, the purpose and scope of the project are defined, and customers are clearly identified.

Needs are then expanded into a full set of design criteria that includes environmental requirements. Successful designs balance environmental, performance, cost, cultural, and legal requirements. Design alternatives are proposed to meet these requirements. The development team continuously evaluates alternatives throughout design. If studies show that requirements cannot be met or reasonably modified, the project should end.

Finally, designs are implemented after final approval and closure by the development team.
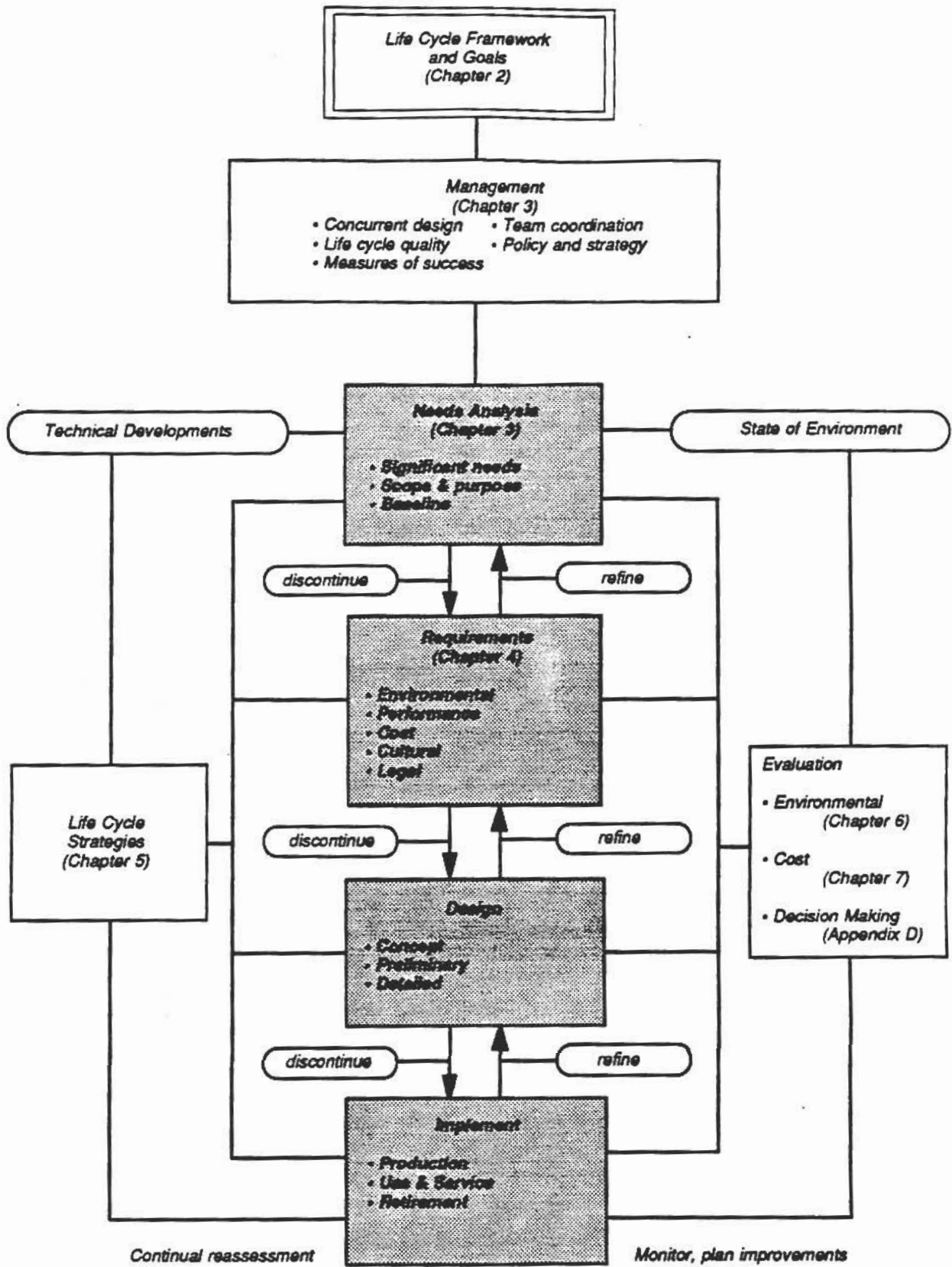
Figure 2. Life cycle design process.

Management

Commitment from all levels of management is a vital part of life cycle design. Corporate environmental policy must be translated into specific criteria to have a significant effect on product and process design activities. Objectives and guidelines need to be established in enough detail to provide useful guidance in design decision making.

The progress of life cycle design programs should be monitored and assessed using clearly established environmental and financial measures. Appropriate measures of success are necessary to motivate individuals within development teams to pursue environmental impact and health risk reductions.

Concurrent Design

Life cycle design is a logical extension of concurrent manufacturing, a procedure based on simultaneous design of product features and manufacturing processes. In contrast to projects that isolate design groups from each other, concurrent design brings participants together in a single team. By having all actors in the life cycle participate in a project from the outset, problems that develop between different disciplines can be reduced. Efficient teamwork also reduces development time, lowers costs, and can improve quality.

Life Cycle Quality

Environmental aspects are closely linked with quality in life cycle design. Companies who look beyond quick profits to focus on customers, multidisciplinary teamwork, and cooperation with suppliers provide a model for life cycle design. The life cycle framework expands these horizons to include societal and environmental needs. Life cycle design may thus build on total quality management, or be incorporated in a TQM program. In life cycle design, the environment is also seen as a customer. Pollution and other impacts are quality defects that must be reduced. Ultimate success depends on preserving environmental quality while satisfying traditional customers and employees.

Team Building

Life cycle design depends on cross-disciplinary teams. These teams may include any of the following life cycle participants: accounting, advertising, community, customers, distribution/packaging, environmental resources staff, government regulators/standards setting organizations, industrial designers, lawyers, management, marketing/sales, process designers and engineers, procurement/purchasing, production workers, research and development staff, and service personnel. Effectively coordinating these teams and balancing the diverse interests of all participants presents a significant challenge.

Needs Analysis

Design projects customarily begin by recognizing the need for change or uncovering an opportunity for new product development. The first step in any project should be identifying customers and their needs. Avoiding confusion between trivial or ephemeral desires and actual needs is a major challenge of life cycle design.

Once significant needs have been identified, the project's scope can be defined. This entails choosing system boundaries, characterizing analysis methods, and establishing a project time line and budget. In addition, development teams should decide whether the project will focus on improving an existing product, creating the next generation model, or developing a new product.

In choosing an appropriate system boundary for design, the development team must initially consider the full life cycle. More restricted system boundaries must be properly justified. Beginning with the most comprehensive system, design and analysis can focus on the:

- full life cycle,
- partial life cycle, or
- individual stages or activities.

Choice of the full life cycle system will provide the greatest opportunities for environmental impact reduction.

Narrowly bounded systems may provide useful results, but the limitations must be recognized and clearly stated. Stages may be omitted if they are static or not affected by a new design. In all cases, designers working on a more limited scale should be aware of potential upstream and downstream impacts.

