Energy and Global Warming Impacts of Not-In-Kind and Next Generation CFC and HCFC Alternatives

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ABSTRACT

Chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs) are used in appliances, refrigeration and air conditioning equipment, and building and appliance insulation. CFCs and HCFCs are greenhouse gases, and their use in these energy consuming applications contributes to global warming both through the direct effect if they are released into the atmosphere and also by an indirect effect resulting from the combustion of fossil fuels to provide the energy to operate equipment these gases. Both CFCs and HCFCs can contribute chlorine to the upper atmosphere where it is a factor in the destruction of the stratospheric ozone layer. CFCs and HCFCs are regulated under the terms of the Montreal Protocol to Protect the Stratospheric Ozone Layer, and the production and use of both classes of chemicals will be phased out between 1996 and 2030 in the developed world (some nations have acted unilaterally to phase out these chemicals sooner).

Alternative chemicals and technologies have been proposed as substitutes for CFCs, HCFCs, and the use of the vapor compression cycle for refrigeration and air conditioning and plastic foam for building and appliance insulation. Among these alternatives are:

- hydrofluorocarbons (HFCs) which have zero ozone depleting potential for use in conventional compression systems for refrigeration and air conditioning and as blowing agents for plastic foam insulation,
- hydrocarbons and CO_2 as refrigerants and foam blowing agents,
- ammonia compression systems,
- Stirling cycle refrigeration using hydrogen or helium as the operating fluid,
- absorption and adsorption heat pumping cycles using ammonia/water, lithium bromide/water, ammonia and activated carbon, ammoniated salts, etc.,
- magnetic and thermoelectric refrigeration,
- Malone cycle refrigeration, and
- evacuated panel insulation.

These technologies will have to compete with conventional vapor compression and insulation technologies at higher and higher levels of efficiency. Future developments such as the use of microcellular foam, linear compressors, and adapting heat exchangers to take advantage of temperature glides will make it increasingly difficult for emerging technologies to displace evolving conventional technologies.

This report examines the current (1994) status of each of these technologies and their potential as alternatives to the existing technologies based on CFCs and HCFCs. In addition, the total equivalent global warming impacts (TEWI) are evaluated for those alternative technologies which are close enough to commercialization that performance data are available.

EXECUTIVE SUMMARY

INTRODUCTION

This report documents the results from the second phase of a project sponsored by the U.S. Department of Energy (DOE) and the Alternative Fluorocarbon Environmental Acceptability Study[1] (AFEAS). The initial study compared the global warming impacts of alternatives to CFCs in refrigeration, air conditioning, insulation, and solvent cleaning applications that could likely be commercialized before the year 2000. The concept of total equivalent warming impacts, or TEWI, was developed to combine the effects of the direct emissions of refrigerants, plastic foam insulation blowing agents, and solvents in end use applications with the indirect effects of energy consumption from the combustion of fossil fuels and generation of electricity used for heating or cooling. An example would be the impact on global warming of a refrigerator/freezer resulting from the foam insulation blowing agent, the refrigerant, and the lifetime energy use of the appliance. TEWI is based on the use of the global warming potentials (GWPs) developed by the Intergovernmental Panel on Climate Change (IPCC) that use carbon dioxide as a reference gas (GWPs of CO_2 = 1 irrespective of time horizon). TEWI provides a measure of the environmental impact of atmospheric losses of greenhouse gases from operation, service, and end of life disposal of equipment. It can be a useful index in comparing alternative technologies, but it should not be used to the exclusion of important factors such as energy efficiency, worker and consumer safety, reliability, and consumer costs.

The first study examined end use applications of CFCs in North America, Europe, and Japan and accounted for cultural and technical differences in each region. The differences included such things as the room and internal compartment temperatures for refrigerator/freezers, the sizes of refrigerators and thicknesses of insulation used, annual driving distances for automobiles, fuel types used for generating electricity, climate differences for building heating and cooling loads. Surprisingly, the principal results and conclusions in the study were similar without regard for the geographic region.

AFEAS and DOE supported this second study to evaluate the energy and global warming impacts of next generation fluorocarbons as well as not-in-kind (NIK)/non-fluorocarbon technologies that may be developed or improved in the future to displace CFCs and HCFCs from vapor compression refrigeration and air conditioning and foam insulations. Next generation technologies are classified in three categories:

- zero ODP fluorocarbon (HFC) refrigerants and blowing agents;
- expanded use and improvement of existing commercial technologies (i.e. ammonia and hydrocarbon compression, desiccant drying, evaporative cooling, and absorption chillers);
- development to a commercial level of novel technologies (i.e. acoustic compression, adsorption, Stirling, supercritical fluids, etc.)

[1] AFEAS is an international consortium of fluorocarbon chemical manfacturers.

The major change in calculational methods between the first and the second study is the use of GWPs based on a 100 year time horizon in the second study. Scientifically, there are arguments for using 500 years to approximate an infinite time horizon. This was the basis of the 1991 AFEAS/DOE project. Scientists have also stated that while long term or full life impact of greenhouse gases is a concern, that there is also a real concern that the continued high release rates of greenhouse gases could affect the rate of climate change in the next several decades. For these reasons, policy makers typically use a 100 year integration time horizon. This second study of AFEAS and DOE presents the principle comparison data on a 100 year time horizon basis.

TECHNICAL EVALUATIONS

There are three separate tasks in this follow-on project:

- evaluate next generation and NIK or non-fluorocarbon alternatives that could be developed to replace CFCs and HCFCs,
- update the previously published results to incorporate significant developments since 1991, and
- determine the sensitivity of the TEWI to the assumptions on fuel source emissions (e.g., CO_2 emissions from burning fossil fuels to generate electricity) and determine the impacts of fuel switching (e.g., substituting a gas-fired absorption chiller for an electric drive centrifugal chiller).

Not-In-Kind and Next Generation Technologies

An exhaustive search was conducted to identify emerging technologies that could replace conventional CFC or HCFC vapor compression refrigeration and air conditioning systems in the long term. Experts in the U.S., Europe, and Japan were asked to contribute additions to a list of technologies prepared at ORNL. Technologies on the combined list were evaluated. These include:

Transcritical CO_2 Compressors: an automobile air conditioning system is under development that uses CO_2 as the refrigerant. The TEWI calculated based on our best estimate of system efficiency and the weight of the laboratory prototype is comparable to the TEWI for an air conditioner with an annual loss rate of 110 g of HFC-134a; TEWI using projected weights of mass produced CO_2 air conditioners are comparable to the HFC-134a air conditioner losing 55 g of refrigerant annually. TEWI based solely on the published efficiency measurements from a laboratory prototype powered by an electric motor on a test stand and projected system weights is about 60% that of compression systems using HFC-134a. These comparisons of TEWI are inconsistent. Further testing of systems using CO_2 and HFC-134a with comparable heat exchangers and high efficiency compressors is required to verify the potential for this technology to reduce TEWI for automobile air conditioning.

Transcritical compression represents a significant departure from conventional air conditioners and further development is needed. Test results have demonstrated potential feasibility of this technology, but there has not been any field testing or independent evaluation of performance, safety, durability, manufacturing cost, or maintenance problems. These are all issues that need to be resolved for this technology to be viable in consumer products. This technology probably could not become a significant commercial factor before 2003 to 2010.

Adsorption Heat Pumps and Air Conditioners: several different concepts of adsorption refrigeration and heat pumps are under development in Europe, North America, and Japan. Each of these systems

uses sorption modules in place of the compressor of a conventional vapor compression refrigerator, air conditioner, or heat pump. A gas is adsorbed into or onto the solid sorbent at a low pressure and temperature; energy from a gas burner or waste heat is used to "desorb" the gas at a high temperature and pressure. The refrigeration circuit is similar to a conventional system using a condenser, evaporator, and expansion valve. System efficiencies can be enhanced by adding regenerators to recover heat from the sorption modules. Operating systems have been developed for space based applications requiring cryogenic temperatures.

In order to provide a continuous source of heating or cooling adsorption systems need to cycle between multiple sorption modules. The performance of similar processes used by the chemical industry for capturing and recycling gases degrade significantly as the number of cycles increases. Rigorous laboratory and field testing are needed to establish the long term efficiency, capacity, reliability and product lifetime of adsorption heating and cooling systems.

Systems are under development for residential heating and air conditioning and for automobile air conditioning. Comparisons are made between the projected performance of an adsorption heat pump, providing both heating and cooling, and a variable-speed electric heat pump for three different regional residential applications in the U.S. Based on standard emission and CO_2 emission factors for the U.S., for northern climates the very high heating efficiency of the adsorption systmem more than offsets the low cooling efficiency and results in a TEWI that is about 80% of that for a variable-speed heat pump. In climates with predominant cooling loads the adsorption unit has a TEWI that is 160% that of a variable-speed heat pump using fluorocarbon refrigerants. The TEWI are essentially the same in climates with balanced heating and cooling loads. Efficiency of peak power or base power generation and the relationship of electrical and natural gas supply and price will be important in real life decisions in choosing between electrical powered vapor compression and natural gas powered adsorption or vapor compression systems. If development proceeds as anticipated in 1993, adsorption heat pumps may be ready for field testing in 1998.

It is difficult to make accurate comparisons of TEWI for automobile air conditioners using sorption modules and waste heat from the engine because of uncertainties in the degree of waste heat utilization and the number of annual operating hours. Waste heat must be able to provide 30% of the cooling energy in order to achieve the same TEWI as an HFC-134a air conditioner losing an average of 100 g of refrigerant per year (50% to 60% to achieve the same TEWI if only 50 to 60 g are lost annually); any lower effectiveness results in a higher TEWI for the water/zeolite adsorption air conditioner. Additional problems concerning system cost, size, ease of maintenance, and operational lifetime need to be solved for adsorption automobile air conditioning to be viable in the marketplace.

Absorption Chillers: absorption chillers are commercially available and represent a major share of the chiller market for commercial air conditioning in Japan, although only a minor share in the U.S. market. Absorption chillers are often used in "hybrid" plants working in tandem with centrifugal chillers to reduce electricity peak demands. Absorption chillers can be effective tools for managing utility demand charges and meeting national energy goals, but have a lower efficiency than electric centrifugal chillers (absorption chillers can be powered by waste heat, in which case the lower efficiency is not important). Gas-fired single- and double-effect absorption chillers have higher TEWI than electric driven centrifugal chillers. Triple-effect absorption chillers are under development which may have TEWI that compare favorably with those of centrifugal chillers using HFC-134a or

HCFC-123 in countries tolerating high refrigerant make-up rates (e.g. leakage, maintenance, and end-of-life losses) or where power is generated by old coal- or oil-fired steam plants.

Absorption Heat Pumps: absorption heat pumps are under development for heating and cooling in residential and light commercial applications. At least one major manufacturer has prototype units undergoing laboratory and field testing and this technology could be commercialized in the 1997-98 time frame. Performance measurements are very similar to those used to calculate TEWI for adsorption heat pumps and the same conclusions can be drawn about TEWI reduction in predominantly heating climates and higher TEWI values for use in predominantly cooling climates with the same caveats mentioned above about electricity and natural gas.

Magnetic Heat Pumps: this is a very long range technology based on the magneto-caloric effect. Design studies have been performed but prototype equipment has not been built or system efficiencies demonstrated. About fifteen years of development effort are needed for breakthroughs in high temperature superconducting materials that are necessary to spur development of magnetic refrigeration equipment. As such it is inappropriate to calculate TEWI for this technology at this time. Magnetic heat pumps and refrigeration equipment will have high costs because of the need for superconducting magnets and magnetic shielding and this will limit the technology to niche markets such as gas liquifaction.

Evaporative Cooling: evaporative cooling is an established technology with approximately four million residential sized systems in use in the United States. Light commercial sized systems for office buildings, hospitals, and schools are also produced. Adiabatic cooling is performed by evaporating water on a wetted medium either directly in the supply air or indirectly with an intermediate heat exchanger to avoid increasing the humidity of the supply air. Energy use is very low in arid climates relative to unitary air conditioners and chillers. Recent and future developments are directed toward expanding the climatic conditions where evaporative cooling can be used effectively.

Manufacturers are working to expand their market share, at the expense of compression systems, and to expand into higher humidity regions. Several obstacles confront the use of evaporative cooling:

- ingrained architectural practices derived from experience,
- belief that a desiccant dryer would be required on secondary evaporative cooling systems to achieve adequate dehumidification during hot humid weather, and
- concerns about maintenance costs and water supply.

Data are not available to compare TEWI for residential vapor compression air conditioners and evaporative coolers. TEWI were only calculated for systems in two North American locations providing 1050 kW (300 tons) of equivalent cooling in a commercial building. The TEWI of an evaporative cooler is only 40% that of the chiller in southwest U.S. (Tucson, Arizona) but 155% in the midwest (Chicago, Illinois). The evaporative system could not meet the cooling load in the midwest; such cases demonstrate the difficulties in using evaporative cooling where humidity conditions vary greatly.

Thermoelectric Refrigeration: thermoelectric refrigeration is a solid-state technology that employs the Peltier effect to cool with no moving parts. It is a proven technology with small refrigeration systems commercially available. The efficiency is very low relative to compression equipment and significant breakthroughs in materials are needed to boost efficiency to broaden its application. Thermoelectric

cooling is promoted both for automobile air conditioning and for household refrigeration. The TEWI for an automobile air conditioner using thermoelectric cooling is 2.5 to 2.7 times the TEWI for a comparable HFC-134a vapor compression system; this is due to the low efficiency and the 50% increase in weight for the thermoelectric modules. TEWI was also calculated for a European size refrigerator (230 liter, 8 ft^3); the TEWI for the thermoelectric system is more than three times that of either a hydrocarbon or HFC vapor compression refrigerator.

Acoustic Compressors and Thermoacoustic Refrigeration: acoustic compressors and thermoacoustic refrigeration use sound waves to compress a refrigerant. Acoustic compressors, thermoacoustic engines, and thermoacoustic refrigeration systems have potential for application in domestic refrigeration, space conditioning, and automobile air conditioning. Thermoacoustic engines and thermoacoustic refrigeration have been demonstrated in hardware prototypes and have attracted interest among OEMs. Efficiency levels that have been reported are low relative to conventional positive displacement compression efficiencies, but high efficiencies are potentially possible with sufficient development. Effort is needed to eliminate the formation of shock waves, develop extremely durable valves for acoustic compressors, eliminate drag and turbulence along the tube walls, and to improve acoustic driver efficiencies. Industrial partners have expressed interest in participating in the development and application of acoustic technologies and the pace of development is likely to accelerate. One developer of acoustic compressors believes that they could be available in 1997 to 2000; projections for thermoacoustic refrigeration are less precise. A calculation of TEWI is not possible for either of these technologies with the data available.

Hydraulic Compressors: prototype air conditioners with hydraulic compressors have been constructed and tested using fluorocarbons and hydrocarbons as the working fluids. Overall, the hydraulic refrigeration system has been shown to work effectively but efficiencies are low relative to HCFC and HFC or other conventional vapor compression systems. The system's commercialization potential is limited by its size and weight (the single-stage prototype employed a 20 m (80 ft), 0.3 m (12 in.) ID vertical pipe). The system requires relatively simple components to use the force of a falling stream of water to compress a refrigerant and may be an attractive technology where there is a limited supply of technical skills for installing and servicing compression equipment. Further work is underway to improve the controls and performance through improved water/refrigerant separation techniques. The safety aspects of using relatively large volumes of hydrocarbons where there are limited technical skills would need to be addressed.

Desiccant Cooling: desiccant dehumidification systems are commercially available and further development efforts are being directed toward improving efficiencies and reducing costs. Work also needs to be done on the effects of cyclic operation on desiccant performance (15,000 to 20,000 cycles per year), the effects of dust buildup on solid desiccant wheels, the effects of continuous high temperature and of cycling temperatures on materials used for air seals. Solid and liquid open cycle desiccant systems are being commercially developed, and several packaged systems are expected to be on the market by 1995. Integrated desiccant cooling systems which combine a desiccant system with a vapor compression system have already been successfully installed in commercial buildings. For desiccant-based systems to penetrate the HVAC market further, their capital costs must be lowered. Improved performance through better performing desiccants, better attention to improving the thermodynamic performance of the system by reducing heat and mass losses could also contribute to greater market penetration. A calculation of TEWI was not possible with the available data.

Stirling Cycle Refrigeration: the Stirling cycle is being used in cryogenic applications and in other niche uses such as space-based applications where its performance is not affected by low gravity conditions. In other applications, it seems clear that there are limited advantages to Stirling. At the present time, interest and research on Stirling technologies for use in domestic refrigeration or space conditioning has waned from what it was only a few years ago because it is not projected to be any more efficient than the best compression systems.

Advanced Vapor Compression Refrigeration: conventional compression technologies continue to be improved and efficiencies of refrigeration and air-conditioning equipment will be higher in the future while refrigerant losses from emissive applications such as automobile air conditioning and supermarket refrigeration will be reduced. Foreseeable developments include the use of higher efficiency motors and compressors, heat exchanger modifications to take advantage of temperature glides of zeotropic refrigerant mixtures, and dual evaporators in two-temperature appliances like refrigerator/freezers. Regulations and costs are also working to reduce emissions of refrigerants by prohibiting venting of refrigerant charges during servicing and improving maintenance practices and procedures for eliminating leaks. Future developments are possible in the design of fittings used in large systems and in reducing refrigerant charge levels that will impact the amounts of refrigerant escaping to the atmosphere (this includes consideration of hermetic, stand-alone supermarket display cases, multiple condensing units in close proximity to the display cases, and secondary refrigerant circuits for supermarket refrigeration and hermetic automotive air-conditioning systems).

While it is acknowledged that compression systems will continue to improve, it is not known how quickly or how great these improvements will be. Almost all of the calculations for fluorocarbon compression systems and foam insulations in this report are based on either demonstrated production model efficiency and refrigerant loss rates or proven performance from drop-in tests (which would be worse than eventual production of optimized designs). Exceptions to this rule include the selection of refrigerant loss rates for automobile air conditioning and unitary air conditioners. Documented loss rates used in past studies are from when it was standard practice to simply vent the refrigerant charge during servicing. Such practices are now prohibited in the United States and elsewhere and it is clear that historical emission rates are not appropriate for current calculations. Current and projected future rates based on information from the industry are used in these cases.

TEWIs of compression systems based on these "hard" data are compared in some sections of the report to TEWIs of emerging not-in-kind technologies which have not been developed beyond laboratory prototypes. In these instances the best available laboratory or computer modeled performance data are used for the emerging technologies. While helpful in identifying future technologies that may have lower global warming impacts, the reader must be careful not to attribute too much significance to comparisons with minor reductions in TEWI because of the "softer" nature of the data used for the not-in-kind technologies.

Evacuated Panel Insulation: evacuated panels are under development which can have very low thermal conductivities. Thin, flat panels are constructed using a filler material (e.g. aerogel, diatomaceous earth, open-cell polyurethane foam, glass fibers) enclosed by one or more plastic or metal membranes under a vacuum. The costs, weight, and thermal properties are all factors of the selection of filler and membrane materials and the degree of vacuum used. "Total panel" thermal conductivities can be significantly higher than the "center-of-panel" measurements frequently cited by developers because heat transfer along the surfaces through the plastic or metal membranes becomes much more important as the heat transfer through the panel is reduced. Evacuated panels

were used briefly by a Japanese appliance manufacturer in the 1980s. Evacuated panel insulation could be an effective means to reduce TEWI, particularly for refrigerator/freezers. There continues to be doubt that panels will be able to retain a vacuum and maintain high thermal resistances over the lifetime of an appliance. If they are used, panels will probably be used in conjunction with blown foam insulations either to improve the thermal properties of appliance cabinets or to achieve comparable performance using thinner walls (permitting more usable internal volume).