Comparative analysis, also referred to as benchmarking, is necessary to demonstrate that a new design or modification is an improvement over competitive or alternative designs.

Requirements

Requirements define the expected design outcome. Design alternatives are evaluated on how well they meet requirements. Whenever possible, requirements should be stated explicitly to help the design team translate needs into effective designs.

Successful development teams place requirements before design. Rushing into design before objectives are defined often results in failed products.

Design Phases

The following phases of development are not significantly altered by life cycle design: conceptual design, preliminary design, detailed design, and implementation. During these phases, the development team synthesizes various requirements into a coherent design. Because life cycle design is based on concurrent practices, activities in several phases may be occurring at the same time.

Limitations

Lack of data and models for determining life cycle impacts makes analysis difficult. Lack of motivation can also be a problem. When the scope of design is broadened from that portion of the life cycle controlled by individual players to other participants, interest in life cycle design can dwindle. It can be difficult for one party to take actions that mainly benefit others.

Chapter 4. Requirements

Formulating requirements is one of the most critical activities in life cycle design. A well-conceived set of requirements translates project objectives into a defined solution space for design.

In life cycle design, environmental functions are critical to overall system quality. For this reason, environmental requirements should be developed at the same time as performance, cost, cultural, and legal criteria. All requirements must be balanced in successful designs. A product that fails in the marketplace benefits no one.

Key Elements

Requirements define products in terms of functions, attributes, and constraints. *Functions* describe what a successful design does. Functions should state *what* a design does, not *how* it is accomplished. *Attributes* are further details that provide useful description of functions. *Constraints are* conditions that the design must meet to satisfy project goals. Constraints provide limits on functions that restrict the design search to manageable areas.

Considerable research and analysis are needed to develop proper requirements. Too few requirements usually indicates that the design is ambiguous.

The level of detail expressed in requirements depends on the type of development project. Proposed requirements for new products are usually less detailed than those set for improving an existing product.

Use of Requirements Matrix

A multilayer requirements matrix provides a systematic tool for formulating a thorough set of environmental, performance, cost, cultural, and legal requirements. A schematic of this multilayer matrix is shown in Figure 3.

A practical matrix should be formed by further subdividing the rows and columns of this conceptual matrix. Matrices allow product development teams to carefully study the interdependencies and interactions between life cycle requirements. They also provide a convenient tool for identifying conflicts between requirements and clarifying trade-offs that must

be made. Issues that can assist designers in defining environmental requirements are introduced in the manual.

Ranking and Weighing Requirements

Ranking and weighing requirements provide designers with an understanding of the relative importance of various requirements. An example of a useful classification scheme follows.

- *Must* requirements are conditions that improvements and design alternatives have to meet. No design alternative is acceptable unless it satisfies all must requirements.
- *Want* requirements are desirable traits used to select best alternatives from proposed solutions that meet *must* requirements. Want requirements help designers seek the best solution, not just the first alternative that satisfies mandatory conditions. These criteria can play a critical role in customer acceptance and perceptions of quality.
- *Ancillary* functions are low ranked in terms of relative importance and can therefore be relegated to a wish list. Designers should be aware that these desires exist and try to incorporate them in designs when it can be done without compromising more critical parameters. Customers or clients should not expect to find many ancillary requirements included in the final design.

Chapter 5. Design Strategies

Effective strategies can only be adopted after project objectives are defined by requirements. Deciding on a course of action before the destination is known can be an invitation to disaster. Strategies flow from requirements, not the reverse.

A successful strategy satisfies the entire set of design requirements, thus promoting integration of environmental requirements into design. No strategy is exclusive. Most development projects should adopt a range of strategies to satisfy requirements. For this reason, no single strategy should be expected to satisfy all project requirements.

The following strategies are outlined in the manual:

Product system life extension
- appropriately durable
- adaptable
- reliable
- serviceable
- remanufacturable
- reusable

Material life Extension
- recycling

Material selection
- substitution

- reformulation

Reduced material intensiveness
Process management
- process substitution
- process control
- improved process layout
- inventory control and material handling
- facilities planning

Efficient distribution
- transportation
- packaging

Improved business management
- office management
- information provision
  labeling
  advertising

Chapter 6. Environmental Analysis Tools

A systematic means of gathering and analyzing data in varying depths is needed from the very beginning of a development project through implementation. In particular, environmental analysis is needed for benchmarking and the evaluation of design alternatives.

Environmental assessments are based on the following two components:
- Inventory analysis
- Impact analysis

An inventory analysis identifies and quantifies all inputs and outputs for a product system. Information about material and energy inputs and waste (residual) outputs for every significant step included in the system under study are compiled during the inventory analysis.

The purpose of impact assessment is to evaluate impacts and risks associated with the material and energy transfers and transformations quantified in the inventory analysis.

Scope of the Analysis

A full life cycle assessment may not be essential for many design activities. Scope can vary from complete quantification of all inputs, outputs, and their impacts to a simple verbal description of inventories and impacts. Boundaries for analysis may range from the full life cycle system to individual activities within a life cycle stage. The development team should be able to justify reducing the scope for design to a partial life cycle system.

The following factors related to analysis should also be considered when setting specific system boundaries: basis, temporal boundaries (time scale), and spatial boundaries (geographic). In general, the basis for analysis should be equivalent use. The time frame or conditions under which data were gathered should be clearly identified. A data collection period should be chosen that is representative of average system perfor-

mance. Spatial boundaries should also be noted because the same activity can have radically different effects in different locations.

Inventory Analysis

The inventory analysis should be conducted to satisfy requirements of the impact analysis. Two main tasks are involved in an inventory analysis:
- Identifying material and energy input and output streams and their constituents
- Quantifying these inputs and outputs

Allocation problems can occur in processes with multiple useful outputs. Proportioning impacts according to the total weight of the main product relative to the coproducts is a commonly used allocation method.

The EPA publication, *Life Cycle Assessment: Inventory Guidelines and Principles* (EPA/600/R-92/036) provides more detailed instructions for conducting an inventory assessment.

Impact Assessment

The final result of an impact analysis is an environmental profile of the product system. The translation of inventory data into environmental effects or impacts is achieved through a wide range of impact assessment models, including hazard and risk assessments models.

Impact analysis represents one of the most challenging analysis functions of product systems development. Although current methods for evaluating environmental impacts are incomplete, impact assessment is important because it enables designers and planners to understand the environmental consequences of a design more fully. The development team must recognize that analysis tools for assessing environmental impacts and risks are constantly improving. Designers, however, cannot wait for the "ultimate" environmental assessment models. Decisions should be based on the best available data and methods of assessment.

Environmental impacts can be organized into the following categories:
- resource depletion
- ecological degradation
- human health effects (health and safety risks)
- other human welfare effects

Resource acquisition has two basic environmental consequences:
- ecological degradation from habitat disruption (e.g., physical disruption from the mining)
- a reduction in the global resource base that effects sustainability

Ecological risk assessment includes many of the elements of human health risk assessment but is much more complex. The ecological stress agents must be identified as

Figure 3. Conceptual requirements matrix.

well as the ecosystem potentially impacted. Ecological stress agents can be categorized as chemical (e.g., toxic chemicals released to the environment), physical (e.g., habitat destruction through logging), and biological (introduction of an exotic species) agents.

Human health risk assessment includes hazard identification, risk assessment, exposure assessment, and risk characterization. Human health and safety risks can also be assessed using models that evaluate process system reliability.

Chapter 7. Life Cycle Accounting

Traditional accounting practices need to be modified to more fully reflect the total costs of pollution and resource depletion. Improved accounting practices can be a key element in facilitating life cycle design. Accounting methods outlined in this chapter are based on the total cost assessment model.

At present, most cost systems used in business are based on financial accounting. Because these systems are designed to serve reporting rather than *management functions*, environmental costs are usually gathered on the facility level. These costs are added to overhead and then assigned to specific products for management purposes. Allocation methods vary in accuracy, but future advances may allow gathering of much more accurate product-specific costs.

Life cycle design benefits from an accurate estimate of costs related to developing and using products. Material and energy flows provide a detailed template for assigning costs to individual products. Following the total cost

assessment model, life cycle accounting adds hidden, liability, and less tangible costs to those costs usually gathered. This expanded scope matches the range of activities included in life cycle design. Time scales are also expanded to include all future costs and benefits that might result from design.

Usual Costs

Life cycle accounting first identifies traditional capital and operating expenses and revenues for product systems. Many low-impact designs offer benefits when evaluated solely by usual costs. Such cost savings can be achieved through material and energy conservation, elimination or reduction of pollution control equipment, nonhazardous and hazardous waste disposal costs, and labor costs.

Hidden Costs

Hidden costs consist mainly of regulatory costs associated with product system development. Many hidden costs incurred by a company are gathered for entire facilities and assigned to overhead.

Hidden regulatory costs include the following (this is only a partial list):
Capital costs
- monitoring equipment
- preparedness and protective equipment
- additional technology

Expenses
- notification
- reporting
- monitoring/testing
- record keeping
- planning/studies/modeling

- training
- inspections
- manifesting
- labeling
- preparedness and protective equipment
- closure/post closure care
- medical surveillance
- insurance/special taxes

Liability Costs

Liability costs include fines due to noncompliance and future liabilities for remedial action, personal injury, and property damage. Avoiding liability through design is the wisest course. Because estimating potential environmental liability costs is difficult, these costs are often understated.

Less Tangible Costs

Many less tangible costs and benefits may be related to usual costs, hidden regulatory costs, and liabilities. Estimating intangibles such as corporate image or worker morale is difficult, as is projecting improvements in market share or benefits derived from improved customer loyalty.

Limitations

The main difficulties in life cycle accounting arise in estimating costs for many nontraditional items and properly allocating those costs to specific products/processes. Liability and less tangible costs are the most difficult to estimate.

Some low-impact designs have probably not been implemented because life cycle costs were not accurately calculated.

Externalities (costs borne by society rather than the responsible parties) also present problems. These costs are beyond the scope of accounting at present. As long as costs for pollution, resource depletion, and other externalities do not accrue to firms, accounting systems will not reflect these costs, and life cycle accounting will remain incomplete.

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APPENDIX D

•Sample thermodynamic calculations for R-12 using pressure-enthalpy diagram

•Sample calculations for R-134a using saturated property tables and ideal gas heat capacities

•Calculation of refrigerant mass flow rate

•Calculation of internal heat transfer coefficients (h), overall heat transfer coefficients (U), and heat exchanger areas

•Maximum allowable energy consumption under the 1993 DOE standard.

•Calculation of total anual energy consumption of refrigerator as a function of the heat load. Used in **Figure 5** and **Figure 6**

Sample thermodynamic calculations for R-12 using pressure-enthalpy diagram

Based on tables and Pressure-Enthalpy Diagram for R-12[1]

- T_1 = -4 °F based on problem statement is the evaporator temperature.

  using tables, for T_1:

  ==> P_1 = 21.9 psia

  ==> h_1 = 77.2 Btu/lb

  ==> rho_1 (density) = 1/1.7507 = 0.5712 lb/ft³

- T_4 = 115 °F based on problem statement is the condenser temperature.

  using tables, for T_4:

  ==> P_4 = 161.4 psia

  ==> h_4 = 35.2 Btu/lb

- 14 °F superheat of evaporator outlet based on problem statement:

  $T_2 = T_1 + 14 = -4 + 14$ ==> T_2 = 10 °F

  no pressure drops ==> $P_2 = P_1$ = 21.9 psia

  using diagram:

  ==> h_2 = 80 Btu/lb

  ==> s_2 = 0.175 Btu/lb-°R

  ideal gas law ==> rho_2 (density) = rho_1 (T_1/T_2) = 0.5712 x (455.67/473.67)

  ==> rho_2 = 0.55 lb/ft³

- superheat from 1 to 2 is subcooling from 4 to 5:

  $h_5 = h_4 - (h_2 - h_1) = 35.2 - (80 - 77.2)$

  ==> h_5 = 32.4 Btu/lb

  from diagram at h_5

  ==> T_5 = 112 °F (3 °F subcooling)

- isentropic compression from 2 to 3, using diagram:

  ==> T_3 = 150 °F

  ==> h_3 = 95 Btu/lb

  ==> rho_3 (density) = 3.5 lb/ft³

  isentropic compression ==> $s_3 = s_2$ = 0.175 Btu/lb-°R

- work input by compressor $w = h_3 - h_2 = 95 - 80$ ==> w = 15 Btu/lb

- isenthalpic expansion ==> $h_6 = h_5$ = 32.4 Btu/lb

•RE = Refrigeration Effect = $h_1 - h_6$ = 77.2 - 32.4 ==> RE = 44.8 Btu/lb

• COP = Coefficient Of Performance = RE/w = 44.8/15 ==> COP = 3.0

• Compressor volumetric efficiency (N_v) is composed of a contribution (N_{cv}) due to the clearance volume (fc = 5% from the problem statement) and a contribution (N_{vo}) from other factors (leakage, throttling, heat loss).

N_{vo} = 0.9 from problem statement.

N_{cv} = 1 - fc ((rho_3/rho_2) - 1) {see Teaching Aids and [2,3] }

==> N_{cv} = 1 - 0.05 ((3.5/0.55) - 1) ==> N_{cv} = 0.73

==> N_v = N_{cv} x N_{vo} = 0.73 x 0.9 ==> N_v = 0.66

•Volumetric Refrigeration Effect (RE_v) is based on the volume of refrigerant entering the compressor

==> RE_v = RE x rho_2 = 44.8 x 0.55 ==> RE_v = 24.6 Btu/ft^3

Sample calculations for R-134a using saturated property tables and ideal gas heat capacities

•Calculations for tables are the same as above, except for the calculation of the interchanger superheating and the compressor. For this part, use is made of the expression for the ideal gas heat capacity (C_p) supplied in Appendix B.[4]

• $C_p(T) = c_1 + c_2 + c_2 T + c_3 T^2 + c_5/T$ where

c_1 = 0.0012557213

c_2 = 0.00043742894

c_3 = -0.1487126 x 10^{-6}

c_5 = 6.802105688

•   in the superheat region:

$\Delta h = \int C_p(T)\ dT$

$\Delta s = \int C_p(T)/T\ dT$

• $h_2 - h_1 = \Delta h\ (T_1 \longrightarrow T_2)$ ==> $h_2 = h_1 + \Delta h\ (T_1 \longrightarrow T_2)$

$s_2 - s_1 = \Delta s\ (T_1 \longrightarrow T_2)$ ==> $s_2 = s_1 + \Delta s\ (T_1 \longrightarrow T2)$

•To calculate T_3, isentropic compression ==> $s_3 = s_2$

s_3 - saturated vapor entropy at saturation temperature (T_4) = $\Delta s\ (T_4 \longrightarrow T_3)$

we know both quantities on the left hand side of the equality and we know T_4 and $\Delta s(T)$, so we can can solve for T_3.