Update of 1991 Analysis

Literature searches, workshops, and personal contacts were used to obtain up to date information on alternatives to CFCs and HCFCs in the applications evaluated in the original study. This information was then used to recalculate the TEWI for those applications where significant developments have occurred since 1991.

Some applications, such as household refrigerator/freezers and unitary equipment, have experienced significant improvements in energy efficiency since 1991 (30% for refrigerators in the U.S.); these improvements result in lower calculated energy use, lower indirect effects, and again increase the percentage of the total (but not the absolute value) from fluorocarbon emissions. Underlying assumptions on equipment lifetimes, CO_2 emissions from power generation, equipment dimensions, etc. are consistent with those used in the 1991 study. Some of the key findings are:

Refrigerator/Freezers: based on the latest available published information, there are no significant differences between the energy consumption for the European refrigerators using HFC-134a as both refrigerant and blowing agent and those using propane/isobutane as the refrigerant and cyclopentane blown foam. The most efficient combination of refrigerants and blowing agents examined is the use of HFC-134a as the refrigerant and HCFC-141b blown foam insulation as developed for North American refrigerators. It should be noted from existing data that the design of the refrigerator is a major factor in minimizing the TEWI of the system. At present, HCFC-141b blown foam has the highest insulating value and lowest thermal conductivity of the foam blowing alternatives investigated, and results in reduced energy consumption for equivalent fridge design and insulation thickness when compared to other foam blowing alternatives. Vacuum panel technology can further improve cabinet thermal performance.

The direct impacts of the new fluorocarbon refrigerants and blowing agents range from 15% to 18% of the total equivalent warming impact for refrigerator/freezers in North America. The refrigerators using HFC-134a as the refrigerant and HCFC-141b blown foam have 15% of the TEWI due to fluorocarbon emissions and those using HFC-134a as both refrigerant and blowing agent have 18% of the TEWI from chemical emissions. In each case, almost all of this direct effect is due to the foam blowing agent; assumptions on mandatory recovery of the refrigerant would result in marginal reductions in the TEWI. The same trends exist for the European refrigerator/freezers, although the percentage of direct effects is somewhat higher, primarily because the refrigerators are smaller and consequently have lower annual energy use; the absolute value is similar. The CO_2 emissions rate for the power generation mix in Europe (higher percentage of nuclear and hydroelectric power generation with virtually zero CO_2 emissions) is also a factor. In fact, the difference between running a fridge in the European country with the lowest emissions rate (CO_2/kWh) and in the country with the highest rate is much greater than the difference in direct effects between using fluorocarbon or hydrocarbons.

In 1993, hydrocarbon refrigerators were only available in manual defrost models and not in the much larger sizes or with the high efficiency evaporators common in North America and Japan (95% of the U.S. market is for automatic defrost models). Industry analyses conducted in Europe and the United States have determined that the risks associated with using flammable refrigerants in these large refrigerators with exposed refrigerant tubing in the freezer compartment and potential ignition sources from the defrost systems are unacceptable.

Unitary Air Conditioning Equipment: fluorocarbon mixtures and hydrocarbons have been proposed as substitutes for HCFC-22 in unitary equipment. Building codes in most developed countries limit the use of hydrocarbons in applications where a refrigerant leak could result in an explosive mixtures in air. These restrictions apply to air-to-air heat pumps and air conditioners used in North America and Japan (90 to 95% of the approximately 200 million unitary air conditioners and heat pumps in the world), although hydrocarbons may satisfy safety requirements for water-to-water units where the entire equipment package remains outdoors as in the hydronic heating only heat pumps in Europe. Leakage rates for these hydronic units are typically lower than for air-to-air systems whatever the refrigerant.

Comparisons were made for four non-flammable fluorocarbon mixtures that are proposed as substitutes for HCFC-22. The direct effects of the TEWI for all four vapor compression systems presented are significant, but managable, fractions of the total in each case. Regulations which prohibit venting of refrigerants will serve to reduce or virtually eliminate the direct effects from hermetic equipment. Energy efficiency is very important for this application and contributions to global warming from energy usage are either lower or about the same as current technology air conditioners using HCFC-22; engineering optimization is expected to reduce energy use and CO_2 emissions with the mixtures in future systems.

Supermarket Refrigeration Systems: these refrigeration systems have historically used large refrigerant charges and experienced high leakage rates. The current high costs of refrigerants and environmental regulations are resulting in better efforts at refrigerant containment and lower loss rates. The most likely substitutes for CFCs and HCFCs in supermarket refrigeration are mixtures of HFCs, although ammonia compression with indirect heat transfer loops is also being used in Europe. The HFC mixtures which have been developed and demonstrated reduce the TEWI for conventional supermarket refrigeration systems without increasing the component from energy consumption. The direct contribution is about the same for each of the mixtures and much lower than the direct contribution for R-502 systems they would replace. Significant reductions in TEWI can be achieved for existing equipment designs through better containment practices; government regulations and policies, such as the U.S. requirement to meet 15% leakage rates in systems containing greater than 22 kg (50 lbs). The industry is targeting 5% leakage rates for new systems. Refrigerant prices are likely to force end-users to reduce refrigerant emissions from their historically high rates with consequential reductions in TEWI.

Building codes in the developed countries make it expensive, or in some cases prohibitive, to use ammonia in most supermarkets because of the public safety risks in the densely populated areas near the stores. When ammonia is used, secondary loops are required so that the refrigerant lines do not enter the retail sales areas of the building; this imposes both first cost and efficiency penalties. Secondary loops are one means of reducing refrigerant charge and controlling leakage and emissions, whether ammonia or HFCs are used as the refrigerant, and reducing TEWI for this application, depending upon the balance between energy consumption and the refrigerant emissions.

Consciencious preventive maintenance programs for conventional systems with rigorous efforts to locate and fix leaks will also reduce TEWI.

Building Insulation: plastic foam insulations are used either as part or the major insulation in residences. The use of HCFCs in place of CFCs in polyurethane, polyisocyanurate, and extruded polystyrene foams can result in significant reductions in TEWI for this application. Hydrocarbon blown foams could result in lower TEWI than HCFC or HFC blown foams, but at increases in energy use as great as 5%. Further optimization of HFC and hydrocarbon foam formulations are likely to improve foam thermal conductivities and reduce energy use and TEWI more than that projected in this report. These conclusions are based on applying equal thicknesses of similar materials, which is a choice made only for the purposes of comparing the alternative blowing agents. In practice the building materials used will be selected based on cost and product availability, and the thickness will be determined by local building codes and practices or the architect's specifications.

Sensitivity Analysis

Sensitivity to Fuel Source Emissions: TEWI are calculated using "average" values for emissions from the combustion of fossil fuels to generate electricity based on regional generating capacities and fuel types and "generic" values for coal, gas, and oil burning power plants. This simplified approach does not account for differences in grades of coal or oil nor for differences between new high efficiency generators or old power plants nearing retirement. In no instance, however, were different trends observed or conclusions drawn where calculations were performed using both the low regional emission rate for Europe and the high North American average. While these rates are important for setting and achieving national CO_2 goals, conclusions drawn from the TEWI for these technologies is relatively insensitive to this assumption.

CONCLUSION

Several broad conclusions can be drawn from the study:

- while use of non-fluorocarbon, not-in-kind technologies may expand in already established niche markets and other NIK technologies may find new market applications, it appears unlikely that conventional and fluorocarbon technologies can be displaced to any large degree in the foreseeable future. In many cases, there is a greater potential for complementary NIK technology use in "hybrid" or combined NIK/fluorocarbon systems. Examples of hybrid systems would be desiccant dehumidification with fluorocarbon compression for supermarket air conditioning, evacuated panels in combination with polyurethane foam for appliance insulation, and absorption and centrifugal chiller plants for space cooling in large buildings.

- both this and the 1991 study showed that energy efficiency is a powerful tool to mitigate future potential climate change since it is directly connected to energy related CO_2 emissions. In most cases HFCs are the most efficient and safest available technology.

- TEWI combines the global warming effects of direct emissions of greenhouse gases and the indirect effects of CO_2 emissions from energy use by using the IPCC GWPs of the gases. TEWI for an application computed using GWPs corresponding to a time horizon several times the lifetime of the equipment accounts for most of the effect of the greenhouse gases (i.e.

refrigerants, insulation blowing agents, and CO_2) on the atmosphere and provides a balance between energy efficiency and fluorocarbon emissions. Using GWPs for short time horizons understate the importance of energy efficiency and the effects of CO_2 on global warming.

- the direct effect of refrigerant and blowing agent emissions in non-hermetic refrigeration and insulation applications is exacerbated by using short-term (100 year) GWP values in calculating the TEWI (the 100 year GWP values are approximately three times the 500 year values used in the 1991 TEWI report). The choice of integrated time horizon is a political issue beyond the scope of this report. The aggressive industry response to reduce fluorocarbon losses, similar to that already developed for the use of ammonia, will result in significantly lower TEWI for applications such as supermarket refrigeration and automobile air conditioning.

- in most applications, not-in-kind technologies will need to compete with the demonstrated improving energy efficiencies of conventional refrigeration, air-conditioning, and insulation technologies; since the mid-1970's energy use for refrigerator/freezers in the U.S. has been reduced by over 50%, the efficiency of unitary heat pumps and air conditioners have improved an average of 2½% per year for the last 25 years, and the power factors of centrifugal chillers have improved from 0.21 kW_{input}/kW_{output} to 0.15 kW_{input}/kW_{output} (0.75 kW/ton to 0.52 kW/ton, a 30% improvement). Efficiencies of gas furnaces have also improved from 60% in 1972 to over 80% in 1993. Both electric and gas-driven technologies will continue to improve. Some calculations in this report are based on efficiencies achieved by early 1994 and in the future efficiencies are likely to be higher as equipment and foam formulations are optimized for replacement refrigerants and blowing agents.

More specific conclusions can be drawn for individual applications:

- the direct effect of refrigerant emissions for HFC based automobile air conditioners is a significant part of the total at present and an aggressive industry response to reduce charge size and leakage could significantly reduce the impact of fluorocarbon loss from this application. Opportunities exist for alternative technologies such as CO_2 compression and adsorption air conditioners to reduce TEWI for this application, but the long term performance, lifetimes, and TEWIs of those technologies must be proven through extensive testing and none are expected to be viable in consumer products before 2003 to 2010.

- evaporative cooling can reduce TEWI substantially in dry climates relative to vapor compression cooling. Evaporative cooling can be integrated with conventional vapor compression air conditioning or desiccant dehumidification with efficiency and TEWI improvements. There are considerable obstacles to the expansion of evaporative cooling into higher or variable humidity regions.

- under most conditions, high efficiency electric centrifugal chillers using either HCFC-123 or HFC-134a have lower TEWIs than commercially available gas-fired absorption chillers. The triple-effect absorption compares favorably with HFC-134a chillers if there are high refrigerant make-up rates and CO_2 emission rates of 0.65 kg/kWh or higher. Absorption chillers are often used in "hybrid" chiller plants working in tandem with centrifugal chillers to reduce peak electrical demand and reduce operating costs.

- gas-fired adsorption and absorption heat pumps can reduce TEWI in climates dominated by heating requirements but could result in significantly larger TEWI values than electric heat pumps in climates with higher cooling requirements; however actual values will be specific to a region's electrical power generating efficiency and CO_2 emissions for the season and time of day. Long term performance and reliability concerns have not been resolved.

- supermarket refrigeration systems have had high TEWI because of high refrigerant leakage rates; efforts to reduce refrigerant emissions are acting to lower TEWI. Minor differences exist between HFC mixtures considered as alternatives to HCFC-22 and R-502 systems currently in use. Ammonia with secondary heat transfer loops could be a viable alternative to HFCs in this application but there is an energy penalty associated with necessary secondary heat exchangers. In the U.S., Japan, France, and other markets system designs will have to comply with stringent regulatory and permitting requirements that ensure safe use in retail and commercial areas. Other countries have fewer regulations affecting the use of ammonia and the governments in these countries are evaluating use of ammonia as well as fluorocarbon and hydrocarbon refrigerants. Stringent refrigerant containment measures will be necessary for ammonia which could also be used with HFCs, resulting in essentially identical TEWI for these alternatives.

- refrigerator/freezers using hydrocarbons as refrigerants have an insignificant or marginal TEWI difference when compared to refrigerators using HFC-134a. Efficiency, worker and consumer safety, and product liability considerations can be of equal or greater importance than the lower direct effects when considering the use of hydrocarbons in refrigeration equipment. Hydrocarbons are being used in some European countries as refrigerants and foam blowing agents for refrigerator/freezers, but with an energy penalty when compared to the best present technology. Hydrocarbons are also being considered in some developing countries. Appliance energy efficiency requirements and both technical and non-technical issues related to safety and manufacturer liability will restrict or prohibit application of hydrocarbon in home refrigeration in other markets.

INTRODUCTION

The initial study by AFEAS and DOE compared the global warming impacts of alternatives to CFCs in refrigeration, air conditioning, insulation, and solvent cleaning applications that could be commercialized before the year 2000. That project developed the concept of total equivalent warming impacts, or TEWI, that combined the effects of the direct emissions of refrigerants, plastic foam insulation blowing agents, and solvents into the atmosphere with the indirect effects of energy consumption from the combustion of fossil fuels and generation of electricity. TEWI is based on the use of the global warming potentials (GWPs) for trace gases developed by the Intergovernmental Panel on Climate Change (IPCC) that use carbon dioxide as a reference gas (GWP of CO_2 = 1).

The first study examined end use applications of CFCs in North America, Europe, and Japan and accounted for cultural and technical differences in each region. The differences included such things as the room and compartment temperatures for refrigerator/freezers, the sizes of refrigerators and thicknesses of insulation used, annual driving distances for automobiles, fuel types used for generating electricity, and climate differences for building heating and cooling loads. Surprisingly, the principal results and conclusions in the study were the same without regard for the geographic region.

AFEAS and DOE supported this second study to evaluate the energy and global warming impacts of not-in-kind (NIK) and next generation alternative technologies that could be developed in the long term to replace vapor compression refrigeration and air conditioning and foam insulations using CFCs and HCFCs. The NIK alternatives include such things as thermoelectric refrigeration, evaporative cooling, evacuated panel insulation, and adsorption heat pumps. Next generation technologies include refrigerants and blowing agents with zero ozone depleting potentials (ODPs), acoustic compressors, and ammonia and hydrocarbon compression systems.

TECHNICAL EVALUATIONS

There are three separate tasks in this followon project:

- evaluate NIK alternatives that could be developed to replace CFCs and HCFCs,
- update the previously published results to incorporate significant developments since 1991, and
- determine the sensitivity of the TEWI to the assumptions on fuel source emissions (e.g., CO_2 emissions from burning fossil fuels to generate electricity) and determine the impacts of fuel switching (e.g., substituting a gas-fired absorption chiller for a centrifugal chiller).

Not-In-Kind and Next Generation Technologies

An exhaustive search was conducted to identify emerging technologies that could replace vapor compression refrigeration and air conditioning systems in the long term. Experts in the U.S., Europe, and Japan were asked to contribute additions to a list of technologies prepared at ORNL. These

technologies are described in Parts 1 and 2 of this report and some of them are considered in quantitative analyses in Parts 3 and 4.

Update of 1991 Analysis

Literature searches, workshops, and personal contacts were used to obtain up-to-date information on alternatives to CFCs and HCFCs in the applications studied in the first cooperative project on energy and global warming. This information was then used to recalculate the TEWI for those applications where significant developments have occurred since 1991. In this study, TEWI are computed with GWPs for fluorocarbons using the 100 year integrated time horizon; this is a major change from the original project since most of the results of that study are presented using 500 year GWPs. Many of the underlying assumptions on equipment lifetimes, CO_2 emissions from power generation, equipment dimensions, etc. are consistent with those used in the 1991 study. Computer models are available from some segments of the energy and manufacturing industries which can be used to perform more sophisticated evaluations which account for seasonal operating efficiencies and power plant emissions more rigorously than could be accommodated in this study (Sioshansi 1994). Discussions and results from the update of the 1991 report are included in Part 3 of this report.

Sensitivity to Fuel Source Emissions

TEWI are calculated using "average" values for emissions from the combustion of fossil fuels to generate electricity based on regional generating capacities and fuel types and "generic" values for coal, gas, and oil burning plants. This simplified approach does not account for differences in grades of coal or oil nor for differences between new high efficiency generators or old plants nearing retirement.

The analyses also have not addressed questions that arise from opportunities to substitute gas-fired technologies such as using double effect absorption chillers in place of electrically driven centrifugal chillers. These trade-offs could be part of individual, corporate, or national plans to reduce CO_2 emissions. The sensitivity of the results to assumptions about fuel source emissions and fuel switching are evaluated and discussed in Part 4.

GLOBAL WARMING POTENTIALS

The evaluation of total impacts of any system using fluorocarbons on global warming inevitably requires some assumptions about the impact of each individual chemical. This report uses the GWPs endorsed by the Intergovernmental Panel on Climate Change (IPCC) and the World Meterological Organization (WMO). These are listed in Table 1 (IPCC 1994). GWPs are dependent on the atmospheric lifetime of each compound and also the integration time horizon, ITH (GWPs are the impact of a mass of the gas relative to the impact of an equal mass of CO_2 over a given period of time, the integration time horizon). The analyses in this report are based on the 100 year ITH; this is a significant departure from the assumptions used in the 1991 report by Fischer, et al, which used the 500 year ITH.

As an example, releases of the refrigerant, blowing agent, and CO_2 from energy use associated with a household refrigerator are illustrated in Fig. 1. The effects of the CO_2 build up as electricity is used

during the lifetime of the appliance, and then drop off sharply and decay in time. The release of the blowing agent is indicated by a very sharp peak when the refrigerator is removed from service and assumed to be crushed upon disposal. The effects of the blowing agent diminish more rapidly than the CO_2 and essentially disappear before the 100 year mark on the horizontal axis. A very narrow, indistinguishable, region above the curve for the blowing agent represents the effects of the refrigerant. The total global warming effects of the refrigerator are represented by the area under the curves from time zero out some designated number of years. Both the blowing agent and the refrigerant are completely accounted for within 100 years; the CO_2 resulting from use of the refrigerator will continue to contribute to global warming even 500 years later. The lower graph in Fig. 1 represents the effects from the release of a single kilogram of CO_2. The total equivalent warming impact, or TEWI, for an application is the area under the curves in Fig. 1a out to a certain number of years, typically 100, divided by the area under the curve in Fig. 1b for the same number of years. The number of years chosen is referred to as the integration time horizon, or ITH. A very long ITH accounts for all of the global warming impacts, but could put too little emphasis on short term effects. Short ITHs also lack balance in that they understate the significant long term effects of CO_2 emissions.

Table 1. Global warming potentials (GWPs) of gases.