•h_3 = saturated vapor enthalpy at T_4 + $\Delta h\ (T_4 \longrightarrow T_3)$

•The following is a Maple program[5] which does the calculations for R-134a:

```
c1:=0.0012557213;
c2:=0.00043742894;
c3:=-0.1487126*10^(-6);
c5:=6.802105688;

T:=459.67+t;

Cp(t):=c1+c2+c2*T+c3*T^2+c5/T;
dh(t):=int(Cp(t),t);
ds(t):=int(Cp(t)/T,t);

evalf(subs(t=150,Cp(t)));

t1:=-4.0;
s1:=0.2229;
h1:=101.17;
t2:=10.0;
t4:=115.0;
s4vap:=0.21555;
h4vap:=116.47;
h4:=49.63;
CpL:=0.341;
rhol:=1.0/2.3454;
rho4vap:=1.0/0.2681;

dels12:=evalf(subs(t=t2,ds(t)))-evalf(subs(t=t1,ds(t)));
delh12:=evalf(subs(t=t2,dh(t)))-evalf(subs(t=t1,dh(t)));

s2:=s1+dels12;
h2:=h1+delh12;

#s3=s2

dels3:=s2-s4vap;

t3:=fsolve(dels3=(evalf(subs(t=t3,ds(t)))-evalf(subs(t=t4,ds(t)))));

delh3:=evalf(subs(t=t3,dh(t)))-evalf(subs(t=t4,dh(t)));

h3:=h4vap+delh3;

h5:=h4-delh12/CpL;

w:=h3-h2;

RE:=h1-h5;

COP:=RE/w;

rho2:=0.962*rhol;

rho3:=rho4vap*((459.67+t4)/(459.67+t3));

nuv:=0.9*(1.0-0.05*((rho3/rho2)-1.0));
```

• The following is the output from the above program:

```
constants := false, gamma, infinity, true, Catalan, E, I, Pi, c1 .. c5

c1 := .0012557213

c2 := .00043742894
                      -6
c3 := -.1487126000*10

c5 := 6.802105688

T := 459.67 + t
```

$$Cp(t) := .2027661110 + .00043742894\, t - .1487126000 \cdot 10^{-6}\,(459.67 + t)^2 + 6.802105688\, \frac{1}{459.67 + t}$$

$$dh(t) := \frac{12359}{60952}\, t + \frac{368}{1682559}\, t^2 - \frac{112}{2259391605}\,\left(\frac{45967}{100} + t\right)^3 + \frac{66545}{9783}\,\ln\left(\frac{45967}{100} + t\right)$$

$$ds(t) := -\frac{56}{753130535}\, t^2 + \frac{11692020536516}{31679663995976625}\, t - \frac{6654500}{9783}\,\frac{1}{45967 + 100\, t} + \frac{4341039569}{2563883404200}\,\ln(45967 + 100\, t)$$

```
.2242613777

t1 := -4.0

s1 := .2229

h1 := 101.17

t2 := 10.0

t4 := 115.0

s4vap := .21555

h4vap := 116.47

h4 := 49.63

CpL := .341

rho1 := .4263665046

rho4vap := 3.729951511

dels12 := .00565694169

delh12 := 2.61722998

s2 := .2285569417

h2 := 103.7872300

#s3-s2
dels3 := .0130069417

t3 := 149.9987034

delh3 := 7.70152939

h3 := 124.1715294

h5 := 41.95483877

w := 20.3842994

RE := 59.21516123

COP := 2.904939732

rho2 := .4101645774

rho3 := 3.515829537

nuv := .5592711051
```

Calculation of refrigerant mass flow rate

- from definition of Pull Down Time, we need to first calculate the load the refrigerator has to handle.

- need properties of air:
  $C_{p,air}$ = 0.25 Btu/lb-°F
  rho_{air} = 0.081 lb/ft³

- fresh food section:
  refrigeration load needed = V of fresh food section x rho_{air} x $C_{p,air}$
  $$= 13 \times 0.081 \times 0.25 = 0.263 \text{ Btu/°F}$$
  $\Delta T = (90 - 38) = 52$
  Load = 0.263 x 52 = 13.7 Btu

- freezer:
  refrigeration load needed = Volume of freezer x rho_{air} x $C_{p,air}$
  $$= 5 \times 0.081 \times 0.25 = 0.10125 \text{ Btu/°F}$$
  $\Delta T = (90 - 5) = 85$
  Load = 0.10125 x 85 = 8.61 Btu

- thus total heat load = 13.7 + 8.61 = 22.31 Btu

- Pull Down Time is 2 minutes
  ==> pull down capacity = 22.31 Btu/2 minutes
  ==> load refrigeration cycle must handle = 22.31 Btu/2 minutes
  $$= 669.3 \text{ BTU/hr} = 670 \text{ Btu/hr}$$

- check for reasonable pull down time when operating at extreme design conditions:
  Load = 0.263 (110 - 37) + 0.10125 (110 - 0) = 30.34 Btu
  Pull Down Time = 30.34/670 = 0.045 hr = 2.7 minutes ==> O.K.

- Calculate refrigerant mass flow rate, e.g. R-12:
  RE = Refrigeration Effect of R-12 = 44.8 Btu/lb (from above calculations)
  ==> mdot = mass flow rate of R-12 = Load / RE
  ==> mdot = 670/44.8 = 14.96 lb/hr

Calculation of internal heat transfer coefficients (h), overall heat transfer coefficients (U), and heat exchanger areas

The following calculations are performed for the cycle utilizing R-12

- condenser - desuperheat part:
  cool from $T_3 \rightarrow T_4$ ==> 150 °F —> 115 °F
  T_{avg} = 115 + (150 - 115)/2 = 132.5 °F

  ==>properties of R-12 at T_{avg}:[1]
  viscosity = μ = 0.033 lb/ft-hr
  thermal conductivity = k = 0.0064 Btu/ft-hr-°R
  heat capacity = Cp = 0.15 Btu/lb-°R

  ==> using supplied expression for the internal heat transfer coefficient of the desuperheat part of the condenser (h_{dh}):
  h_{dh} = 10.3 (0.15/0.033)$^{0.4}$ x (0.0064)$^{0.5}$ x 14.96 = 13.63 Btu/hr-ft²-°F

- condenser - condensation part:
  condensation at constant T_4 = 115 °F

  ==>use average of saturated gas and saturated liquid properties of R-12 at T_4:[1]
  viscosity = μ = 0.237 lb/ft-hr
  thermal conductivity = k = 0.0207 Btu/ft-hr-°R
  heat capacity = Cp = 0.225 Btu/lb-°R

  ==> using supplied expression for the internal heat transfer coefficient of the condensation part of the condenser (h_c):
  h_c = 592.5 (0.225/0.237)$^{0.4}$ x (0.0207)$^{0.5}$ x 14.96 = 847.57 Btu/hr-ft²-°F

- evaporator:
  evaporation at constant T_1 = -4 °F

  ==>use average of saturated gas and saturated liquid properties of R-12 at T_1:[1]
  viscosity = μ = 0.397 lb/ft-hr
  thermal conductivity= k = 0.0267 Btu/ft-hr-°R
  heat capacity= C_p = 0.181 Btu/lb-°R

  ==> using supplied expression for the internal heat transfer coefficient of the evaporator (h_{ev}):
  h_{ev} = 515.2 (0.181/0.397)$^{0.4}$ x (0.0267)$^{0.5}$ x 14.96 = 640.3 Btu/hr-ft²-°F

- using the internal heat transfer coefficients h_{dh}, h_c, and h_{ev} calculated above, and the expressions for the overall heat transfer coefficients U_{dh}, U_c and U_{ev}, the respective areas A_{dh}, A_c, and A_{ev} can be calculated for each of the desuperheat part of the condenser, the condenser, and the evaporator. This is done using a Maple program:[5]

```
# data
mdot:=14.96;
hdh:=13.63;
hc:=847.57;
hev:=640.3;
T3:=150.0;
h3:=95.0;
# hsv is saturated vapor enthalpy at T4
hsv:=89.0;
h4:=35.2;
RE:=44.8;
Uich:=40.0;
h5:=32.4;
T5:=112;
T1:=-4;
T2:=10;
T4:=115;

# desuperheat part of condenser
Udh:=1.0/((0.86/hdh)+0.079);
delT:=T3-115.0;
Q:=mdot*(h3-hsv);
Adh:=Q/(delT*Udh);

#Condensation
Uc:=1.0/((0.86/hc)+0.079);
delT:=25;
Q:=mdot*(hsv-h4);
Ac:=Q/(delT*Uc);

#Total Condenser area
Atc:=Adh+Ac;

#Evaporator
Uev    :=1.0/((0.98/hev)+0.092);
delT:=9;
Q:=mdot*RE;
Aev:=Q/(delT*Uev);

# Interchanger
delTlm:=((T4-T1)-(T5-T2))/evalf(ln((T4-T1)/(T5-T2)));
Aich:=((h4-h5)*mdot)/(Uich*delTlm);
```

- the output from the above program is:

```
# data
mdot := 14.96

hdh := 13.63

hc := 847.57

hev := 640.3

T3 := 150.0

h3 := 95.0

# hsv is saturated vapor enthalpy at T4
hsv := 89.0

h4 := 35.2

RE := 44.8

Uich := 40.0

h5 := 32.4

T5 := 112

T1 := -4

T2 := 10

T4 := 115

# desuperheat part of condenser
Udh := 7.037490255

delT := 35.0

Q := 89.760

Adh := .3644156277

#Condensation
Uc := 12.49770894

delT := 25

Q := 804.848

Ac := 2.575985739

#Total Condenser area
Atc := 2.940401367

#Evaporator
Uev := 10.69169578

delT := 9

Q := 670.208

Aev := 6.964990127

# Interchanger
delTlm := 110.2817061

Aich := .009495681895
```

Maximum allowable energy consumption under the 1993 DOE standard.

- AV = Adjusted Volume of our 18 ft³ refrigerator
  AV = 13 + 1.63 x 5 = 21.15 ft³

- Allowable energy consumption in KWatt-hour/year is
  KWatt-hour/year = 329 + 11.8 x 21.15 = 578.6 ≈ 578

Calculation of total anual energy consumption of refrigerator as a function of the heat load. Used in **Figure 5** and **Figure 6**

- energy consumption as a function of load. Power consuming devices are:
  evaporator fan ==> 10 watt
  condenser fan ==> 14 watt
  anti sweat heater ==> 19 watt (assume 30% running time)
  compressor ==> depends on refrigeration load, COP, total efficiency

- compressor power: (using R-12 as an example)
  COP = 3.0
  N_v = 0.66
  total efficiency = N_{tot}
      N_{tot} = isentropic efficiency (0.7) x motor efficiency (0.8) x N_v
    = 0.7 x 0.8 x 0.66
      = 0.37

compressor power = P_{comp} = (670/(COP x N_{tot})) x (1 watt-hr/3.414 Btu)
  ==> P_{comp} = 177 watt

- compressor is ON only a fraction of the time. This depends on the total heat that needs to be removed from the refrigerator (TRL)
  ==> TRL = heat gain through insulation + anti-sweat heater contribution
  anti-sweat heater contribution = 0.3 x 19 watt-hr/hr x 3.414 Btu/watt-hr
                    = 19.5 Btu/hr
  ==> TRL = heat gain through insulation + 19.5

  ==> fraction of the time compressor (and fans also) is ON = FRT
  ==> FRT = TRL / 670

- Total annual energy consumption of a device is the fraction of the time it is ON multiplied by its power mutiplied by the number of hours in a year, which is (365 x 24) 8760 hours.

- Total Energy Consumption = {FRT x (P_{comp} + 10 + 14) + (0.3 x 19)} x 8760

  in watts/year; for R-12 this simplifies to (201 x FRT + 5.85) x 8760, which is a linear dependence on the heat gain through the insulation.

- in general terms, the total energy consumption in Kwatt/year is:

  ==> {FRT(P_{comp} + 24) + 5.85)} 8.76

  ==> {FRT([350.45/(COP x N_v)] + 24) + 5.85)} 8.76

  the above shows that energy consumption as a function of the heat gain has a parametric dependence on the COP and N_v of the refrigerant used.

Heat gains by the fresh food compartment and the freezer

- The following are the starting assumptions from problem statement:

  => total inside height of refrigerator = H = 53 inches

  => total inside width of refrigerator = W = 26.5 inches

  => total inside depth of refrigerator = D = 22.1 inches

  => outside heat transfer coefficient = h_o = 1.47 Btu/hr-ft²-°F

  => inside heat transfer coefficient = h_i = 1.0 Btu/hr-ft²-°F

  => gasket heat gain coefficient for freezer = hg_i = 0.0055 Btu/hr-in-°F

  => gasket heat gain coefficient for cabinet = hg = 0.0014 Btu/hr-in-°F

  => no heat exchange between freezer and fresh food section, 3 inch separation between the two.