Compound	Atmospheric Lifetime (years)	Integration Time Horizon		
		20 years	100 years	500 years
CFC-11	50	5000	4000	1400
CFC-12	102	7900	8500	4200
CFC-115	1700	6200	9300	13000
HCFC-123	1.4	300	93	29
HCFC-124	5.9	1500	480	150
HCFC-141b	9.4	1800	630	200
HCFC-142b	19.5	4200	2000	630
HCFC-22	13.3	4300	1700	520
HFC-125	36	4800	3200	1100
HFC-134a	14	3300	1300	420
HFC-32	6	1800	580	180
HFC-143a	55	5200	4400	1600
HFC-152a	1.5	460	140	44
Perfluorohexane	3200	4500	6800	9900
CO_2		1	1	1
c-pentane*		31	11	6
n-pentane*		31	11	6

* indirect effects of non-methyl hydrocarbons (IPCC 1990)

The significance of the integration time horizon on conclusions drawn from the TEWI is evident in Fig. 2 where results for a 530 liter (18 ft^3) refrigerator/freezer are plotted. The contribution from energy consumption varies from over 80% to over 95% of the TEWI depending on whether a 20 year or 500 year ITH is used. The contribution from the foam blowing agent is large at 20 years, narrows quickly to about 5% at 100 years, and is around 3% at 500 years. Assuming that the refrigerant is not recovered, the fraction from the refrigerant is between 1 and 3% of the total (The fractions for the fluorocarbons are somewhat higher than reported in 1991 because of assumptions made about energy use to reflect system energy efficiency improvements made since 1991).

Scientifically, there are arguments for using infinite time horizons or approximately the same by using a 500 year time horizon. This was the basis of the 1991 AFEAS/DOE study. Scientists have also stated that while long term change is a concern, that there is also a real concern that the continued high release rates of greenhouse gases could affect the rate of climate change in the next several decades. For these reasons, policy makers typically use a 100 year integration time horizon. This second study of AFEAS and DOE presents the principle comparison data on a 100 year time horizon basis. A consistent conclusion of the two studies is that energy efficiency is a powerful tool to mitigate future potential climate change since it is directly connected to energy related CO_2 emissions. Furthermore, emissions of fluorocarbons need to be minimized wherever practical or their direct contribution to global warming can affect or overwhelm the efficiency benefit when looking at TEWI values for most applications using fluorocarbons as refrigerants or insulation blowing agents.

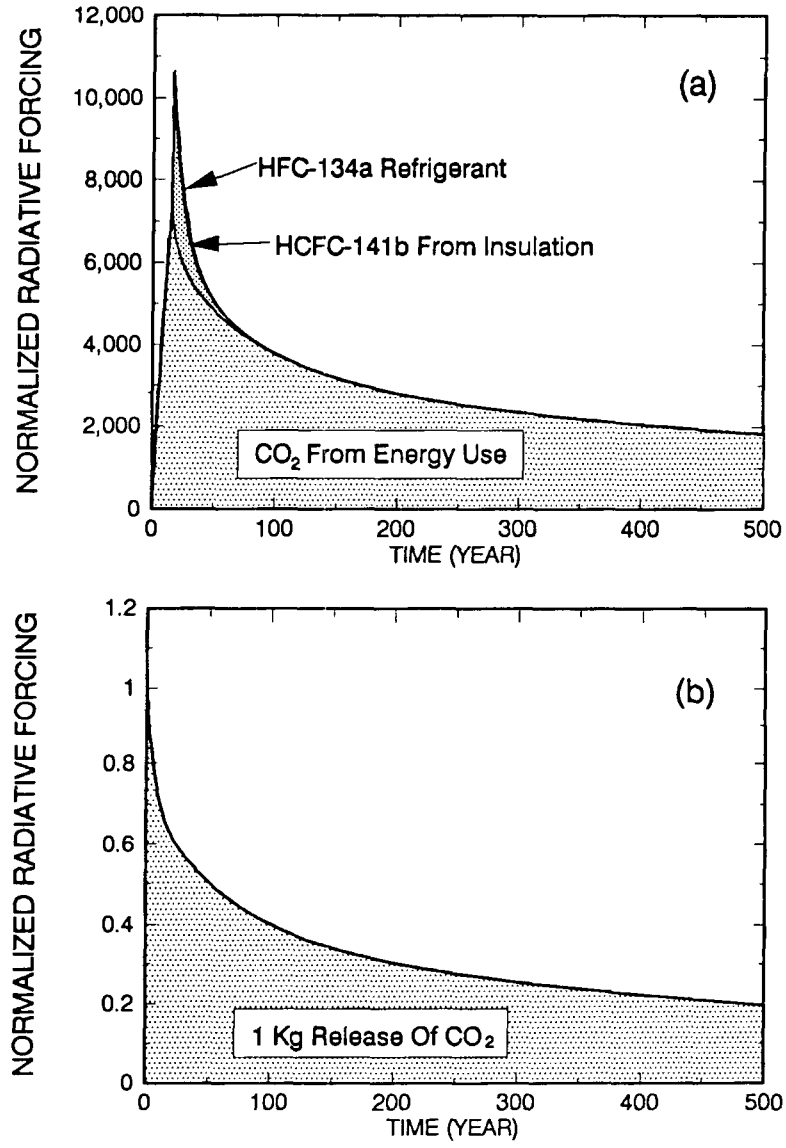

Fig. 1. Radiative forcing of greenhouse gases from a refrigerator/freezer.

CO_2 EMISSION RATES

The TEWI also depends on assumptions on carbon dioxide emission from energy use, either from the use of fossil fuels in automobile engines or natural gas in furnaces or indirectly from combustion of coal, natural gas, or fuel oil for generation of electricity used for refrigeration or air conditioning.

Electricity

The assumed emission rates for power generation are 0.513 kg CO_2/kWh delivered for Europe, 0.581 kg/kWh for Japan, and 0.650 kg/kWh for North America (Marland 1990). These values were derived for the 1991 study based on installed generating capacity for each of the three regions and emission rates of:

- 0.583 kg/kWh (1.28 lb/kWh) for power plants burning natural gas,
- 0.96 kg/kWh (2.11 lb/kWh) for plants burning fuel oil,
- 1.25 kg/kWh (2.50 lb/kWh) for coal fired plants, and
- 0 kg/kWh for hydroelectric and nuclear power plants (Fischer 1991).

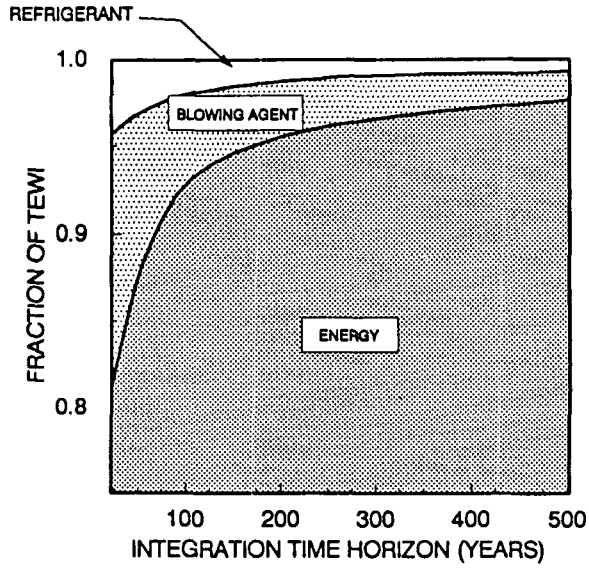

Fig. 2. Fraction of TEWI as a function of time horizon for a refrigerator/freezer.

Admittedly these are not "life-cycle" values that account for fuel use in the construction of power plants and hydroelectric dams and production and transportation of the fuel itself.

Fuel Use for Space Conditioning

Carbon dioxide emissions from fuel use for furnaces or gas-fired chillers and heat pumps are assumed to be 51.1 g CO_2/MJ (53.9 g/1000 Btu) of input heat value for natural gas and 69.5 g CO_2/MJ (73.3 g/1000 Btu) of input heat value for fuel oil. These assumptions are consistent with the data used in the 1991 study (Fischer 1991).

CO_2 Emissions from Gasoline

Carbon dioxide emissions from combustion of gasoline for automobile air conditioners is assumed to be 2.32 kg/liter (8.79 kg/gallon) of fuel (Fischer 1991).

ORGANIZATION OF THE REPORT

This report consists of an executive summary, an introduction, four technical sections, and a set of conclusions. The four technical sections in the body of the report cover descriptions of emerging

technologies, new refrigerant compression systems (i.e. hydraulic and acoustic compressors), refrigeration and insulation applications affected by the CFC and HCFC phase-outs, and an analysis of the sensitivity of the total equivalent warming impact to assumptions on efficiency, refrigerant loss rates, and CO_2 emission rates for power generation. Table 2 contains a cross-reference of the emerging technologies with relevant discussions in the sections on end-use applications and sensitivity to assumptions.

Table 2. Cross-reference of emerging technologies and end-use applications (page numbers).

Alternative Technology	Automobile A/C	Centrifugal Chillers	Refrigerator/ Freezers	Supermarket Refrigeration	Unitary Equipment	Building Insulation
HFC & HC Refrigerants	89, 96	101-102, 160-164	110, 115-117	122, 125, 127-130	132-134, 139-141	
Ammonia Compression		102, 160-164		124	135	
HFC & HC Blowing Agents			111, 115-117			145, 149
Evacuated Panel Insulation			113, 115-117			§
Adsorption Cycle	92, 96				137, 141	
Absorption Cycle		103, 160-164	111		136, 141	
Stirling Cycle	94, 97		§	125	§	
Brayton (Air) Cycle	95					
Thermoelectric Refrigeration	93, 97		111			
Transcritical Compression	90, 96					
Thermoacoustic Refrigeration			§	§		
Evaporative Cooling		104			138	
Desiccant Cooling				§	§	
Magnetic Heat Pumps		§		126	†	
Malone Refrigeration	†		†	†	†	
Thermoelastic Heat Pumps					†	

§ possible application, but not discussed
† very long term application, not discussed

PART 1: NOT-IN-KIND TECHNOLOGIES

ADSORPTION REFRIGERATION AND AIR CONDITIONING

Introduction

Adsorption refrigeration and air conditioning is similar to vapor compression cooling with the notable difference that the electric motor driven compressor is replaced with a sorption reactor (also referred to as "module," "canister," or "bed") that uses a heat source to create the change in pressure. A low pressure gas (such as an HFC, ammonia, or water) is physically adsorbed onto or chemically absorbed into a solid (such as activated carbon or a zeolite). This process occurs at or near room temperature. When the solid in the sorbent module has adsorbed as much of the gas as it can, a burner or waste heat is used to raise its temperature an additional 100° to 300°C (180° to 540°F). The elevated temperature causes the gas to desorb from the solid at a high pressure. The high pressure gas passes through a condenser, an orifice expansion plate, and an evaporator just as in a vapor compression system. The low temperature, low pressure two phase refrigerant coming out of the orifice is evaporated to provide useful cooling and to repeat the cycle.

The principal advantages of the adsorption cycle are that it is inherently simple, there are no wear-related moving parts (the only moving parts are a valve and a small pump), lack of vibration, and potentially long life. The historically low efficiency relative to compression cooling has been a disadvantage limiting application of adsorption in the past. The absence of vibration and wear related moving parts makes sorption refrigeration of interest for space based applications. Operating adsorption systems have been constructed for cryogenic temperatures as a part of the space program and Strategic Defense Initiative in the United States. These include a hydrogen chemisorption system operating at 20 to 30 K (-424° to -406°F) and oxygen chemisorption at 65 to 90 K (-343° to -298°F) for space based sensor cooling (Jones and Christophilos 1993).

Technology Description

Adsorption air conditioners have also been developed which are designed for consumer applications. A schematic diagram for a heat pump using HFC-134a as the sorbate or refrigerant and activated charcoal as the sorbent is shown in Fig. 3. Gas would be adsorbed into the charcoal at 38°C (100°F) and 334 kPa (48 psia) charging the sorption reactor on the right; a gas burner is used to discharge the fully charged sorption reactor on the left where refrigerant is desorbed at 204°C (400°F) and 960 kPa (140 psia) (Jones and Christophilos 1993). Liquid evaporating at 4.4°C (40°F) and 334 kPa (48 psia) provides the useful cooling. Refrigerant circuiting and heat exchangers are used to preheat the 4.4°C (40°F) vapor from the evaporator to the adsorption temperature of 38°C (100°F). When the right hand reactor is fully charged and the left one discharged, valves are actuated to reverse the flow of sorbate between reactor modules. The newly charged reactor on the right is heated to desorb high pressure refrigerant and the reactor on the left is charged. Two disadvantages of this system are the fluctuating supply of useful heating and cooling as the modules cycle and the low efficiency that results from wasting the heat stored in the discharged sorption reactor.

The disadvantages of discontinuous capacity and low efficiency are addressed together in advanced designs. Jones and Christophilos (1993), Shelton (1993), and Ryan (1993) each describe the cycling of multiple reactors to provide heating or cooling with small fluctuations in capacity. In each of these concepts, two or more pairs of sorption reactors are used to generate a "thermal wave" in the sorbent reactors. Various ideas for regenerative loops have been investigated to recover and use some of the heat stored in the most recently discharged reactor. Jones reports that a system with four reactors is more efficient than a system with two, but that there is no significant advantage to using more than four (1993).

One possible circuiting for a regenerative fluid is shown in Fig. 4 for a four sorption reactor heat pump to transfer heat from a recently discharged (hot) reactor to the reactor be discharged (heated to 400°F). A pump and a series of solenoid and check valves are used to route the heat transfer fluid from one sorption reactor to the next. In Fig. 4 closed solenoid valves are indicated by a "⊗" and open solenoids by "●". A check valve represented by a "▲" is open so fluid can flow in that direction and one represented by a "△" is closed to flow. The refrigerant (HFC-134a) lines are not shown in Fig. 4; they have been removed to simplify the diagram.

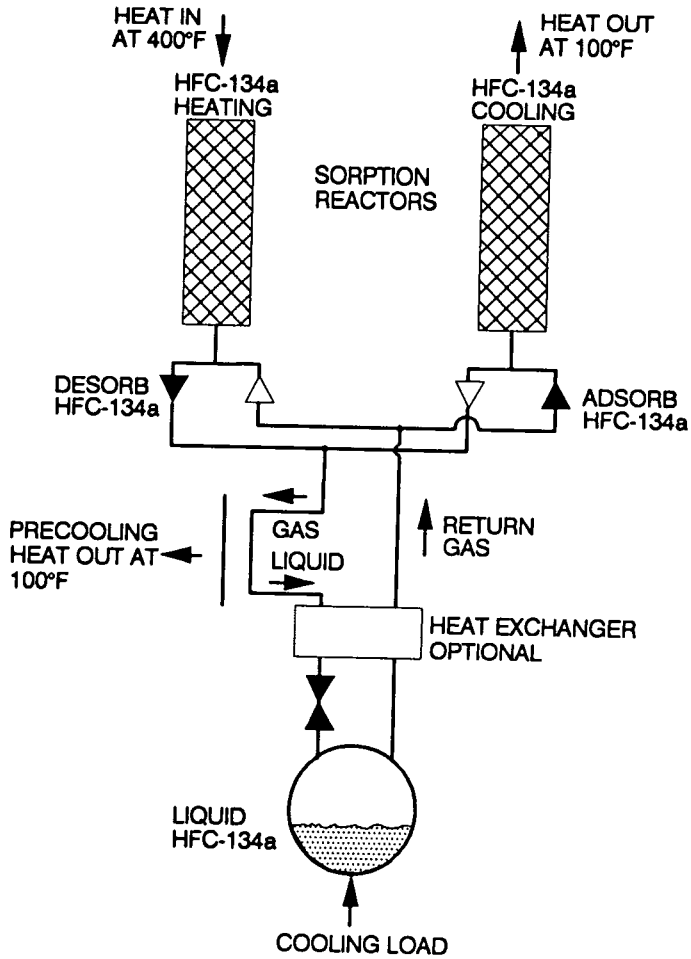

Fig. 3. Nonregenerative sorption heat pump using HFC-134a (Jones and Christophilos 1993).

Reactor #1 in Fig. 4 is adsorbing refrigerant at 38°C (100°F) while reactor #3 is being heated to discharge high pressure refrigerant. The valves have been activated so coolant can be circulated from the pump at the bottom of the schematic to reactor #1 at the top, down to reactor #4 at the bottom to pick up stored heat, to reactor #3 which is being heated to 204°C (400°F), and then to reactor #2. The regenerative fluid goes from reactor #2 to a radiator and back to the pump.

Reactors are charged and discharged successively in pairs (#1 adsorbing refrigerant and being charged while #3 is heated to desorb high pressure refrigerant, etc.) while heat stored in the "inactive" pair is recovered and used. The temperature gradients from each of the four reactors during one step of the four step cycle are indicated in Fig. 5.

State of the Technology

Working prototypes have been constructed of heat pumps for residential space condtioning and automobile air conditioners. Proponents of these technologies are optimistic about their future potential. In 1993 it was believed that heat pumps based on ammoniated salts could be ready for field testing in 1996 with long term durability testing in 1996-1997 and a commercial product on the market in 1998 (Ryan 1993); ammonia/activated carbon heat pumps could be on the market in 1996 (Shelton 1993). These are optimistic schedules which are heavily dependent on government and industry R&D support.

Adsorption using water and zeolites is also being investigated for use in automobile air conditioning. The concept is well developed and a laboratory prototype has been constructed (Schwärz 1993). According to Schwärz the prototype is "voluminous" and further design development is necessary for essential size reductions. It is unclear when a second prototype sized for installation on a vehicle can be constructed or how soon field testing and commercialization could occur. Field testing alone is likely to require several years to demonstrate technical feasibility, and should that testing be successful at least another five years would be needed before units could be incorporated into new cars and sold.

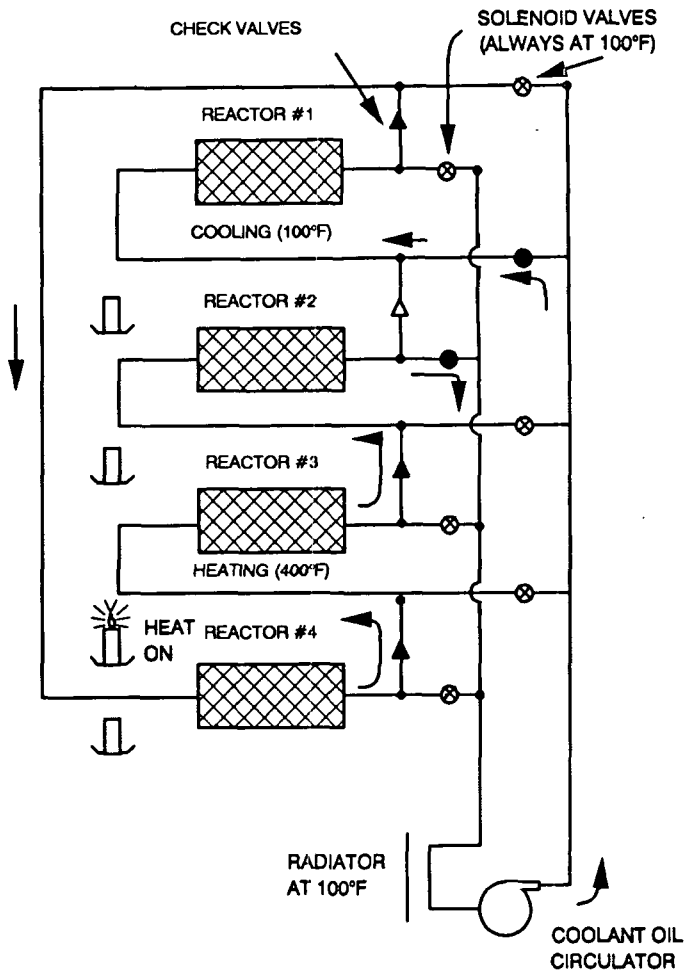

Fig. 4. Advanced fluid regenerative sorption system (Jones and Christophilos 1993).