- The following symbols are used below:

  X1 : insulation thickness of fresh food section

  X2 : insulation thickness of freezer

  AF_i, AF_o : inside and outside areas of freezer section

  A_i, A_o : inside and outside areas of fresh food section

  QF : heat gain by freezer through insulation

  Q : heat gain by fresh food section through insulation

  VIF : freezer insulation volume

  VI : fresh food section insulation volume

  QGF : heat gain by freezer through gasket

  QG : heat gain of fresh food section through gasket

- The calculation of heat gains is a simple energy balance:

  Q = A U ΔT

  where the product A U is calculated as resistances in series composed of the outside convective heat transfer, insulation, and inside convective heat transfer. Gasket gains are assumed to be based on length rather than area. Rvalues for insulation are converted to thermal conductivity (k) by 1/Rvalue.

- The complexity arises in trying to utilize insulation as effectively as possible in both the fresh food section and the freezer. The optimal criterion is that the heat gain per unit insulation volume in the freezer must be equal to that in the fresh food section. Stated mathematically:

given a value for X1, X2 must be such that QF/VIF=Q/VI

X2 must be calculated maintaining a freezer volume of 5 ft³

Calculations for **Figure 4** and **Figure 7**.

- The following is a Maple program[5] that was used to generate data for **Figure 4** and **Figure 7** based on the above conditions.

- The following is the analytical and part of the numerical output of the above program:

```
D:=-22.1;
W:=26.5;
ho:=1.47/144;
hi:=1.0/144;
hgf:=0.0055;
hg:=0.0014;
dTF:=85;
dT:=52;


# Freezer Calcs

WF:=W+2*(X1-X2);
DF:=D+2*(X1-X2);
HF:=(5*144*12)/(WF*DF);
H:=53-3-HF;

AFi:=WF*DF+2*(WF*HF)+2*(DF*HF);
AFo:=(WF+2*X2)*(DF+2*X2)+2*(WF+2*X2)*(HF+X2)+
     2*(DF+2*X2)*(HF+X2);

UAF:=1/((1/(ho*AFo))+(X2/(k*AFi))+(1/(hi*AFi)));

QF:=UAF*dTF;

# Fresh Food Section Calcs

Ai:=W*D+2*(W*H)+2*(D*H);
Ao:=(W+2*X1)*(D+2*X1)+2*(W+2*X1)*(H+X1)+
    2*(D+2*X1)*(H+X1);

UA:=1/((1/(ho*Ao))+(X1/(k*Ai))+(1/(hi*Ai)));

Q:=UA*dT;

Qtot:= QF+Q;

WFT:=WF+2*X2;
DFT:=DF+2*X2;
HFT:=HF+X2;
WT:=W+2*X1;
DT:=D+2*X1;
HT:=H+X1;

# Insulation Volume Calcs.

VIF:=(2*WFT*HFT+2*DFT*HFT+DFT*WFT)*X2;
VI:=(2*WT*HT+2*DT*HT+DT*WT)*X1;

VIT:=VIF+VI;

#heat lost per cubic inch of insulation
rat1:=QF/VIF;
rat2:=Q/VI;

QGF:=hgf*2*(HF+WF)*dTF;
QG:=hg*2*(H+W)*dT;

Qgtot:=QGF+QG;

QALL:=Qtot+Qgtot;


# R-4 Insulation .
k:=1.0/(144*4);

for X1 from 0.5 by 0.5 to 4.0 do
z:=solve(rat1=rat2,X2);
z2:=subs(X2=z,QF);
z3:=subs(X2=z,Q);
z4:=subs(X2=z,QALL);
z5:=subs(X2=z,VIF);
z6:=subs(X2=z,VI);
z7:=subs(X2=z,VIT);
appendto(outdata4);
print(k,X1,z,z2,z3,z4,z5,z6,z7);
writeto(terminal);
od;
```

```
# R-6 Insulation .
k:=1.0/(144*6);

for X1 from 0.5 by 0.5 to 4.0 do
z:=solve(rat1=rat2,X2);
z2:=subs(X2=z,QF);
z3:=subs(X2=z,Q);
z4:=subs(X2=z,QALL);
z5:=subs(X2=z,VIF);
z6:=subs(X2=z,VI);
z7:=subs(X2=z,VIT);

appendto(outdata6);
print(k,X1,z,z2,z3,z4,z5,z6,z7);
writeto(terminal);
od;

# R-7 Insulation .
k:=1.0/(144*7);

for X1 from 0.5 by 0.5 to 4.0 do
z:=solve(rat1=rat2,X2);
z2:=subs(X2=z,QF);
z3:=subs(X2=z,Q);
z4:=subs(X2=z,QALL);
z5:=subs(X2=z,VIF);
z6:=subs(X2=z,VI);
z7:=subs(X2=z,VIT);
appendto(outdata7);
print(k,X1,z,z2,z3,z4,z5,z6,z7);
writeto(terminal);
od;

# R-8 Insulation .
k:=1.0/(144*8);

for X1 from 0.5 by 0.5 to 4.0 do
z:=solve(rat1=rat2,X2);
z2:=subs(X2=z,QF);
z3:=subs(X2=z,Q);
z4:=subs(X2=z,QALL);
z5:=subs(X2=z,VIF);
z6:=subs(X2=z,VI);
z7:=subs(X2=z,VIT);
appendto(outdata8);
print(k,X1,z,z2,z3,z4,z5,z6,z7);
writeto(terminal);
od;

# R-15 Insulation .
k:=1.0/(144*15);

for X1 from 0.5 by 0.5 to 4.0 do
z:=solve(rat1=rat2,X2);
z2:=subs(X2=z,QF);
z3:=subs(X2=z,Q);
z4:=subs(X2=z,QALL);
z5:=subs(X2=z,VIF);
z6:=subs(X2=z,VI);
z7:=subs(X2=z,VIT);
appendto(outdata15);
print(k,X1,z,z2,z3,z4,z5,z6,z7);
writeto(terminal);
od;

# R-20 Insulation .
k:=1.0/(144*20);

for X1 from 0.5 by 0.5 to 4.0 do
z:=solve(rat1=rat2,X2);
z2:=subs(X2=z,QF);
z3:=subs(X2=z,Q);
z4:=subs(X2=z,QALL);
z5:=subs(X2=z,VIF);
z6:=subs(X2=z,VI);
z7:=subs(X2=z,VIT);
appendto(outdata20);
print(k,X1,z,z2,z3,z4,z5,z6,z7);
writeto(terminal);
od;

# R-25 Insulation .
k:=1.0/(144*25);

for X1 from 0.5 by 0.5 to 4.0 do
z:=solve(rat1=rat2,X2);
z2:=subs(X2=z,QF);
z3:=subs(X2=z,Q);
z4:=subs(X2=z,QALL);
z5:=subs(X2=z,VIF);
z6:=subs(X2=z,VI);
z7:=subs(X2=z,VIT);
appendto(outdata25);
print(k,X1,z,z2,z3,z4,z5,z6,z7);
writeto(terminal);
od;
```