Development Issues

Some of the technical issues facing adsorption heating and cooling are the physical size and weight of the systems, the possible deterioration of performance of solid sorbate beds as they are subjected to thousands of sorption cycles and temperature swings, and relatively low cooling efficiency relative to compression systems. Further development can be directed toward overcoming these possible obstacles to commercialization. Non-technical barriers also exist that will need to be overcome. The consumer economics of residential adsorption heat pumps will be affected not only by system first costs and operating costs, but also be the relative costs of gas and electricity. In the U.S. the gas/electric cost ratios vary widely for different regions of the country and they can even vary by time of the year. Such variations are likely to exist throughout the world and will affect the success or

failure of adsorption products in the marketplace.

Potential To Replace CFCs and HCFCs

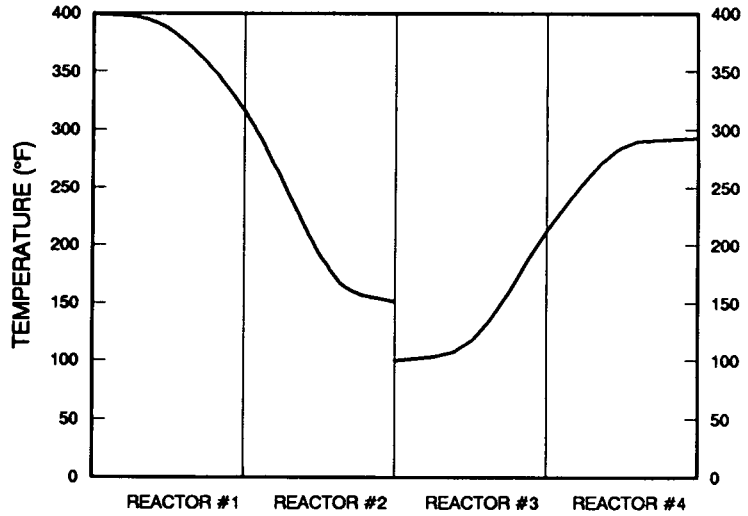

Fig. 5. Thermal gradients for advanced fluid regenerative system (Jones and Christophilos).

Adsorption heat pumps and air conditioners are not commerical products, although products based on this technology could be developed and introduced into the marketplace. The greatest potential would be in space conditioning markets dominated by heating requirements where summertime cooling is also desired (see Fig. 36). The heating efficiency is potentially higher than most gas-fired alternatives and higher than the best electric heat pumps. The cooling efficiency would compare poorly with conventional vapor compression air conditioning and consequently the adsorption heat pump would not be as attractive in applications dominated by cooling requirements. The adsorption automobile air conditioner has potentially lower global warming impacts than vapor compression systems using HFC-134a depending on how effectively waste heat can be used (see page 96). There are also major technical obstacles that must be overcome to develop either type of consumer product based on this technology (see pages 93 and 138).

ABSORPTION CYCLE

Introduction

The absorption cycle is a heat activated alternative to the conventional electrically driven vapor compression cycle. The simplest cycle uses an absorber, generator or desorber, condenser, evaporator, and solution pump. Internal heat exchangers and additional components can be added to enhance efficiency. Equipment using the absorption cycle can either be direct-fired, where a gas burner is heating the generator directly, or they can use a boiler and be steam fed. Single- and double-effect chillers are commercially available that use absorbent/refrigerant pairs as the working fluids and do not use any chemicals with direct global warming potentials.

Technology Description

A simple single-effect cycle is shown in Fig. 6. Using lithium bromide and water as a sample absorbent/refrigerant pair, the basic operation of the cycle is:

- A strong solution of refrigerant (water) and absorbent (LiBr) enters the generator. Heat is added in the generator separating high pressure refrigerant (water) vapor and high pressure liquid absorbent (LiBr)
- High pressure, high temperature refrigerant (water) vapor enters the condenser where it condenses, rejecting heat, to form high pressure, high temperature liquid refrigerant (water)

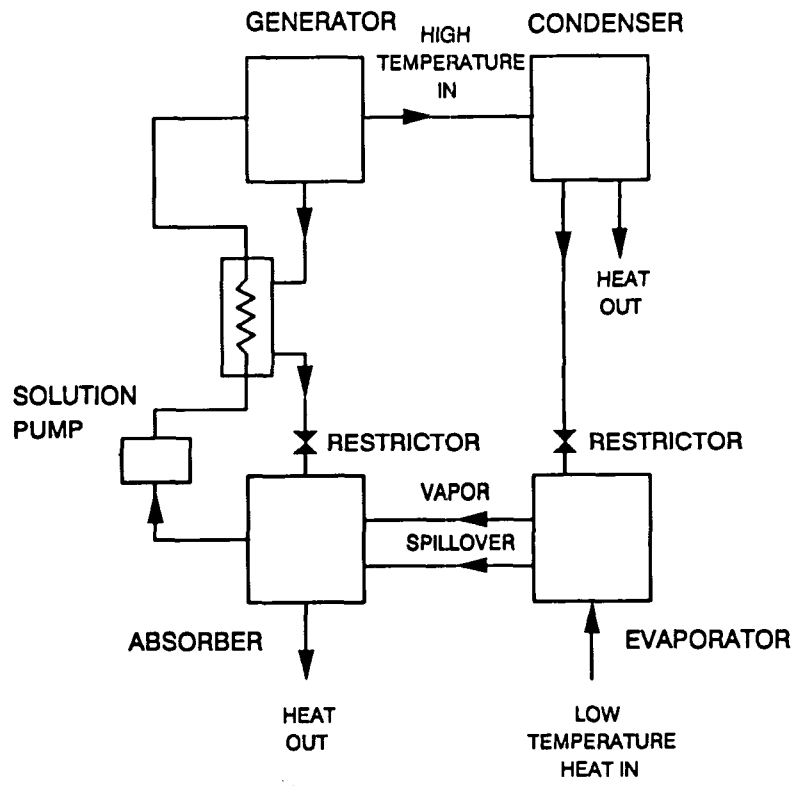

Fig. 6. Schematic for single-effect absorption cycle.

- High temperature, high pressure liquid refrigerant (water) is expanded through a restrictor creating low pressure, low temperature two phase refrigerant into the evaporator

- Low temperature, low pressure refrigerant in the evaporator accepts heat input, evaporating all, or most, of the refrigerant so there is low temperature refrigerant vapor with some liquid spillover into the absorber

- High temperature, high pressure weak solution (low concentration of refrigerant, water, and almost entirely absorbent, LiBr) leaves the generator, passes through a heat exchanger where it is cooled and is expanded into the absorber

- Low pressure weak solution in the absorber absorbs refrigerant vapor and liquid spillover from the evaporator, heat released during absorption is rejected

- A solution pump pumps concentrated absorbent/refrigerant solution from the absorber, through the heat exchanger, back to the generator.

It is important that solution concentrations and temperatures be maintained so that the absorbent does not crystalize in the line between the generator and the absorber, and that the materials and inhibitors used prevent the formation of non-condensibles, such as hydrogen, that degrade the performance of the evaporator and absorber.

While simple, the basic single-effect cycle has low efficiency and is practical primarily in applications where waste heat from another process is available as input to the generator. The efficiency can be boosted by going to more complicated double- and even triple-effect cycles or by making use of internal heat exchange to pre-heat the strong solution entering the generator or to supply some of the desorption energy for the generator. Modifications to the cycle to improve heat recuperation include the absorber heat exchanger (AHE) where heat rejected during absorption is used for heating the strong liquid solution entering the generator and the generator absorber heat exchange (GAX) where heat from the absorber is used to provide some of the heat of desorption in the generator.

Double-effect chillers have two generators with heat input to the system occurring in the high temperature generator. The high temperature vapor from the high temperature generator is used as the heat source for the low temperature generator in which the solution is concentrated and refrigerant boiled off. Triple-effect absorption chillers are under development.

State Of Technology

Both single-effect and double-effect LiBr chillers are manufactured and marketed, primarily in Japan and the U.S. Gas COPs of 1.0 are common for direct-fired chillers and 1.2 for steam-fed absorption chillers with the lower COP for the direct-fired systems resulting from combustion inefficiencies (Kohler 1993). First costs are higher than centrifugal chillers, and absorption chillers may require larger equipment rooms and condensing units (the relatively lower COPs require absorption chillers to reject more heat than centrifugal chillers so larger heat exchangers are needed). Absorption heat pumps are under development for residential and light commercial applications, but are not yet commercialized.

Development Issues

Technical issues that must be resolved in developing the next generation of absorption heat pumps include:

- developing a solution pump that has a long life, low cost, and can operate without a lubricant in a harsh environment,
- eliminating the production of non-condensible gases in the refrigerant/absorbent loop,
- reducing parasitic electric power consumption,
- developing cost effective manufacturing techniques and quality control.

The next generation of GAX equipment will push the limits of single-stage efficiency and will require the development of cost-effective simultaneous heat and mass transfer surfaces for the absorber and generator which are required to maintain the desired temperature and concentration profiles needed to achieve full performance from the sealed absorber system (Marsala 1993).

The double-effect absorption chiller is a relatively mature technology and the triple-effect cycle must be developed to make significant improvements in efficiency. Developments in heat transfer surfaces and cost reductions could improve the position of double-effect absorption chillers relative to centrifugal chillers. Significant issues need to be resolved on the triple-effect in materials, control, and design.

Potential To Replace CFCs and HCFCs

Both gas-fired absorption chillers and electric driven centrifugal chillers are commercially available and to a large extent "mature" technologies. Barriers slowing or limiting the increased use of absorption chillers result from a complex mixture of economic factors. These include the relative costs of natural gas and electricity, the efficiencies of each kind of equipment, utility rate structures (particularly demand charges for electricity), and government energy policies. Absorption chillers are common in Japan and their share of the market is growing in the U.S. Improvements in efficiency and reductions in costs could help shift this balance further toward absorption.

STIRLING SYSTEMS

Introduction

The Stirling refrigeration cycle is based on the tendency of a gas to rise in pressure when heated at constant volume. This simple fact was used in the invention of an external combustion heat engine by Robert Stirling, and the cycle derived from this work carries his name. Cryocoolers based on the Stirling cycle were commercialized in the 1950's as small, single cylinder air liquefiers capable of producing liquid air at about 80 K (-315 °F). Later, these systems broadened in size to include large machines for industrial liquefaction to small machines for electronics cooling (ASME 1988).

Research programs originating from the energy crisis of the 1970's have addressed some of the problems associated with developing engines and refrigeration devices based on the Stirling cycle. Unfortunately, the two applications (i.e. engines and refrigeration equipment) had few problems in common and much of the knowledge gained from the engine R&D projects is not relevent to the refrigeration applications.

Technology Description

The ideal Stirling cycle consist of two isothermal reversible processes and two constant volume reversible processes as shown in the center of Fig. 7. One mechanism for accomplishing the Stirling cycle is also shown in Fig. 7. By a series of piston movements, some of the gas working fluid (generally H_2 or He) is transferred back and forth between the hot and cold spaces. As shown, the Stirling mechanism consists of a power piston and cylinder connected to a displacer cylinder arrangement. A regenerator—a heat storage heat exchanger—along with heat exchangers for heat input and rejection are located as shown. Both the power piston and displacer are actuated in a manner to follow the path on the P-V diagram shown in Fig. 7.

The power piston, as the name suggests, produces work for the engine application or is driven by an electric motor for a refrigeration device. During the power stroke of an engine, this work can be used as motive power to drive equipment such as an axle in the case of a Stirling-powered automobile, or a compressor as in the case of a Stirling-powered heat pump.

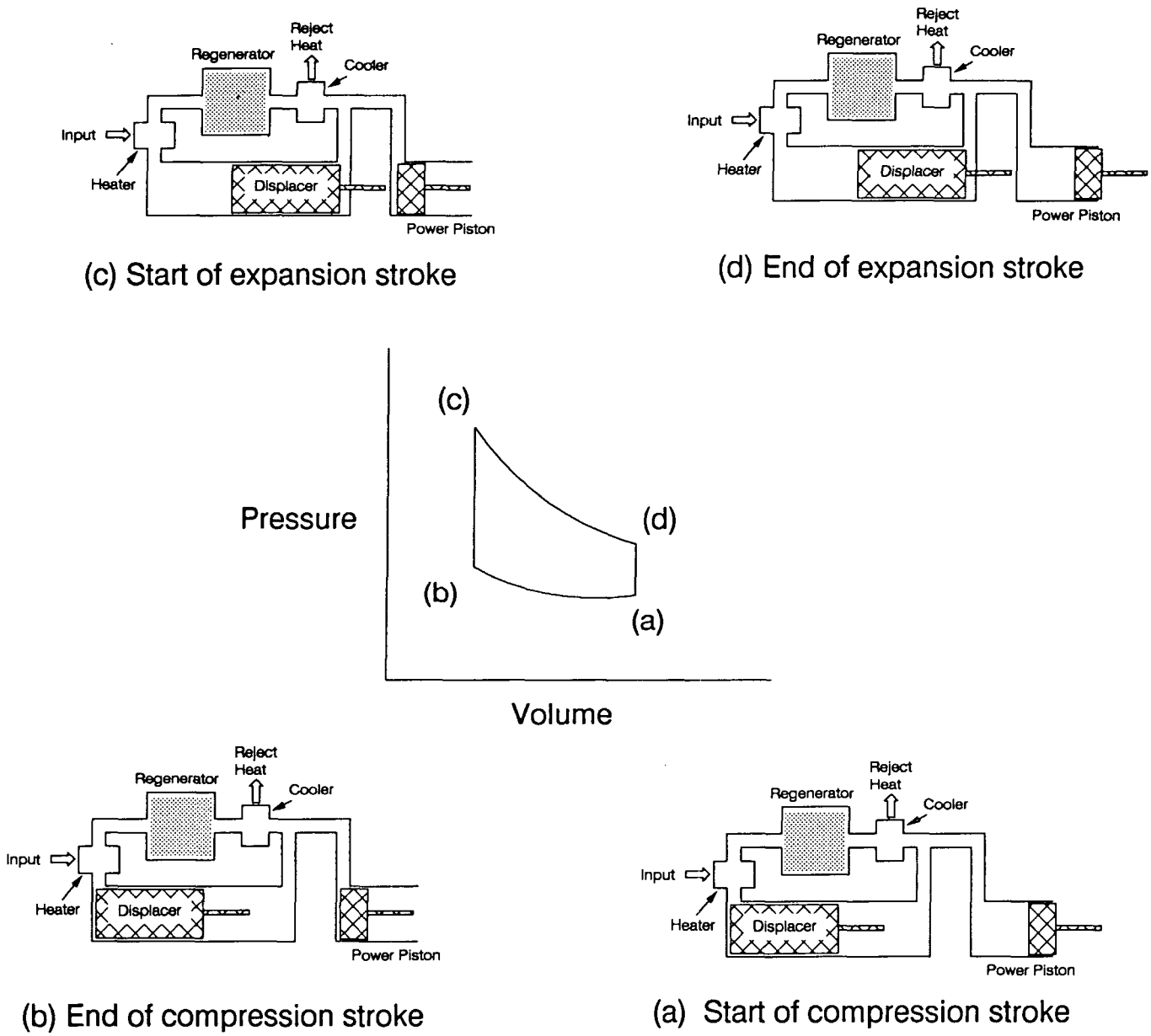

Fig. 7. Pressure/volume path and piston movements of Stirling cycle.

The displacer on the other hand, requires little energy to move it from one end of its cylinder to the other since the pressure difference is small. The cycle begins with the displacer at the hot end of its cylinder, and the power piston at bottom dead center. At point (b), the power piston has completed its compression stroke, and this isothermal compression increases the working gas pressure. From (b) to (c), the displacer piston moves to the opposite end of its cylinder, forcing the gas from the cool side of the displacer to the hot side. As the gas moves into the hot side of the displacer cylinder, it is heated both by the regenerator and the heat input from combusted fuel, and its pressure rises. In the process from (c) to (d), gas now at high temperature and high pressure in the hot end of the

displacer cylinder moves the power piston outward in an adiabatic power stroke. Along the path (d-a), the displacer piston moves from the cold to the hot end of its cylinder forcing the hot gas to the cool end of the displacer piston. Process d-a is a constant volume process since only the displacer piston moves. Furthermore, during this part of the cycle, heat is stored in the regenerator as the gas from the hot end of the displacer passes through it.

The regenerator, a key of the original patent by the Stirling brothers, is a very important component. It temporarily stores heat that would otherwise be rejected, and allows this heat to be delivered halfway through the cycle when heat must be supplied to the working gas. There is a temperature gradient established and maintained across the regenerator by the hot gas that always enters the high-temperature end of the regenerator and the cooler gas that always enters the cooler end. It has been found that a simple regenerator configuration consisting of a wire mesh or screen works well.

The preceding description of the Stirling cycle indicates the need for two pistons, a power and a displacer piston, which must move through certain motions relative to one another in order for the Stirling cycle to work. There is a wide array of techniques being used or considered for controlling the movement and phasing of the pistons in a Stirling machine. In the kinematic machines, the pistons are driven by cranks, a swash plate, or some other method to maintain and control piston motion. In the free piston configuration, the pistons are moved by pneumatic forces exerted on them, and they may or may not have connecting rods.

Like most heat engines, a Stirling engine can be operated as a prime mover by providing a heat source and heat sink, or it can be operated as a heat pump where motive power is applied to the power and displacer pistons, and heating and cooling is produced at the heat exchangers on either side of the regenerator. With this flexibility, there are a number of ways in which the Stirling cycle can be used including: (1) the duplex Stirling in which two Stirling machines—one used as a motor and the other as a refrigerator—are coupled together, and (2) the Stirling engine/Rankine cycle heat pump which is obvious.

State Of Technology

The Stirling cycle has been demonstrated as effective in low temperature refrigeration applications, and efforts have been mounted to extend the Stirling concept to residential refrigeration applications where energy efficiency is an important component which affects the commercialization potential. To understand how the performance of an existing low-temperature Stirling technology changes with application, researchers determined the performance of an off-the-shelf kinematic Stirling cryocooler operated over a range of conditions potentially suited for household refrigeration (Chen 1988). Over a 77 to 250 K (-320° to -10°F) range of cooling temperatures, the efficiency of this unit varied from 7 to 17.5% Carnot based on the ambient and cooling head temperatures. At a temperature of

250 K (-10°F), the COP of the unit was measured to be 0.369. This performance measurement was based on the quotient of cooling provided by the unit to electrical energy into the unit. More recent investigations by Berchowitz et al. (1993) showed the efficiency of a 110 K (-261°F) free-piston Stirling cryocooler developed by Sunpower to be 32% Carnot in lifting 250 W (850 Btu/h) from -26°C to 41°C (-15° to 34°F). Berchowitz noted that this compares favorably with today's domestic refrigerator Rankine efficiency of approximately 25% Carnot. An estimate of the efficiency of a Stirling refrigerator made by De Jonge and Kuijpers for the Phase I AFEAS report (Fischer 1991) indicates that an optimum design would have a COP of 1.19 under approximately these same temperature conditions. This estimate puts the Stirling optimum design efficiency at 31% Carnot—a value equal to the efficiency found in the experimental study performed by Berchowitz. If both efficiency measurements were performed on a consistent basis, it would suggest that the current Stirling technology is about as efficient as could be expected for a mature technology. In this study as well as with others examined as part of this assessment, there appears to be a paucity of information concerning how the efficiency data were determined; consequently, estimates regarding Stirling versus Rankine comparative performance cannot be generalized. The Berchowitz study does suggest, however, that Stirling and Rankine machines can have similar efficiencies. For evaporator temperatures lower than -26°C (-15°F), the Stirling machine does better; at higher temperatures, the Rankine vapor compression system shows higher efficiencies.

The free piston Stirling concept is being applied to the Vuilleumier cycle (a variant of the Stirling cycle) by the Mitsubishi Electric Corporation (Melco), and a 3 kW (10,000 Btu/h) Vuilleumier heat pump (VMHP) system based on this technology will be completed in 1995. This technology replaces the kinematic VMHP system which uses mechanical couplings between the two pistons. The newly developed system uses a linear motor to drive the high temperature piston, and resulting pressure fluctuations drive the low temperature displacer piston. Although no performance results were reported (JARN 1993) the free piston concept reportedly reduces the physical size of the device by one-half.

Characteristic Advantages

All Stirling machines have the advantage of not requiring valves or subjecting the working volume to hot combustion gases. In the case of a free-piston duplex Stirling machine (e.g. Stirling engine driving a Stirling cycle refrigeration unit) there is no requirement for hermetic piston rods and a highly loaded crankcase with associated bearings and seals. The small number of moving parts argues for long life and good reliability, although this has not been sufficiently demonstrated. Proponents of the free-piston configuration argue that its costs should be low since even in the Stirling duplex configuration there are only three moving parts, but this would need to be verified with an independent engineering cost analysis.

Without a high intensity combustion system, the lack of valves and use of straightforward mechanical balancing in the kinematic version, sound levels in the Stirling engine are anticipated to be low. However, attention should be paid to vibrations inherent in the free-piston Stirling engine/compressor system.

Development Issues

Seals are the perennial problem of Stirling machines (Walker). This issue appears less of a problem with free piston machines than with kinematic machines that use oil for a lubricant. In the case of the latter, the seal must prevent oil from entering the working space, for if it does, the oil can contaminant and block the regenerator passages. It is important to provide a seal between the displacer piston and cylinder because any gas which is allowed to pass through this gap does not pass through the heat exchangers or the regenerator Gas lubricated bearings have been studied for the free-piston design; however small clearances must be maintained.

There are also issues associated with the bearings of free-piston Stirling machines. Hydrodynamic bearings have an advantage in that they are simple, self-acting and require no external support system. Perhaps their greatest disadvantage is some surface contact dragging which is inevitable during startup. There are smaller piston/cylinder side loads with a free-piston than with a kinematic machine; consequently, wear between the piston and cylinder should be less of a concern with the free-piston than with kinematic machines. Mechanical techniques have and are being studied to maintain adequate clearance and close tolerance between the piston and cylinder walls in the Stirling machine. Flexure bearings which allow the piston to move axially and yet constrain most radial motion have been used with success. Lifetime of these mechanical components may be an issue requiring attention.

Issues of high first cost remain as barriers which must be overcome for some Stirling systems. This issue was explored in a major multi-year effort by Mechanical Technology, Inc. to develop a free-piston duplex Stirling heat pump system (Stirling engine driving a linear HCFC-22 compressor). Although several studies and redesigns of the engine/transmission/-compressor assembly were successful in reducing the anticipated cost of production units, the final expected first cost of this unit ($665) fell short of the $500 cost goal. This unit also experienced problems in meeting the design capacity. This suggests that the modifications needed to meet the performance goal may further compromise the system economics.

In general and with particular reference to free-piston designs, questions of economics remain due to uncertainties in (1) the high tolerances and extreme care and cleanliness required by the gas bearing and clearance seals, and (2) the need for components (such as air preheaters, regenerators) which must exhibit long life in a working system.

Potential To Replace CFCs and HCFCs

Arguably, one of the most important contributions from Stirling system research and development has been refinement of the linear engine (compressor). This oil-free compressor can be used in a number of Stirling concepts as well as in the Rankine cycle as a substitute for conventional reciprocating or rotary compressor technology. A major research effort has been mounted to develop this compressor for use in refrigerators (Air Conditioning News 1993), and early tests have shown a 15% overall improvement in refrigerator efficiency using linear compressor improvements developed through Stirling cycle research. Refinements in linear compressor technology predicated on this research will tend to improve future efficiencies of both Rankine and Stirling refrigeration systems.

The Stirling cycle has been a demonstrated success in low temperature applications where it is more efficient than Rankine cycle refrigeration. It has also enjoyed success in niche uses such as space-based applications where its performance is not affected by low gravity conditions.

Stirling cycle development efforts have not been successful in other applications (e.g. household refrigeration, space conditioning) because of the firmly established efficiency and reliability of conventional vapor compression equipment.

THERMOELECTRIC REFRIGERATION

Introduction

Thermoelectric refrigeration is based on the observation first made by Peltier in 1834 that an electric current passing through a circuit formed by two dissimilar conductors or semiconductors will cause a temperature difference to develop at the junctions of the two conductors. A refrigeration effect develops at the cold junction, and heat is rejected at the hot junction.

Technology Description

There are several thermoelectric phenomena associated with joining two dissimilar materials. Perhaps the most familiar is the fact that if a temperature difference is maintained between the two junctions, an open circuit electrical potential difference ΔV develops which is proportional to the temperature difference ΔT between the junctions, or:

$$\Delta V = \alpha \times \Delta T \tag{1}$$

The relationship in (1) is the basis for the widely used measurement of temperature using thermocouples. The constant of proportionality, α, is termed the Seebeck coefficient after the scientist who first described this process. The absolute Seebeck coefficient for a material is determined by using this material as one leg of a junction and lead as the reference leg. The α for the reference leg is taken to be zero. Therefore, for any material A the absolute Seebeck coefficient, α_A, is a property of that material and can be positive or negative.

If an electrical current is made to flow through a junction of two dissimilar materials, A and B, then the heat energy which is produced or absorbed at the junction is proportional to the current and to the absolute Seebeck coefficients by the relation:

$$Q = (\alpha_A - \alpha_B) \times i \times T \tag{2}$$

where T is the absolute temperature of the junction and i is the electrical current. This relation was developed by Peltier. Whether this heat is absorbed or produced depends on the direction of current flow.

The magnitude of heat flow at a junction depends on the values of the absolute Seebeck coefficient for the individual materials. In the case of metals, this coefficient does not exceed 50μV per °C (ASHRAE 1977). However, in semiconductors, α can be much higher. Some materials with the highest α ($\approx$250μV per °C) are alloys of tellurium (Te) doped with antimony tri-iodide (SbI_3) to produce an "n-type" semiconductor and with excess Te to make a "p-type" material (Houst 1982).

State Of Technology

The relation between the cooling delivered by a thermoelectric couple and its material properties and configuration can be used to determine the efficiency of thermoelectric cooling. A simple thermoelectric cooling device consisting two semiconductors and electrical connectors, which also serve as heat exchangers, is shown in Fig. 8. An electric current passes through the p-type semiconductor on the right of Fig. 8, through the finned copper or aluminum conducting strap at the top, and then through the n-type material on the left. At the cold junction at the top, heat Q_C is removed from the ambient (the cooling effect), and at the hot junction, heat Q_H is rejected.

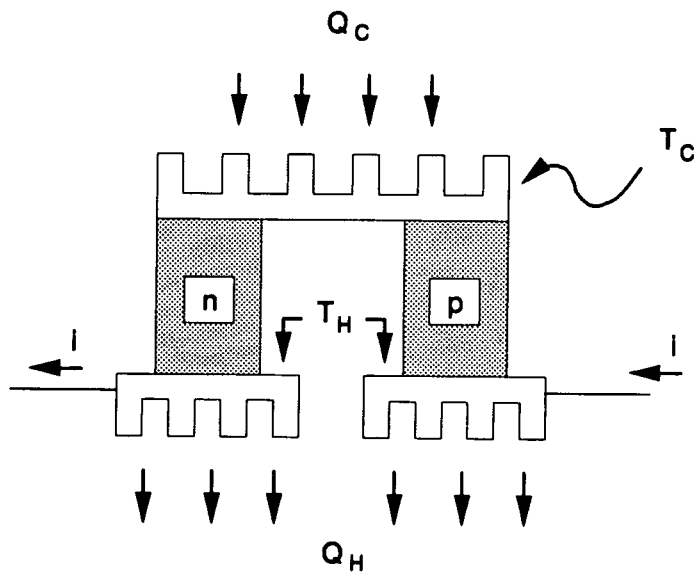

Fig. 8. Schematic for a simple thermoelectric couple.

Theoretically the maximum efficiency is given by (3) where the variable "Z" is defined according to (4), and T_H and T_C are the absolute temperatures at the hot and cold junctions. The relations in (4) shows that the maximum COP of a thermoelectric refrigerator depends simply on the Seebeck coefficients α, thermal conductivity K, and semiconductor resistivity ρ of each leg, and on the operating temperatures. The expression Z is called the figure of merit for the couple. Selecting materials with high difference in α's, low K, and low ρ raises the figure of merit and improves the maximum COP. At an average temperature T=25°C

(77°F) for space cooling applications, typical off the shelf thermoelectric modules produce a value of Z=0.0026 (Mei 1992).

$$COP_{MAX} = \frac{T_C}{T_H - T_C} \times \frac{\left[1 + Z\times\frac{T_H+T_C}{2}\right]^{1/2} - \frac{T_H}{T_C}}{\left[1 + Z\times\frac{T_H+T_C}{2}\right]^{1/2} + 1} \qquad (3)$$

$$Z = \frac{\left[\alpha_p - \alpha_n\right]^2}{\left[\sqrt{K_p\rho_p} + \sqrt{K_n\rho_n}\right]^{1/2}} \qquad (4)$$

The high performance alloys of tellurium have been found to have Z=0.003/K (Houst 1982). This appears to be about the highest figure of merit attained to date in the development of thermoelectric materials, and it was obtained under laboratory conditions. This general practical limit on the figure of merit has been observed by others.

"The superconducting materials used in today's [thermoelectric] modules have a ZT [T is the mean operating temperature, the average of T_H and T_C] of about 1.0, which is not enough. A ZT value of 3 and above is required to produce energy efficient cooling units (Mathiprikasam 1993)." At an absolute temperature of 300 K (27°C or 80°F), ZT = 1 would correspond to a figure of merit Z = 0.0033; Z = 0.0035 is the highest figure of merit attained under laboratory conditions (Mei 1994).

Theoretical limits: researchers have developed theories on the thermoelectric processes and projected limits on what values for the figure of merit may be possible with ideal materials. In developing his ideas Mahan found "the maximum value of ZT to be about one to two. This value is close to experimental reality, where it is hard to find values of ZT greater than one (1989)." Such an analysis adds emphasis to the difficulty in making the material breakthrough necessary to achieve the ZT = 3 that is cited by Mathiprikasam as necessary to produce energy efficient cooling systems.

Relative efficiencies: Fig. 9 shows both the theoretical and actual COPs across a range of temperature differences between the hot and cold sides ($\Delta T = T_H - T_C$) for a fixed hot temperature of 27°C (80°F). The solid line in Fig. 9 represents the Carnot efficiency, the dashed and dotted lines are theoretical efficiencies for thermoelectric refrigeration assuming figures of merit Z = 0.03/°C and 0.003/°C, respectively, and the long dashed line between ΔT of 20°C (T_C = 7°C or 44°F). If it were possible to increase Z by a factor of 10, to 0.03/°C, then the cooling efficiency would improve to almost 50% of Carnot. Based on slow progress in the development of high-Z thermoelectric couples, attaining these material property targets seems quite ambitious. However, even if the optimistic figure of merit is reached in a material, the efficiency of a thermoelectric cooler remains well below the

efficiency found in conventional vapor compression systems.

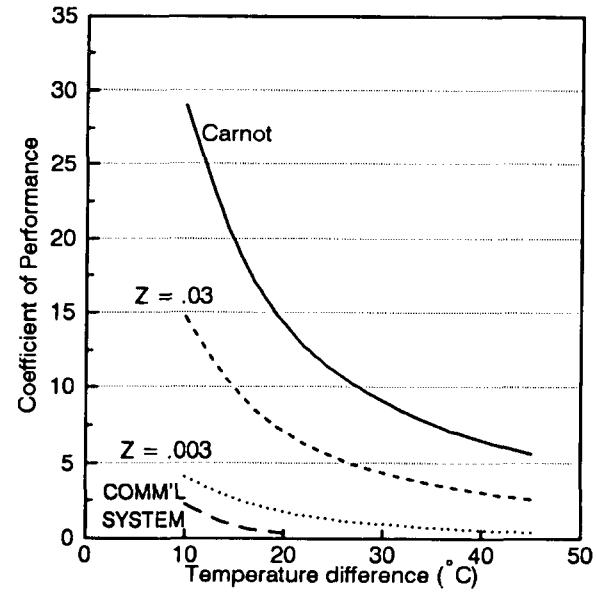

Fig. 9. Coefficients of performance for thermoelectric refrigeration.

Characteristic Advantages

Thermoelectric refrigeration has some advantages over other cooling technologies that make it the best choice for some applications. Among the advantages are:

- completely solid state construction (except for large systems employing pumps and fans for heat transfer); robust technology,
- systems can have quiet operation with the proper choice of pump and fan,
- systems can be reversed to provide useful heating, and
- low capacity systems are very small and light weight.

The major disadvantage with thermoelectric refrigeration is the low efficiency.

Development Issues

The greatest obstacle to widespread application of thermoelectric cooling in consumer applications is the relative low efficiency relative to other alternative technologies. Significant breakthroughs in materials are needed to be competitive on energy use.

Potential To Replace CFCs and HCFCs

Thermoelectric cooling will continue to be successful in niche applications where portability, size, and ruggedness are more important factors than efficiency. Refrigeration in portable coolers and spot cooling of consumer and military electronics are examples where thermoelectric cooling will be preferable to compression refrigeration. Thermoelectric cooling must achieve much higher efficiencies to be successful in larger equipment.

TRANSCRITICAL COMPRESSION SYSTEMS

Introduction

Conventional vapor compression equipment is based on the Rankine cycle and employs a refrigerant operating at a high side pressure which is below the critical point of the working fluid. The hot vapor in the high side heat exchanger condenses, forming a hot liquid which is expanded through a capillary tube, orifice plate, or expansion valve. The phase change in the condenser permits refrigerants with high heats of vaporization to be used in compact and efficient heat exchangers. Compounds like CO_2 have attractive properties as refrigerants (e.g. non-flammable, non-toxic, zero ozone depleting potential, extremely low global warming potential, good thermodynamic and thermophysical properties), yet have low critical points which limit their use in conventional compression cycles; the hot gas in the high side heat exchanger does not condense because it is above the critical point. Development effort has been directed toward designing equipment to use one of these gases at a high side pressure above the refrigerant's critical point. This section is specific to the work that has been done using CO_2 as the refrigerant because most of the R&D effort has been devoted to CO_2 for automobile air conditioning.

Technology Description

A prototype automobile air conditioner has been developed and tested which uses CO_2 as the working fluid (Pettersen 1993, Lorentzen 1993). The high side pressure of this system is above the critical pressure of CO_2, so in place of a conventional condenser the system uses a gas cooler on the high side. The low side employs an evaporator and receiver. Carbon dioxide has a significantly lower specific volume at operating suction conditions than either CFC-12 or HFC-134a and consequently a much higher capacity than either gas. This difference could permit an 80% to 90% reduction in the displacement for a compressor in the CO_2 system relative to CFC-12 or HFC-134a (Lorentzen 1993). The prototype tested by Pettersen used a 20 to 25 cm^3 (1.2 to 1.5 in^3) displacement compressor (1993); displacements range from 80 to 180 cm^3 (5 to 11 in^3) for conventional automobile air conditioner compressors designed for CFC-12 (Siewert 1983).

Although the CO_2 system operates at higher high- and low-side pressures, 100 bar (1400 psia) and 35 bar (500 psia), than an HFC-134a air conditioner, the pressure ratio across the compressor is only 2.5 to 3.5 compared to 5 to 7 for CFC-12 and HFC-134a (Lorentzen 1993). The lower pressure ratio should lead to higher isentropic efficiencies. In one series

of tests the CO_2 compressor showed an isentropic efficiency of 70% with a pressure ratio of 3.8 versus an efficiency of 50% for the benchmark CFC-12 compressor with a pressure ratio of 6.5 (Lorentzen 1993). This point is illustrated graphically in Fig. 10.

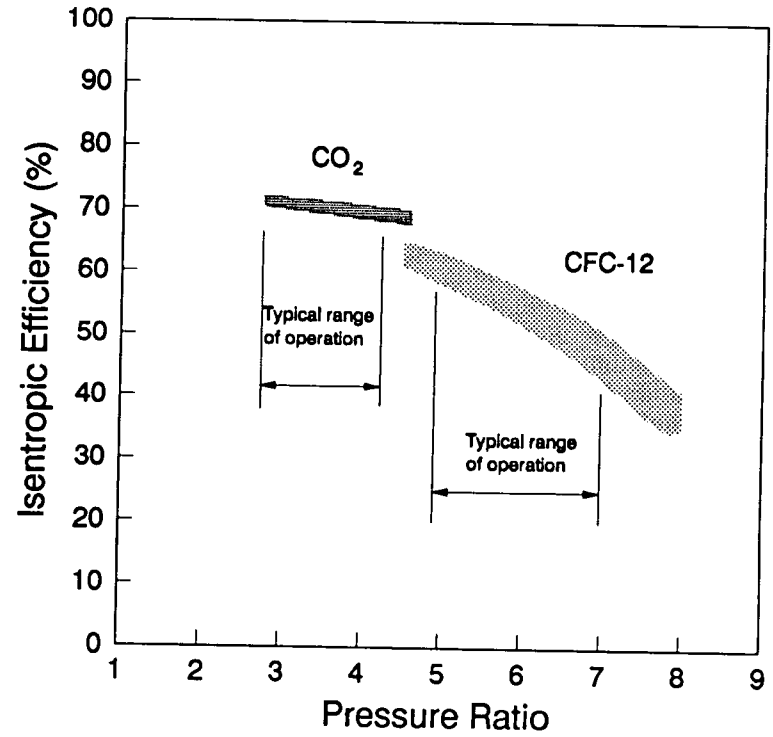

Fig. 10. Efficiency curves for automobile air-conditioning compressors (Pettersen 1993).

State Of Technology

Test Results

Some of the results from system tests presented by Lorentzen and Pettersen are shown in Table 3 along with estimates for "ideal" COPs and condensing temperatures for each fluid. These data indicate an improved performance for the CO_2 system relative to the CFC-12. The improvements can be attributed to several factors:

- comparison to a poor CFC-12 compressor (50% vs. 60-65% commonly used in 1993),
- larger air-side surface area (within the same heat exchanger volume), the refrigerant-side area is much lower than for the CFC-12 unit, and
- significantly lower condensing temperatures resulting from the higher air-side areas for the CO_2 system.