• The following is the analytical and part of the numerical output of the above program:

```
D := 22.1

W := 26.5

ho := .01020833333

hi := .006944444444

hgf := .0055

hg := .0014

dTF := 85

dT := 52

# Freezer Calcs
WF := 26.5 + 2 X1 - 2 X2

DF := 22.1 + 2 X1 - 2 X2

                            1
HF := 8640 ----------------------------------------
              (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

                            1
H := 50 - 8640 ----------------------------------------
                 (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

                                                              1
AFi := (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2) + 17280 ------------------
                                                          22.1 + 2 X1 - 2 X2

                     1
    + 17280 ------------------
             26.5 + 2 X1 - 2 X2

AFo := (26.5 + 2 X1) (22.1 + 2 X1)

                                1
    + 2 (26.5 + 2 X1) (8640 ---------------------------------------- + X2)
                              (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)


                                1
    + 2 (22.1 + 2 X1) (8640 ---------------------------------------- + X2)
                              (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

UAF :=

  1 / (97.95918371

      1 / ((26.5 + 2 X1) (22.1 + 2 X1)

                                      1
          + 2 (26.5 + 2 X1) (8640 ---------------------------------------- + X2)
                                    (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

                                      1
          + 2 (22.1 + 2 X1) (8640 ---------------------------------------- + X2))
                                    (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

        + X2

         / k

         / ((26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2) + 17280 ------------------
                                                               22.1 + 2 X1 - 2 X2

                            1
            + 17280 -------------------)
                     26.5 + 2 X1 - 2 X2

      + 144.0000000

        1 / ((26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

                            1                         1
            + 17280 ------------------ + 17280 ------------------))
                     22.1 + 2 X1 - 2 X2          26.5 + 2 X1 - 2 X2
```

QF :=

B5

1 / (97.95918371

1 / ((26.5 + 2 X1) (22.1 + 2 X1)

+ 2

(26.5 + 2 X1)

$$(8640 \; \frac{1}{(26.5 + 2 X1 - 2 X2)(22.1 + 2 X1 - 2 X2)} + X2)$$

+ 2

(22.1 + 2 X1)

$$(8640 \; \frac{1}{(26.5 + 2 X1 - 2 X2)(22.1 + 2 X1 - 2 X2)} + X2))$$

+ X2

/ k

/ ((26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

$$+ 17280 \; \frac{1}{22.1 + 2 X1 - 2 X2} + 17280 \; \frac{1}{26.5 + 2 X1 - 2 X2})$$

+ 144.0000000

1 / ((26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

$$+ 17280 \; \frac{1}{22.1 + 2 X1 - 2 X2} + 17280 \; \frac{1}{26.5 + 2 X1 - 2 X2}))$$

Fresh Food Section Calcs

$$Ai := 5445.65 - 839808.0 \; \frac{1}{(26.5 + 2 X1 - 2 X2)(22.1 + 2 X1 - 2 X2)}$$

Ao := (26.5 + 2 X1) (22.1 + 2 X1)

$$+ 2 (26.5 + 2 X1) (50 - 8640 \; \frac{1}{(26.5 + 2 X1 - 2 X2)(22.1 + 2 X1 - 2 X2)} + X1)$$

$$+ 2 (22.1 + 2 X1) (50 - 8640 \; \frac{1}{(26.5 + 2 X1 - 2 X2)(22.1 + 2 X1 - 2 X2)} + X1)$$

UA :=

1 / (97.95918371

1 / ((26.5 + 2 X1) (22.1 + 2 X1)

+ 2

(26.5 + 2 X1)

$$(50 - 8640 \; \frac{1}{(26.5 + 2 X1 - 2 X2)(22.1 + 2 X1 - 2 X2)} + X1)$$

+ 2

(22.1 + 2 X1)

$$(50 - 8640 \; \frac{1}{(26.5 + 2 X1 - 2 X2)(22.1 + 2 X1 - 2 X2)} + X1))$$

$$+ \; \frac{X1}{k \; (5445.65 - 839808.0 \; \frac{1}{(26.5 + 2 X1 - 2 X2)(22.1 + 2 X1 - 2 X2)})}$$

+ 144.0000000

```
                                            1
         -----------------------------------------------------------------)
                                                   1
         5445.65 - 839808.0 -----------------------------------------------
                              (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

Q :=
   52
      1 / (97.95918371

         1 / ((26.5 + 2 X1) (22.1 + 2 X1)

            + 2

               (26.5 + 2 X1)

                                          1
                  (50 - 8640 ----------------------------------------- + X1)
                              (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

            + 2

               (22.1 + 2 X1)

                                          1
                  (50 - 8640 ----------------------------------------- + X1))
                              (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

                                          X1
         + ---------------------------------------------------------------
                                                1
           k (5445.65 - 839808.0 -----------------------------------------)
                                  (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

         + 144.0000000

                                            1
         -----------------------------------------------------------------)
                                                   1
         5445.65 - 839808.0 -----------------------------------------------
                              (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

Qtot :=
   85
      1 / (97.95918371

         1 / ((26.5 + 2 X1) (22.1 + 2 X1)

            + 2

               (26.5 + 2 X1)

                                     1
                  (8640 ----------------------------------------- + X2)
                         (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

            + 2

               (22.1 + 2 X1)

                                     1
                  (8640 ----------------------------------------- + X2))
                         (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

         + X2

            / k

            / ((26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

                              1                        1
               + 17280 ------------------- + 17280 -------------------)
                        22.1 + 2 X1 - 2 X2           26.5 + 2 X1 - 2 X2

         + 144.0000000

            1 / ((26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

                              1                        1
               + 17280 ------------------- + 17280 -------------------))
                        22.1 + 2 X1 - 2 X2           26.5 + 2 X1 - 2 X2
```

```
        + 52

          1 / (97.95918371

            1 / ((26.5 + 2 X1) (22.1 + 2 X1)

              + 2

                (26.5 + 2 X1)

                                              1
                    (50 - 8640 ------------------------------------- + X1)
                               (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

              + 2

                (22.1 + 2 X1)

                                              1
                    (50 - 8640 ------------------------------------- + X1))
                               (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

                                          X1
            + ------------------------------------------------------------
                                              1
              k (5445.65 - 839808.0 -------------------------------------)
                                    (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

            + 144.0000000

                                          1
              ------------------------------------------------------------)
                                              1
              5445.65 - 839808.0 -------------------------------------
                                 (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

WFT := 26.5 + 2 X1

DFT := 22.1 + 2 X1

                           1
HFT := 8640 ------------------------------------- + X2
            (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

WT := 26.5 + 2 X1

DT := 22.1 + 2 X1

                              1
HT := 50 - 8640 ------------------------------------- + X1
                (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

# Insulation Volume Calcs.
VIF :=

  ((26.5 + 2 X1) (22.1 + 2 X1)

    + 2 (26.5 + 2 X1) (8640 ------------------------------------- + X2)
                            (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

                                              1
    + 2 (22.1 + 2 X1) (8640 ------------------------------------- + X2))
                            (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

    X2

VI :=

  ((26.5 + 2 X1) (22.1 + 2 X1)

    + 2

                                              1
        (26.5 + 2 X1) (50 - 8640 ------------------------------------- + X1)
                                 (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

    + 2

                                              1
        (22.1 + 2 X1) (50 - 8640 ------------------------------------- + X1)
                                 (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

      )

    X1
```