A comparison with an appropriate compressor efficiency and comparable air-side areas for both condensers would change the relative performance of CO_2 and CFC-12 (see page 91). The performance of the CO_2 system also benefits from:

- increased air-flow due to lower air-side pressure drop,
- better heat transfer characteristics for CO_2,
- higher refrigerant mass velocity, and
- the absence of a superheated region in the evaporator (Lorentzen 1993).

These factors contribute to a higher evaporating temperature and higher COP. Although evaporator refrigerant-side temperatures are higher for the CO_2 system than for conventional fluorocarbon systems, the air-side surface temperatures are not significantly

Table 3. Performance of CFC-12 and CO_2 automobile air conditioners.

Engine Speed	Condenser Inlet Air Temperature	CFC-12			CO_2		
		COP		Estimated Condensing Temperature§	COP		Estimated Condensing Temperature§
		Measured	Ideal*		Measured	Ideal*	
Low†	25°C	2.90	6.8	40°C	3.07	5.0	33°C
Low†	43°C	1.67	3.9	58°C	2.04	3.3	40°C
Low†	55°C	1.09	2.6	75°C	1.50	2.4	55°C
High‡	25°C	1.97	4.6	54°C	1.97	3.2	42°C
High‡	43°C	1.35	3.2	65°C	1.38	2.3	53°C
High‡	55°C	0.93	2.2	80°C	1.16	1.9	60°C

† 700 rpm for CFC-12 and 640 rpm for CO_2
‡ 1740 rpm for CFC-12 and 1600 rpm for CO_2
* 50% compressor efficiency for CFC-12 and 72% compressor efficiency for CO2, 15% losses for drive and blower
§ based on 4°C evaporating, 0°C superheat, and 10°C subcooling (see Fig. 24, p. 89)

different and the latent capacity is expected to be about the same as for conventional air-conditioning systems.

Characteristic Advantages

The principal advantage of this technology is that it permits efficient use of a naturally occuring, environmentally benign gas as a refrigerant. There may be benefits in the potential compressor size reduction.

Development Issues

Relative efficiency, system weight, equipment lifetime, and safety (due to the extremely high operating pressures) are all major development issues. Although the operating pressures are greatly higher than traditional fluorocarbon compression systems, the energy released in case of an accidental rupture of the system is dependent on the volume of refrigerant as well as the high pressure. The lower refrigerant-side volume of the CO_2 air conditioner reduces the stored energy so that it is typically only 60% higher than an HFC or CFC air conditioner (Pettersen 1994).

Potential To Replace CFCs and HCFCs

Successful development of consumer equipment using transcritical compression of CO_2 would not rely on CFCs, HCFCs, or hydrocarbon refrigerants, but it is far too early in the development process to speculate on how much of the market these products might capture.

Past development of this technology has led to the construction and testing of a single laboratory prototype air conditioner. This prototype has been operated and studied to demonstrate the proof of concept, but not undergone extensive testing, including endurance testing. Significant efforts are needed in further laboratory development and field testing to demonstrate its viability in a consumer product. The developers believe that properly designed components for high pressure operation should not experience any greater tendency toward wear and failure than conventional systems designed for lower operating pressures (Pettersen 1994). Test to failure is needed to demonstrate this. Acknowledging the long lead times required by the automobile industry to make major design changes, this technology probably could not be introduced in the marketplace before 2003 to 2010.

THERMOACOUSTIC REFRIGERATION

Introduction

Glass blowers have known for several centuries that sound can be generated by an imposed temperature gradient, and cryogenic researches have known the same thing for decades. Only recently, however, has anyone studied the reverse process of generating a temperature gradient through the use of an imposed acoustic wave (Garrett 1993). Current R&D activities focus separately on thermoacoustic refrigeration and acoustic compressors for application in conventional Rankine cycle equipment (see page 83).

Technology Description

The diagram in Fig. 11 shows a thermoacoustic refrigerator which uses a standing wave to create a temperature gradient to provide useful cooling. One method to generate the sound wave is to mount an ultracompliant loudspeaker at the "open" end of the tube and "bounce" the sound wave off the opposite end which is sealed. An alternative is to seal both ends and oscillate the entire tube at 300 to 400 Hz (distance of oscillation is on the order of 50 μm).

An acoustic wave is an adiabatic (constant energy) compression and expansion of a gas which causes a temperature change. If the gas inside the tube is considered to be ideal, then it can be shown that the localized temperature, T, and pressure, P, are related by (5).

$$\frac{dT}{T} = \left[\frac{\gamma - 1}{\gamma}\right] \times \frac{dP}{P} \qquad (5)$$

where γ, the polytropic coefficient, is the ratio of the specific heat of the gas in the tube at constant pressure and the specific heat at constant volume. The relation in (5) shows that the pressure increase, dP, from the adiabatic compression of the gas results in an increase in the localized temperature, dT. A "regenerator" system (or "stack" as shown in Fig. 11) is located at a pressure antinode where the pressure change in the gas is large in order to make use of this change in temperature.

The regenerator is simply an open structure that allows the sound wave to pass through, but which serves as a means for storing and releasing heat from the temperature change that

results from the pressure oscillations. Each gas "parcel" passing through the regenerator transfers heat from the opposite end of the tube to the regenerator, creating a temperature gradient along the length of the tube. A cooling load placed on that end and a heat rejection system affixed to the end with the regenerator forms the basis of a thermoacoustic engine capable of refrigeration.

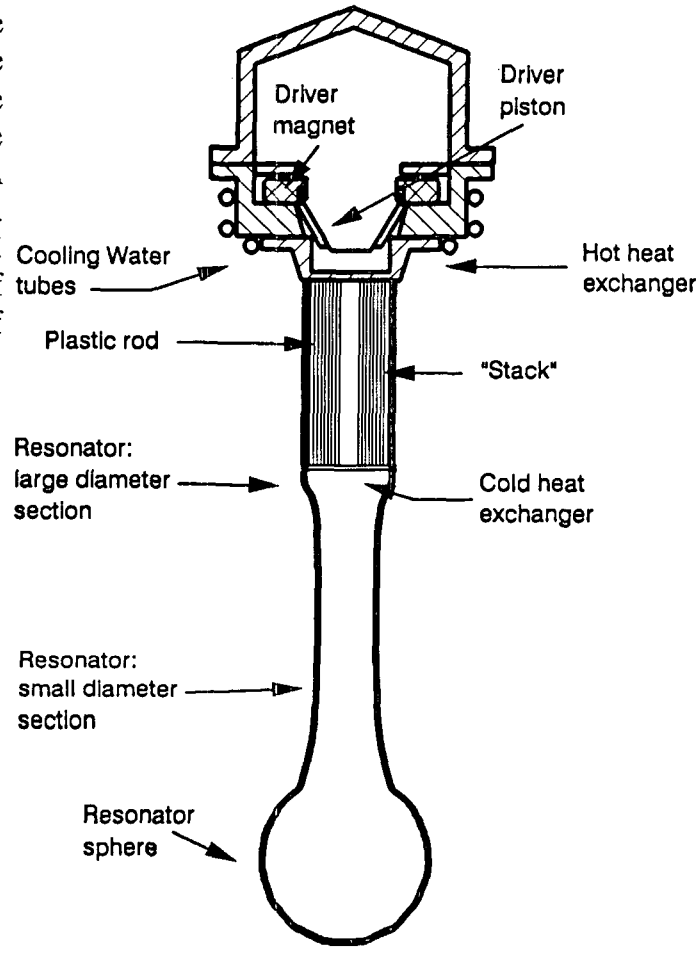

Fig. 11. Thermoacoustic refrigerator.

State Of Technology

Small thermoacoustic refrigeration systems have been built and tested at Los Alamos National Laboratory and the Naval Postgraduate School in California. These systems use a mixture of helium and argon pressurized to 10 atmospheres as the sound medium. The regenerator is a cylindrical stack created by rolling mylar with monofilament line spacers between subsequent layers of mylar. Heat exchangers located at each end of the stack accommodate heat transfer from a source at the cold end of the stack to the hot end. Test data from an early prototype thermoacoustic refrigerator showed that the highest efficiency was 12% of Carnot. At this efficiency, the unit provided 3 Watts of cooling at a temperature of -29°C (-20°F) and a sink temperature of 25°C (77°F) (Swift 1988). Other designs have produced efficiencies as large as 20% of Carnot (Susalla 1988).

Characteristic Advantages

The principal advantages of the thermoacoustic refrigeration system are the use of helium as the working fluid, a single moving part which does not require lubrication, and simple variable capacity operation. The only moving part is the diaphragm of the loudspeaker, which eliminates the need for lubrication. In principle, acoustic driver systems can be developed which convert electrical energy to acoustic energy with efficiencies around 90%

at the 150 to 200 decibel sound levels (Garrett 1993). Variable capacity can be achieved by increasing or decreasing the amplitude of the standing wave (e.g. the sound level) to match the imposed load or the operating conditions (e.g. condenser and evaporator temperatures in response to the ambient temperature).

Development Issues

Technical issues exist that require further development work before there can be practical application of thermoacoustic engines for refrigeration. Useful refrigeration is provided by heat conduction from the heat exchangers at each end of the stack. The size of these heat exchangers is limited by constraints on the geometry and shape of the resonant cavity needed to maintain the standing acoustic wave. This results in limited flexibility in choosing heat exchanger size to reduce approach temperatures and improve overall system performance and efficiency. Second, the acoustic power levels needed to develop the necessary pressure variations for domestic refrigeration systems tend to induce shock waves that impose losses on the system and reduce efficiency. Resonant cavity design must be carefully chosen to reduce the tendency for shock waves to be created. This is an issue for thermoacoustic refrigerators and acoustic compressors. Other issues related specifically to acoustic compressors must also be addressed (see page 84). Work is needed to boost electroacoustic conversion efficiencies and secondary heat transfer loop performance for refrigeration loads greater than 100 W (340 Btu/h) (Garrett 1993).

Potential To Replace CFCs and HCFCs

Thermoacoustic refrigeration systems use helium as the working fluid and have limited potential for replacing conventional compression systems using CFCs and HCFCs in domestic refrigeration, space conditioning, and automobile air conditioning. Prototypes have been built and operated and have attracted interest among manufacturers. Reported efficiency levels are low relative to conventional positive displacement compression efficiencies, but there is no fundamental reason why efficiencies cannot be improved with sufficient development. Industrial partners have expressed interest in participating in the development and application of acoustic technologies and the pace of development is likely to accelerate. Projections for commercialization of thermoacoustic refrigeration are imprecise (Garrett 1993), but unlikely before 2000.

EVAPORATIVE COOLING

Introduction

Evaporative cooling is often used in dry climates instead of vapor compression air conditioning. Over four million residential evaporative coolers are in use in the U.S.; approximately ¾ of these are in the western states (Foster 1993, DOE 1992).

Technology Description

An evaporative cooler works to lower the dry bulb temperature of the air in the conditioned space, either directly or indirectly, by evaporating water from a spray or a porous media. The sensible heat of the air is used to evaporate water, reducing the air temperature, and providing a more comfortable air supply. Evaporative coolers work more effectively in dry climates than in more humid areas. Evaporative coolers use only fans and small water pumps so their energy use is small relative to conventional air conditioning.

Several different designs of evaporative cooling systems are common.

Direct Evaporative Cooling

A direct evaporative cooler uses a spray or porous wetted pads to provide adiabatic evaporation of water into the supply air stream. Direct evaporative cooling requires higher air flow rates than vapor compression air conditioning and hence different duct sizes. Comfort cooling is accomplished by the combination of reduced dry bulb temperatures and cooling sensation provided by higher air flows. Direct evaporative cooling increases the humidity of the inlet air and hence is only practical in low humidity areas; comfortable space conditioning can be accomplished in about 40% of the geographic area of the U.S. through direct evaporative cooling (Foster 1993). Direct evaporative cooling is used primarily in residential applications.

Indirect Evaporative Coolers

The primary (supply air) for an indirect evaporative cooler is cooled sensibly without adding humidity to the air. Supply air is cooled through use of a heat exchanger where a secondary air stream is cooled adiabatically by evaporation of water. In arid regions, indirect evaporative cooling can provide a larger capacity than direct evaporative cooling because

it allows a larger overall temperature difference (Mathur 1992). Indirect evaporative coolers are primarily used in niche and commercial applications such as offices and restaurants.

Indirect/Direct Evaporative Coolers

Many different configurations of staged indirect and direct evaporative cooling units can be put together and can also be used in combination with vapor compression cooling in high humidity, high cooling load climates. Indirect/direct evaporative cooling systems are primarily limited to small to medium sized commercial applications because of the high first costs.

Evaporative Precoolers

Evaporative cooling can be used in conjunction with air-cooled vapor compression air conditioners such as chillers and rooftop unitary equipment to reduce the load on the vapor compression equipment (allowing smaller sized equipment) and to enhance its efficiency. Precooling can add 5 to 15% to the capacity (Foster 1993), depending on geographic region; it can also provide energy savings of 5 to 10% in the eastern and Gulf coast regions of the U.S. and along the Pacific coast, 10 to 15% in the Great Plains, 15 to 20% in the Rocky Mountain states, and over 20% in Nevada, Arizona, and the non-coastal portion of California (Foster 1990).

State Of Technology

Evaporative coolers are a mature technology and products are commercially available. Designs are evolving to increase the market share of evaporative coolers relative to vapor compression air conditioners. These efforts include development of indirect/direct evaporative air conditioning and desiccant/evaporative cooling hybrid systems so evaporative cooling can be used effectively where direct evaporative air conditioners are unacceptable (geographic areas with higher relative humidities).

Characteristic Advantages

Evaporative coolers do not use ozone depleting or global warming gases as refrigerants and can use much less energy than vapor compression air conditioning to provide the same comfort levels.

Development Issues

The major limitations to the use of evaporative cooling is performance degredation in high humidity climates.

Potential To Replace CFCs and HCFCs

Evaporative cooling is an established, commercially available technology which is limited in its potential to replace vapor compression equipment using fluorocarbon or hydrocarbon refrigerants because of low humidity requirements. Manufacturers are working to expand their market share, at the expense of compression systems, and to expand into higher humidity regions. Gradual gains in market share may occur, but it will be slow and difficult to effect major shifts in the market because of the low population densities in favorable climates.

DESICCANT COOLING

Introduction

A variety of space cooling methods based upon evaporation or condensation of water into an air stream have been studied, and some of these methods have resulted in commercialized systems. In dehumidification, the latent component of the space cooling load can be accomplished by chemical sorption of a portion of the water vapor contained in the air. This must be accompanied by subsequent drying, usually by heating the sorbing material, or desiccant. Dehumidification both reduces the required size of the conventional vapor compression machine and also improves its efficiency by allowing the evaporator to operate at temperatures higher than what would normally be needed for dehumidification.

Technology Description

Solid- and liquid-based desiccant dehumidification methods have been used. In liquid dehumidification systems, the air stream is contacted with a desiccant liquid which can absorb a portion of the water vapor in the air stream. Solutions of lithium salts and various glycols have been used with some success. In solid dehumidification systems, the air stream is contacted with a solid which removes a portion of the moisture in the air by a sorption process.

Desiccant Materials

Since all materials attract and hold water vapor (i.e. are drying agents), they can be considered "desiccants." However, unlike substances such as wool and nylon which can hold about 20% of their dry weight in water, a commercial desiccant can hold up to 10 times its dry weight of water. These materials generally have very large total surface areas per unit of mass. For example, silica gel desiccant adsorbents with surface areas of more than 4600 m^2/g (50,000 ft^2/g) are available. The adsorption process always involves a sensible heat transfer to the desiccant which is larger than the latent heat of the water vapor taken up by the desiccant. This additional heat occurs through the action of surface forces between the desiccant and water vapor and varies between 5 and 25% of the latent heat of vaporization of water. Consequently, as the desiccant adsorbs water vapor, sensible heat (largely the heat of condensation) is transferred to the desiccant and it becomes warmer.

Desiccant materials can be classified as silicas, aluminas, zeolites (from the Greek translation for "wet stones"), hydrateable salts, mixtures, and polymers. The silica materials commonly referred to as gels, have been treated so that they are hydrophilic. They are formed by precipitation from water or other solutions, and the pore size and distribution can be customized to some extent. Aluminas are comprised of oxides and hydrides and are manufactured in much the same way as silicas. Zeolites are aluminosilicate complexes; they occur naturally as minerals, or they can be made in the laboratory. Microscopically, zeolites have a cage-like crystalline structure and each molecule of the adsorbent is held inside a cell according to the molecular diameter of the adsorbent. This feature provides synthetic zeolites with the capability to separate gas species based on molecular size. As a result, these materials are sometimes referred to as molecular sieves.

Hydrateable salts are a special desiccant form. They can exist as solids over several levels of hydration, or they can be salts that are more or less soluble in their water of hydration. This class of material forms the basis for liquid desiccant systems.

As the name indicates, desiccant mixtures can be prepared to achieve overall properties that are not available in a single material. For example, desiccants that can provide low humidity levels can be mixed with ones that have large total uptakes to improve the overall properties.

Finally, solid polymers have been found to have good potential for use as desiccants. Long molecules such as those of polystyrenesulfonic acid sodium salt (PSSASS) are twisted together so that the material can bind several water molecules. The spaces between the molecules can contain condensed water thereby providing additional sorption capacity (ASHRAE 1993).

Desiccant Sorption Processes

The sorption processes within a desiccant are controlled by two kinetic steps. The first is adsorption of water molecules at the surface of the desiccant. As this is a surface phenomenon, this rate can be quite rapid. The second process is one of diffusion of water vapor away beyond the surface of the desiccant and into its bulk volume. In the adsorption process, the desiccant heats up at the surface, and a thermal profile is established as heat is carried into the desiccant. The rate at which water vapor permeates beyond the surface of the desiccant (i.e. the sorption rate) is determined by the desiccant local temperature and local water vapor concentration along with the thermophysical properties of the material itself. Consequently, promising desiccant materials must possess the correct combination of several thermophysical parameters along with stability of these properties as well as stability of the desiccant material itself.

It has been found that one of the more important determinants of desiccant potential is the shape of the isotherm. Isotherms are generally plots of fraction loading (weight of water per unit weight of desiccant) against the degree of saturation (i.e. relative humidity). Through

research supported by computer modelling, Collier et al. (1986) concluded that if an "ideal" desiccant could be identified, the thermal COP of a single stage desiccant cooling system can be improved from 0.85 to 1.05. This work led to the idea that there is an "ideal" desiccant as characterized by its isotherm. The Type 1M (moderate) isotherm was proposed as the optimum shape for this "ideal" desiccant, and the sorption process for the desiccant chosen should follow this isotherm from 5% to 60% relative humidity for best results (Collier 1989). The Type 1M desiccant characteristic is a compromise between the gas and solid interactions during adsorption and ease of regeneration during desorption. Fig. 12 shows the isotherms for several desiccant types as characterized by generic equilibrium sorption characteristics.