VĮT :-

((26.5 + 2 X1) (22.1 + 2 X1)

$$+ 2 (26.5 + 2 X1) (8640 \frac{1}{(26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)} + X2)$$

$$+ 2 (22.1 + 2 X1) (8640 \frac{1}{(26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)} + X2))$$

X2

+ ((26.5 + 2 X1) (22.1 + 2 X1)

+ 2

(26.5 + 2 X1)

$$(50 - 8640 \frac{1}{(26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)} + X1)$$

+ 2

(22.1 + 2 X1)

$$(50 - 8640 \frac{1}{(26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)} + X1))$$

X1

#heat lost per cubic inch of insulation
rat1 :=

85

1 / (97.95918371

1 / ((26.5 + 2 X1) (22.1 + 2 X1)

+ 2

(26.5 + 2 X1)

$$(8640 \frac{1}{(26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)} + X2)$$

+ 2

(22.1 + 2 X1)

$$(8640 \frac{1}{(26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)} + X2))$$

+ X2

/ k

/ ((26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

$$+ 17280 \frac{1}{22.1 + 2 X1 - 2 X2} + 17280 \frac{1}{26.5 + 2 X1 - 2 X2})$$

+ 144.0000000

1 / ((26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

$$+ 17280 \frac{1}{22.1 + 2 X1 - 2 X2} + 17280 \frac{1}{26.5 + 2 X1 - 2 X2}))$$

/ ((26.5 + 2 X1) (22.1 + 2 X1)

+ 2

(26.5 + 2 X1)

$$(8640 \frac{1}{(26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)} + X2)$$

+ 2

(22.1 + 2 X1)

1

```
                                                    (8640 ------------------------------------------- + X2))
                                                          (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

          / X2

rat2 :=

     52

       1 / (97.95918371

          1 / ((26.5 + 2 X1) (22.1 + 2 X1)

             + 2

                (26.5 + 2 X1)

                                                                    1
                   (50 - 8640 ------------------------------------------- + X1)
                              (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

             + 2

                (22.1 + 2 X1)

                                                                    1
                   (50 - 8640 ------------------------------------------- + X1))
                              (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

                                                        X1
          + ------------------------------------------------------------------------------
                                                              1
            k (5445.65 - 839808.0 ------------------------------------------- )
                                  (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

          + 144.0000000

                                                           1
            ------------------------------------------------------------------------ )
                                                              1
            5445.65 - 839808.0 -------------------------------------------
                               (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

          / ((26.5 + 2 X1) (22.1 + 2 X1)

             + 2

                (26.5 + 2 X1)

                                                                    1
                   (50 - 8640 ------------------------------------------- + X1)
                              (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

             + 2

                (22.1 + 2 X1)

                                                                    1
                   (50 - 8640 ------------------------------------------- + X1))
                              (26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

          / X1

                                    1
```

$$QGF := 8078.4000 \frac{1}{(26.5 + 2\ X1 - 2\ X2)\ (22.1 + 2\ X1 - 2\ X2)} + 24.77750 + 1.8700\ X1$$

$$- 1.8700\ X2$$

$$QG := 11.13840 - 1257.9840 \frac{1}{(26.5 + 2\ X1 - 2\ X2)\ (22.1 + 2\ X1 - 2\ X2)}$$

$$Qgtot := 6820.4160 \frac{1}{(26.5 + 2\ X1 - 2\ X2)\ (22.1 + 2\ X1 - 2\ X2)} + 35.91590$$

$$+ 1.8700\ X1 - 1.8700\ X2$$

QALL :=

    85

        1 / (97.95918371

            1 / ((26.5 + 2 X1) (22.1 + 2 X1)

                + 2

                    (26.5 + 2 X1)

                        $$(8640 \frac{1}{(26.5 + 2\ X1 - 2\ X2)\ (22.1 + 2\ X1 - 2\ X2)} + X2)$$

                + 2

                    (22.1 + 2 X1)

                        $$(8640 \frac{1}{(26.5 + 2\ X1 - 2\ X2)\ (22.1 + 2\ X1 - 2\ X2)} + X2))$$

            + X2

            / k

            / ((26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

                $$+ 17280 \frac{1}{22.1 + 2\ X1 - 2\ X2} + 17280 \frac{1}{26.5 + 2\ X1 - 2\ X2})$$

        + 144.0000000

            1 / ((26.5 + 2 X1 - 2 X2) (22.1 + 2 X1 - 2 X2)

                $$+ 17280 \frac{1}{22.1 + 2\ X1 - 2\ X2} + 17280 \frac{1}{26.5 + 2\ X1 - 2\ X2}))$$

    + 52

        1 / (97.95918371

            1 / ((26.5 + 2 X1) (22.1 + 2 X1)

                + 2

                    (26.5 + 2 X1)

$$\left(50 - 8640\ \frac{1}{(26.5 + 2\ X1 - 2\ X2)\ (22.1 + 2\ X1 - 2\ X2)} + X1\right)$$

$$+\ 2$$

$$(22.1 + 2\ X1)$$

$$\left(50 - 8640\ \frac{1}{(26.5 + 2\ X1 - 2\ X2)\ (22.1 + 2\ X1 - 2\ X2)} + X1\right)\right)$$

$$+\ \frac{X1}{k\ \left(5445.65 - 839808.0\ \frac{1}{(26.5 + 2\ X1 - 2\ X2)\ (22.1 + 2\ X1 - 2\ X2)}\right)}$$

$$+\ 144.0000000$$

$$\frac{1}{5445.65 - 839808.0\ \frac{1}{(26.5 + 2\ X1 - 2\ X2)\ (22.1 + 2\ X1 - 2\ X2)}}\right)$$

$$+\ 6820.4160\ \frac{1}{(26.5 + 2\ X1 - 2\ X2)\ (22.1 + 2\ X1 - 2\ X2)} + 35.91590$$

$$+\ 1.8700\ X1 - 1.8700\ X2$$

```
# R-4 Insulation .
k := .001736111111

z := .6665792542

z2 := 279.0156935

z3 := 393.7967232

z4 := 720.3914998

z5 := 1491.701868

z6 := 2105.355798

z7 := 3597.057666

appendto(outdata4)

.001736111111, .5, .6665792542, 279.0156935, 393.7967232, 720.3914998,

   1491.701868, 2105.355798, 3597.057666

z := 1.277043531

z2 := 179.3426217

z3 := 254.0102622

z4 := 480.9511950

z5 := 3125.055392

z6 := 4426.143283

z7 := 7551.198675

appendto(outdata4)
```

```
.001736111111, 1.0, 1.277043531, 179.3426217, 254.0102622, 480.9511950,
    3125.055392, 4426.143283, 7551.198675
z := 1.869310500
z2 := 133.1915083
z3 := 186.8892218
z4 := 367.7000475
z5 := 4970.647430
z6 := 6974.622046
z7 := 11945.26948
appendto(outdata4)
.001736111111, 1.5, 1.869310500, 133.1915083, 186.8892218, 367.7000475,
    4970.647430, 6974.622046, 11945.26948
```

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