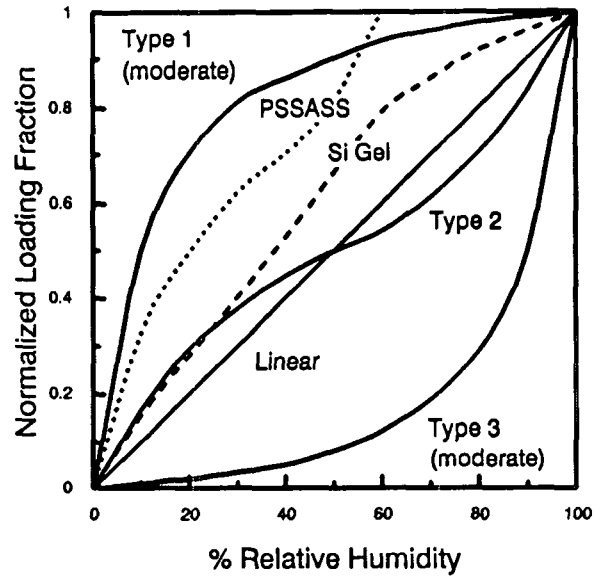

Fig. 12. Comparison of desiccant isotherms.

Water pickup in the Type 3M desiccant for example becomes significant only under high relative humidity conditions. Generally speaking, a Type 2 desiccant reaches about 50% loading at 50% relative humidity, although this is not a firm rule. With the Type 1M desiccant, water pickup changes most rapidly with relative humidity under low relative humidity conditions. This figure also shows the isotherms of silica gel, one of the most common desiccants, and the PSSASS polymer. Based on Collier's notion of an "ideal" desiccant, the polymer system would appear to be advantageous.

Although much work has been done and is continuing to identify the "ideal" desiccant, the concept of the ideal desiccant and the importance of the shape of the Type 1M desiccant on the performance of a desiccant cooling system is not universally embraced (Farooq 1991). It was suggested that the effect of isotherm shape on moisture removal rate can be adequately compensated by appropriate adjustment of the cycle time between adsorption and regeneration. This can be done by careful attention to system design.

Although they have not received as much R&D attention in the past, liquid desiccants have also been studied and are being used in some desiccant cooling systems (Pesaran 1992). Leading commercially available liquid desiccants are lithium chloride solutions and triethylene glycol. They have perceived problems of corrosion or carryover in HVAC applications although they have been demonstrated to be quite satisfactory for industrial and some commercial dehumidification applications.

Desiccants can be particularly useful in air conditioning systems where one of the following conditions must be met:

- the latent load is large in comparison to the sensible load,
- the cost of energy needed to regenerate the desiccant is low when compared with the cost of energy to dehumidify the air by dropping its temperature below the dew point with a vapor compression system,
- the moisture level required is below practical levels attainable by using conventional vapor compression equipment operating at practical temperature levels,
- there is a requirement for continuous subfreezing temperature air delivery, or
- it makes sense to decouple sensible and latent cooling loads.

In these cases, the cost of operating a vapor compression system to accomplish the objective may be quite high, and a desiccant-based system could prove to be the best option. Desiccant systems are particularly attractive in supermarket applications for dehumidification where conventional electric-driven air conditioning systems are unable to keep the air in the store dry enough to absorb moisture, particularly from food cases. Unchecked, this excess moisture is attracted to the frozen food cases where it condenses, potentially forms frost and requires extra defrost cycles. A two-stage desiccant unit has been developed (Mei 1992) and used in supermarket applications to keep the relative humidity at 35 to 40% rather than the usual 50 to 55%. As a result, the conditioned air is dry enough to absorb the moisture generated in the store as well as the moisture from ventilation air. The ramifications of using this two-stage desiccant system in supermarkets is a 30-45% reduction in energy consumption for refrigeration than available with a conventional vapor compression system.

Open Cycle Desiccant Cooling Systems

The most common desiccant device is an open cycle system for space dehumidification and cooling. The basic open cycle system was patented by Pennington (the Pennington cycle) in 1955 (Pennington 1955). This cycle shown in Fig. 13 is used to cool and provide ventilation air to a building.

Outside air is passed through a rotary desiccant bed where it is dehumidified and becomes hot. The hot, dry air at point 2 is cooled by the counterflow heat exchanger, and cooled further by the evaporative cooler as it enters the conditioned space at point 4. Ventilation air from the space is first cooled by a second evaporative cooler, and then exchanges heat from the entering airstream so that its temperature is raised. A heat source (gas, solar, waste heat) is added to this air stream to raise its temperature (and reduce its relative humidity). The hot, somewhat dry air enters the rotary desiccant wheel where it is used to regenerate the desiccant.

A recent example of this type of system is being developed by ICC Technologies using a desiccant material discovered by Englehard Company (Coellner 1993). Like other systems,

this desiccant wheel system uses natural gas for desiccant regeneration. The desiccant itself is a recently-discovered molecular sieve made of titanium silicate. Reportedly, this material acts like a zeolite at ambient temperatures, and like a silica gel at elevated temperatures. Its principal advantage lies in the fact that it can be regenerated at lower temperatures than standard materials. Reportedly, the Englehard titanium silicate (ETS) material can be regenerated at about 82°C (180°F) instead of the 175°C (350°F) required by present-day systems (Duffie 1992). The ETS material also possesses a heat of absorption that is significantly lower than for zeolites. The sorption heats for typical zeolites ranges up to 300% of the latent heat of vaporization of water. The heat of sorption of the ETS material is about equal to the latent heat of vaporization of water. This would tend to validate the fact that the material can be regenerated at somewhat lower temperatures than the more conventional materials. In this system, supply air passes through the rotating wheel and reportedly more than 90% of the moisture is removed by the desiccant. Controlled addition of moisture back into the air reduces its temperature to provide the cooling needed. Initial specifications for the demonstration units indicate a cooling COP = 0.9. Research is underway to improve the desiccant for better performance at lower regeneration temperatures with an aim of combining this system with a conventional vapor compression cycle. Condenser heat from the vapor compression cycle would be used to regenerate the desiccant.

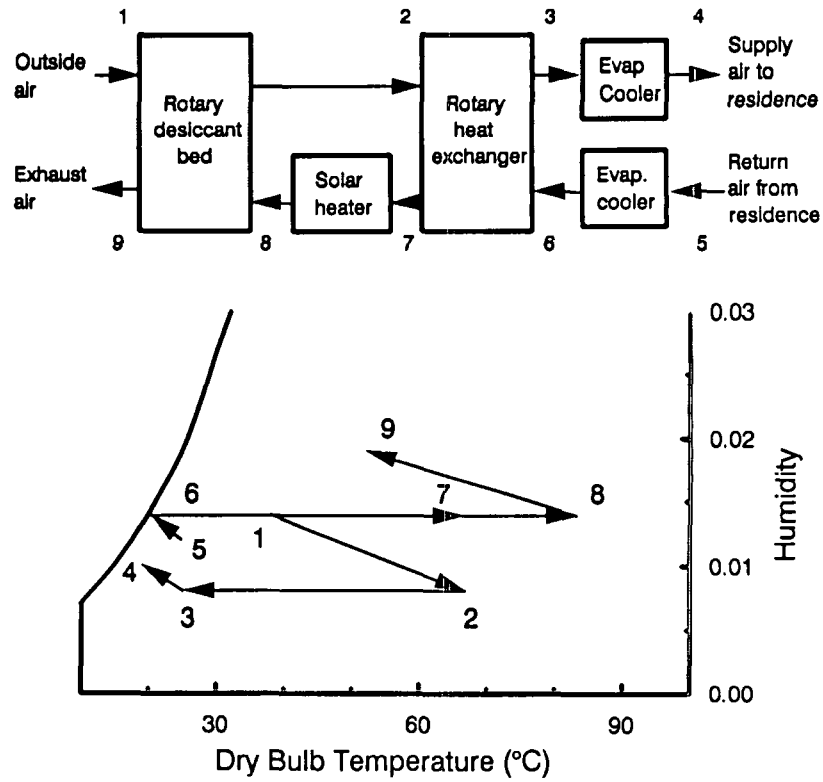

Fig. 13. Psychrometric chart and schematic of a desiccant cooling ventilation cycle.

State Of Technology

In the context of desiccant cooling systems, the system efficiency or COP is usually assumed to be the ratio of the cooling delivered to the heat needed for desiccant regeneration; that is, electrical parasitics are ignored. Many predictions of the performance of desiccant cooling

systems have been made. The performance has been found to be a strong function of the heat exchanger wheel effectiveness. With effectiveness values assumed to be 0.90 or better, most of these analyses indicate that the COP ranges from 0.5 to about 1.1 in systems that use staged regeneration (Pesaran 1992). Laboratory performance measurements conducted by the Institute of Gas Technology and Exxon Research/Engineering Company on staged regeneration systems showed that COPs near 1.1 could be achieved. American Solar King introduced a packaged desiccant cooling/heating system (the Energymaster) for residential applications, and a thermal COP of 1.1 was also achieved.

The efficiency of desiccant cooling systems is most affected by several parameters. Effective sealing of the desiccant wheel to prevent air leakage and to allow the wheel to turn relatively freely is an issue. In addition, the thermal heat capacity of the desiccant wheel has an adverse effect on the thermal performance of the system since heat is carried from the cooler side of the loop to the hotter side as the wheel rotates. With greater wheel rotation rates, this loss mechanism grows. Since the thermal performance is affected by the geometry of the wheel itself, the size of the wheel and the rotational speed, a compromise between these various parameters must be made to maintain the overall system efficiency. Presently, work continues in the laboratory to develop better wheel materials and geometries so that the thermodynamic performance of the system can be raised.

Characteristic Advantages

Development Issues

Technical Issues

Efforts to push the efficiencies of desiccant systems to higher levels raise issues that need to be addressed concerning long term performance (Belding 1993):

- how is the performance of the desiccant material itself affected by repetitive adsorption/desorption cycles (typically a desiccant wheel will be exposed to 15,000 to 20,000 regeneration cycles per year),

- what are the long term effects of dust build-up in the channels of a desiccant wheel and what maintenance practices are effective at maintaining peak performance,

- how are all of the materials, particularly those used for seals, affected by continuous high temperature operation and by temperature cycling.

Finally, rating procedures and standards may not be adequate for use in comparing different types of equipment (Belding 1993).

Potential To Replace CFCs and HCFCs

Open cycle solid desiccant cooling systems have been studied analytically and experimentally. These system use desiccant dehumidification to accommodate the latent cooling load combined with heat exchangers for sensible cooling. With the simplest cycles, COPs of around 1 have been achieved, and with advanced, multistage components, COPs of about 1.4 were obtained.

Liquid desiccant cooling systems offer greater design flexibility than solid systems and since the desiccant can be moved around, this system can be installed in situations where the supply and exhaust ducts are widely separated. Liquid desiccants can have either carryover or corrosion problems if not properly designed. The thermal COP of liquid desiccant systems is usually lower than for solid systems. However, recent progress in liquid-based systems has narrowed the performance gap. Solid and liquid systems utilizing open cycle desiccant cooling are being commercially developed, and several packaged systems are expected to be on the market by 1995. Integrated desiccant cooling systems which combine a desiccant system with a vapor compression system have been successfully installed in some commercial buildings. Another successful application of integrated cooling systems in the HVAC market has been in supermarkets.

For desiccant-based systems to penetrate the HVAC market further, their capital costs must be lowered. Improved performance through better performing desiccants, better attention to improving the thermodynamic performance of the system by reducing heat and mass losses will also contribute to more rapid market penetration.

MAGNETIC HEAT PUMPS

Introduction

Magnetic refrigeration is based on the magneto-caloric effect. This effect causes certain magnetic materials to warm adiabatically upon application of a magnetic field and cool when the field is removed. This process is highly reversible. Gadolinium metal is one magnetocaloric material for which this process occurs near room temperature. The maximum adiabatic temperature change (for gadolinium) of about 2°C (3.6°F) per Tesla of magnetic field occurs around 20°C (68°F) (a magnetic field of 1 Tesla is more than 10,000 times the magnetic field of the earth; cryogenically cooled superconducting magnets can produce peak fields of approximately 7 Tesla). Since applications at room temperature conditions, such as refrigeration, require temperature spans of 20°C (36°F) or more, regeneration is used to increase the temperature span to the desired value.

Magnetic refrigeration technology remains in the proof of concept or demonstration phase, and the planned retail size refrigeration system remains in concept/design phase. Two major obstacles constrain development of commercially available heat pumps that use magnetic refrigeration technology:

- the need for cryogenic cooling of the superconducting system, and
- development of low cost superconducting magnets and low cost magnetocaloric materials.

While most current research indicates that the high projected initial investment cost of a retail-size system may not be justified by the projected life-time return on investment over the near term, some industry sources hold that practical magnetic heat pumps could be ready for commercial distribution by the end of this decade.

The energy efficiency of a magnetic heat pump has not been demonstrated in working prototype equipment.

Technology Description

The magnetocaloric effect is due to the change one entropy component of a magnetic material as magnetic dipoles align themselves along the applied field lines or become randomly oriented as the field is removed. Since entropy is a measure of disorder in a thermodynamic system, the alignment process produced by an externally applied field increases magnetic order and therefore, reduces magnetic entropy. If the magnetic material is thermally isolated (i.e. if adiabatic conditions exist), the applied field raises the temperature of the material, and if magnetization proceeds isothermally, heat is rejected. The temperature rise in the adiabatic case is generally small—perhaps 1° to 2°C (1.8° to 3.6°F) per Tesla (T) of magnetic field strength.

In the case of ferromagnetic materials, the Curie point is that temperature below which spontaneous alignment of magnetic dipoles begins to outweigh the disordering effect of temperature. Therefore, above the Curie temperature, magnetic domains do not exist and there is no net magnetization unless an external field is present. Therefore, it is reasonable to expect that the magnetocaloric effect behavior depends on the type of material: ferromagnetic with magnetic domains below the Curie point, or paramagnetic without domains. Gadolinium, a rare earth material, is ferromagnetic and has been considered as one of the more promising magnetocaloric effect materials. If gadolinium is magnetized adiabatically, its temperature rises and reaches a peak (cusp) at the Curie point (20°C, 68°F) as shown in Fig. 14 (Brown 1981).

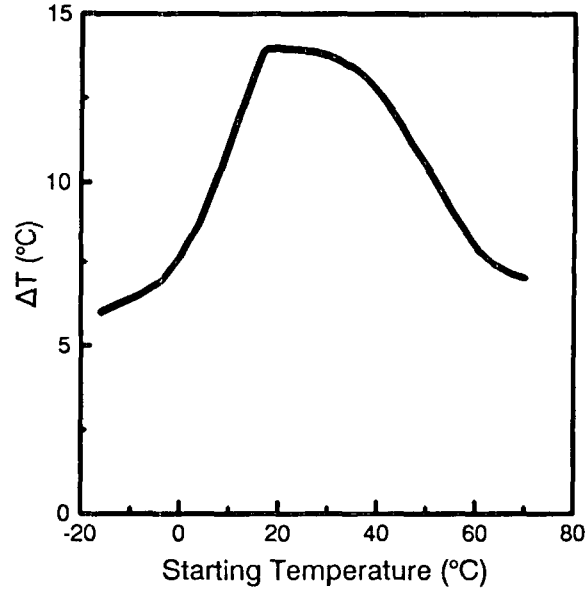

Fig. 14. Adiabatic MCE in gadolinium with an applied 7T field.

Above and below the Curie point, the strong (7T) external magnetic field increases the alignment of magnetic dipoles in the gadolinium which decreases the associated entropy component and raises the temperature of the material. Below the Curie temperature, the external field causes magnetic domains present in the material to align with the field and also increases the dipole alignment within each domain. Above the Curie temperature, there is no domain in the magnetic material; however, the magnetic field aligns individual dipoles so that the magnetization is proportional to the applied field as in a paramagnet. In either temperature range, the presence of the external magnetic field increases the alignment of the magnetic dipoles (increases magnetic order) and consequently decreases the entropy. At the Curie point, the entropy change with external magnetic field strength is a maximum giving rise to the cusp shown in Fig. 14.

Therefore it seems reasonable that a heating or cooling system based on the magnetocaloric effect in a ferromagnetic material should operate over a temperature range near the Curie point if at all possible. With a Curie point of 20°C (68°F), gadolinium has garnered much interest as a candidate material for magnetic cooling or refrigeration in the room temperature range.

Magnetic heat pumps, like all heat pumps, operate in a cycle by removing heat from a cool reservoir, and discharging that heat along with the heat equivalent of applied work to a reservoir at a higher temperature. The T-S diagram for the working material is a useful tool for understanding magnetic heat pump cycles. The temperature entropy (T-S) diagram in Fig. 15 depicts a generic paramagnetic material at various stages of magnetization (Waynert 1989). Overlaid on this diagram are two adiabats and isotherms which form a Carnot cycle.

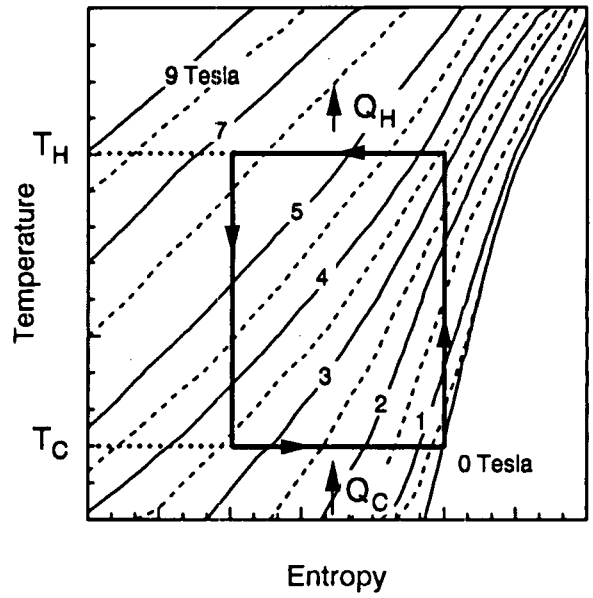

Fig. 15. Temperature-entropy behavior of a magnetic material undergoing a Carnot cycle.

The heat pump (refrigeration) action may be described by starting at point 1 with the magnetic material in thermal contact with a sink at temperature, T_H. As the applied magnetic field is increased, the entropy of the material decreases and heat Q_H is transferred isothermally to the sink. Beginning at point 2, the magnetic material is isolated and demagnetized adiabatically so that its temperature falls to T_C. From point 3, the material now in contact with the cool reservoir is further demagnetized resulting in an isothermal increase in entropy as heat, Q_L is absorbed from the cool reservoir by the magnetic material. Finally, along the path 4-1, the material is isolated, and the field strength is increased causing the temperature of the magnetic material to rise.

Paths 2-3 and 4-1 are ones of constant entropy, yet are traversed through a changing magnetic field that tends to change different components entropy as previously described. As the applied magnetic field decreases (thereby increasing disorder or entropy) as it does along path 2-3, the temperature of the material falls thereby restoring the entropy balance. A similar description holds for the adiabat 4-1 where an increasing magnetic field causes the temperature of the material to rise.

These relationships can be illustrated further by considering the magnetocaloric material as a system composed of magnetic dipoles (one subsystem) sitting on lattice sites (a second subsystem). When a magnetic field is applied to this entire system, magnetic dipoles are aligned, and the entropy of the magnetic subsystem is decreased. In an adiabatic process, the decreased entropy of the magnetic subsystem is balanced by increased entropy in the lattice

subsystem, which raises the temperature of the magnetocaloric material, and keeps the entropy of the entire system (dipoles plus lattice) constant. The exchange of entropy between the two subsystems allows the entropy of the entire system to remain constant. This exchange reverses when the applied magnetic field is decreased.

The cycle shown in Fig. 15 is bounded by two isotherms and two adiabats and is clearly the Carnot cycle. Therefore, in theory, the ideal magnetic cycle possesses Carnot efficiency. This and other cycles can be accomplished by moving either the magnetic field or the magnetic material (Hull 1989), or by changing the magnetic field strength. Methods of altering the field include:

- physically moving the magnet that produces the field,
- moving a shield between the magnet and the magnetic material, or
- switching the field on and off.

Methods of moving the magnetic material include:

- mechanical means such as reciprocating it through an applied external magnetic field, and
- continuously rotating a disk containing magnetic material so that a portion of the disk is in the applied magnetic field at all times.

While some laboratory studies have been performed using most of these techniques, evaluation of design alternatives is still in the embryonic stage.

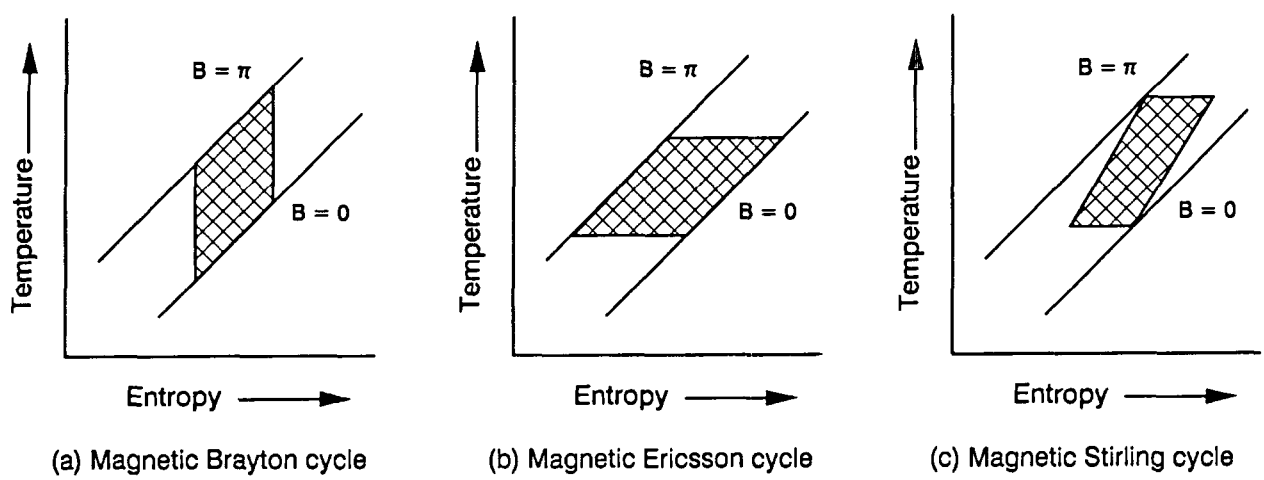

Fig. 16. Potential magnetic heat pump cycles.

Magnetic heat pumps may be based on several cycles, each of which has a unique representation on a temperature-entropy diagram. The basic cycles shown in Fig. 16 are:

(a) the magnetic Brayton cycle formed by two adiabatic and two constant magnetic field lines;

(b) the Ericsson cycle formed by two isotherms and two constant applied magnetic field lines;

(c) the Stirling cycle defined by two isotherms and two lines of constant magnetization.

State Of Technology

Theoretically, the ideal Stirling cycle has Carnot efficiency. However, the real cycle has irreversiblities which lower the efficiency. The four main sources of irreversibility are:

- heat transfer temperature difference between the working material and the regenerator fluid,
- specific heat variation between magnetized and demagnetized working material,
- pump work needed to move the regenerator fluid through the porous working material, and
- variation in specific heat of the working material as a function of temperature.

In the case of a rotary magnetic heat pump, these loss factors have been incorporated into simulation models and demonstrations to estimate efficiency and performance.

A recent model predicted performance expected from a 1050 kW (300 ton) air conditioning magnetic heat pump at rating conditions for a centrifugal chiller (Waynert 1994). The model considered losses associated with regenerator heat transfer, axial dispersion, and conduction. It also accounted for non-ideal magnetic material properties, pumping work and inefficiencies, and power required to cool the superconducting magnets. Results indicated a final coefficient of performance (COP) of 8.14 (0.43 kW per ton). This performance is significantly better than that of centrifugal chillers (see page 102).

Researchers at Los Alamos National Laboratory built and tested a rotary magnetic heat pump designed to execute a magnetic Brayton cycle to evaluate performance in a working equipment system,. The rotor segments were filled with gadolinium chips, and the segments were rotated into and out of a high strength magnetic field. The performance of this working system was reportedly much poorer than anticipated (Kirol 1984).

Development Issues

One of the major hurdles for magnetic heat pumps is lack of a high temperature superconductor. If such a material were discovered and produced at reasonable cost, the outlook for the magnetic heat pump would be improved. An Electric Power Research Institute (EPRI) assessment of the outlook for magnetic refrigeration (Lewis 1992) indicated that cost-effective magnetic heat pumps would materialize only after breakthroughs in high temperature superconductivity are made.

While the EPRI report holds that such breakthroughs will occur only in the long term (15^+ years), other recent studies indicate that the timeframe may be much shorter. Several of these studies have found that high temperature superconducting magnets could be used in combination with existing commercially available cryocoolers, rather than requiring liquid helium for cooling as do conventional superconducting magnets. The continuing development of such practical high temperature superconducting magnets is a key factor in overcoming this major technical hurdle for magnetic heat pumps.

If the technical and economic barriers could be reduced to acceptable levels, there are safety issues that must be resolved to extend the magnetic heat pump beyond niche applications. As indicated, the magnetic field strengths needed to cause the significant entropy changes necessary for space heating/cooling or refrigeration applications can be quite large (e.g. 7 T) and can lead to high field strengths near the heat pump. While the potential danger from magnetic fields is a significant safety issue, it may not be a debilitating one in the case of magnetic heat pumps.

Magnetic fields from magnetic heat pumps are static and appear to decrease rapidly as a function of distance from the equipment. Research on the impact of magnetic fields on living organisms has found few deleterious effects from static fields, as opposed to the more significant negative impacts associated with changing or radiating magnetic fields. Further, calculations indicate that this static field will be only about 10 Gauss at a distance of 3 m (9.8 ft) from a 1050 kW (300 ton) heat pump—a stray field level similar to that produced by hospital magnetic resonance imaging equipment.

Despite the paragraph above, operational precautions and/or protective hardware may be a necessary. For example, operating practices may require that personnel not be near a magnetic heat pump except for infrequent times needed for maintenance and other short operations. If special magnetic shielding is found to be a necessity, it will lead to higher first costs of a working system. Further research will be needed to determine the level of protection to assure safety for an operating system.

Potential To Replace CFCs and HCFCs

Market studies indicate that the most favorable initial market for magnetic heat pump applications is in the area of gas liquefaction, with a market niche for small (900 kg/d, 2000 lbs/day) hydrogen liquefiers (Uherka 1989). Much of the work on magnetic heat pumps in the United States has been conducted through the DOE and has been directed towards study and development of a system to reduce industrial energy consumption.

However, recent interest and advances in high temperature superconductivity and the desire for heating and cooling systems that do not require chlorofluorocarbon has expanded interest by DOE and private industry in magnetic heat pumps to include near room temperature applications. Current development efforts for large magnetic heat pumps are focusing on:

- reducing the cost of magnetocaloric materials to lower the capital costs of a commercially available magnetic heat pump;
- performing detailed effectiveness and efficiency measurements on the active magnetic regenerator to compare with modeled results and to validate efficiency predictions; and
- preparing full-scale demonstrations of magnetic heat pumps (Waynert 1994).

Earlier analyses (Waynert 1989) have indicated that although magnetic heat pumps could be made to compete on a performance basis, they were too expensive to replace existing vapor compression devices. Although to date, no device that would qualify as a commercial prototype has been constructed, some industry sources hold that a full scale prototype magnetic heat pump air conditioner could be designed, built and tested within three years. If followed by effective field testing and accelerated life-cycle reliability testing of a number of units, these sources feel that practical magnetic heat pump air conditioners could be ready for commercial distribution within five to six years.

MALONE REFRIGERATION

Introduction

Malone refrigeration uses a liquid near its critical point as a working fluid operating through a Brayton or Stirling cycle. The resulting machine, which was first studied by John Malone, can be driven externally to produce a refrigerating effect, or can be used in a reverse cycle as a heat-driven engine. Malone's first complete system was an engine which burned coal, used high pressure liquid water as the working medium, and produced 37 kW (50 shaft horsepower) shaft power in 1925. Malone claimed that with perseverance, this engine would have eventually produced 373 kW (500 horsepower).

Initial experiments conducted by Malone on the 37 kW (50 hp) engine were followed by the design and operation of a smaller, more versatile engine which also developed 37 kW (50 hp). Although this engine was a success, Malone failed to find the capital needed to complete development and to commercialize the liquid engine. Due perhaps in part to the Great Depression, liquid engine development came to an abrupt halt. In 1977, Malone's work was rediscovered (Swift 1989), and a liquid engine based on the Stirling cycle was designed for systematic study of liquids in heat engines.

Technology Description

In a refrigeration system, the Malone cycle would use the cooling accompanying the expansion of a liquid, but without a phase change. It has been suggested (Swift 1990) that acceptance of the liquid engine concept was delayed by a lack of understanding concerning the compressibility of liquids and how the compressibility changes in the neighborhood of the critical point. The approach taken in conventional engineering practice embraces the notion that liquids are "incompressible", and that liquids are most useful in cases where heat addition or removal can be accomplished through a change in the temperature of the fluid (i.e. using the sensible heat content of the fluid). However, Malone and others (Allen 1980) have shown that liquids can be used in thermodynamic cycles with some advantages over gases.

The Malone cycle using a single phase liquid is quite similar to the Stirling cycle. The main difference lies in the configuration of the displacer and the way in which the working fluid transfers heat to and from the displacer as shown in Fig. 17.

The Malone mechanism uses check valves and a relatively small displacer piston. As the displacer piston oscillates, the check valves maintain a unidirectional, pulsating flow of fluid with counterflowing streams on either side of the displacer rod. With the Malone cycle, both the displacer rod and the "pile" or regenerator assembly are actuated either directly or indirectly by a crank. In the case of the Stirling system, only the displacer and the power piston move. Consequently, the original Malone system is somewhat more complex than the Stirling cycle from a mechanical standpoint and a good deal more complex to analyze. For this reason, recent investigations using liquids as thermodynamic working fluids used the Stirling cycle with a single power piston and displacer piston.

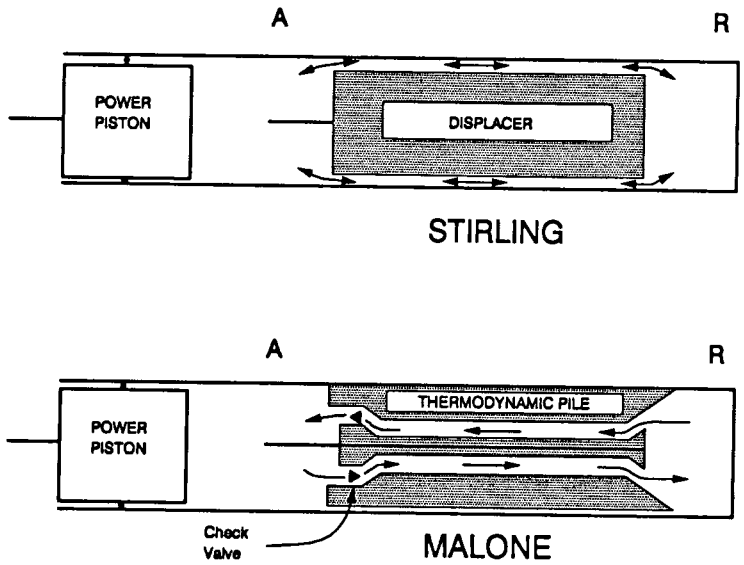

Fig. 17. Stirling and Malone machines.

One of the most important properties that a single phase fluid can possess in this cycle is a large thermal expansion coefficient, β. This is due to the fact that the heat absorbed by a fluid during an isothermal expansion at temperature T, is directly proportional to β. In the case of air and many other gases that are typically considered for Stirling or Brayton cycle applications, β = 0.36%/°C (0.2%/°F) whereas for liquid water, β is less than 1/10 this value (i.e water is virtually incompressible). However, at higher temperatures, the thermal expansion of water and other liquids rises: it is about 1/2 that of a gas at 177°C (350°F), equal to that of a gas at 260°C (500°F), and double that of a typical gas at 316°C (600°F). Although these high temperatures present a thermodynamic advantage, high pressures must be maintained to prevent the liquid from flashing. The operating pressure in one of Malone's early engines operating on water was 69 MPa (10,000 psia).

Another important thermophysical property of single phase working fluids in heat engines is compressibility, κ, which is a measure of how the volume of a fluid changes with pressure. Fluids with low compressibility can exhibit large changes in pressure with little change in volume. Fluids which undergo large pressure changes with small volume changes (i.e. fluids with low compressibility) would be at an advantage since:

- the power of a heat engine is proportional to the pressure change of the working fluid, and

- the fluid volume change determines the size of the piston/cylinder configuration.

For ideal gases, $P\kappa = 1$, whereas for many liquids, $P\kappa < 0.1$ where P is the operating pressure near the critical point. Consequently, in the neighborhood of the critical point, liquids have advantages over gases when used as working fluids in heat engines. Another attribute of liquids is a high volumetric heat capacity as compared to gases. The volumetric heat capacity of liquids over a small temperature range is comparable to the latent heat capacity of many conventional refrigerants. This difference affects the size and configuration of any heat exchangers used in the cycle. Less volumetric displacement is required to transfer a quantity of heat through a heat exchanger from a liquid than for a gas. This suggests that there might be less mechanical power required to pump a liquid through a heat exchanger than would be required for a gas. Experiments are being conducted to test and evaluate these considerations and to increase understanding of the operations characteristics of both liquid engines and refrigeration based on liquids used in a thermodynamic cycle.

State Of Technology

Indications are that most of the current research on the Malone cycle is being conducted by groups at Los Alamos National Laboratory and the University of California at San Diego. An experimental engine based on propylene as a working fluid was developed, constructed and is being tested at Los Alamos. This engine is sufficiently versatile to allow determination of individual loss mechanisms such as viscous flow, bearing and seal friction and inherent cycle loss. Generally, measurements of each loss mechanism were found to agree with calculations based on the physics of the system and the thermophysical properties of propylene. Propylene rather than water was chosen because its critical temperature is relatively close to room temperature, and its critical pressure (46 bar, 667 psia) is manageable with small displacement piston/cylinder configurations.

Characteristic Advantages

The behavior of many liquids near their critical points are similar; consequently, the technical choice of liquids for the Malone cycle appears to be driven primarily by critical temperature. The non-toxic, nonflammable and environmentally acceptable nature of CO_2 would make it a good candidate for the Malone cycle. However, the critical temperature of CO_2 (31 °C, 88 °F) is too low for efficient operation in many HVAC/R applications. Other fluids which may be considered are methyl alcohol, ethyl alcohol, acetone or sulfur dioxide which have higher critical temperatures but present safety issues. Other fluids which have been considered include sulfur hexafluoride and various fluorocarbons such as HFC-134a. In general, these compounds have critical temperatures more suited to HVAC applications and critical pressures lower than CO_2.

Some results comparing the efficiency of the liquid propylene cycle with a Rankine cycle based on CFC-12 have been reported (Swift 1990). The results of this analyses are shown in Table 4. In the "Realistic cycle" Rankine system, the effects of pressure drops in tubing and other irreversibilities have been taken into account, and in the "Practical hardware" entry, additional losses such as temperature differences between the working fluid and air as well as motor inefficiencies and power for fans are considered. In the case of the Propylene Stirling engine, the "idealized cycle" COP/COP_c was calculated to be 0.7 based on a $\Delta T = 45°C$ (81°F). The "Present hardware" entry results from performance measurements at the most efficienct operating point. This performance measurement includes blow-by and seal drag and assumes that the efficiency of the motor is 100%. It was determined that seal drag and blow-by losses constitute more than half of the test engines total torque, and it is estimated that the overall efficiency of the Malone engine can be improved with attention focused in these areas. Interestingly, these major sources of inefficiency become smaller as the refrigerator is scaled up to more practical size. However, as Table 4 indicates, the theoretical COP of the liquid Stirling machine is no greater than that of vapor compression.

Table 4. Comparisons of liquid-Stirling engine and conventional engine.

CFC-12 Rankine engine	COP/COP_c
Idealized cycle	0.8
Realistic cycle	0.4
Practical hardware implementation	0.2
Propylene Stirling engine	
Idealized cycle	0.7
Present hardware (ΔT=30°C; without motor or auxiliaries)	0.2

Development Issues

There are several issues associated with the Malone cycle, and many of these issues are similar to the issues connected with the Stirling engine. In the case of the Malone (liquid) systems, the high specific heat of liquids means that only a very small adiabatic temperature change occurs during the expansion portion of the cycle. This means that where liquids are used in a simple Carnot cycle, only small temperature intervals can be spanned. This means that a regenerative or recuperative components must be used, and these components lead to irreversibilities associated with heat transfer. These processes and associated losses occur with the conventional Stirling cycle which operates with single phase gases.

Losses associated with seals and piston blow-by are of particular concern in liquid engine systems. Although a small displacement piston/cylinder system can be used, the pressures (approaching the critical pressure) required for most known fluids can be quite high. In the case of the original Malone engine operating with water, pressures in the cylinder reached about 69 MPa (10,000 psia). While other fluids are being considered, the fact remains that

pressures in the cylinder are likely to remain high, and issues of piston blow-by and adequate sealing will be of continuing concern.

A third issue, one that is not restricted to the Malone concept is the availability of a suitable working fluid. The advantages of using liquids in thermodynamic cycles become greater at operating temperatures and pressures near the critical point. Consequently, fluids with critical temperatures near the application temperature are a requirement. This places a limitation on the liquids that may be used under a given set of source/sink conditions.

Potential To Replace CFCs and HCFCs

It must be understood that the Malone engine (or refrigeration system) operating in a Stirling cycle has been studied only in the laboratory to provide an understanding of the factors that affect performance. It is difficult to predict what the performance of a well-engineered liquid refrigeration system might be with research aimed at improving its efficiency. Only a theoretical analysis coupled with limited laboratory research on a working Malone system has been conducted which was aimed at understanding and quantifying the loss mechanisms associated with the Malone cycle rather than any serious attempt at mitigating these losses to improve the efficiency of the system. Analyses have shown that the theoretical efficiency of the Malone cycle is almost as high as for the Rankine cycle; however, a good deal of applied research will be needed to reach even this potential. It is very unlikely that this technology can be commercialized in the foreseeable future.

THERMOELASTIC HEAT PUMP

Introduction

The observation that the internal temperature of an elastomer (e.g. rubber) rises when it is stretched provides the basis for a heat pump. The mechanical work required to stretch the elastomer is converted into heat energy which can provide useful heating as the elastomer cools. Following this extension and heat removal, the elastomer is relaxed and cools in the process. By alternately stretching and relaxing the elastomer, a heat pumping action is created.

Technology Description

An elastomer consists of a network of long polymer chains which are crosslinked with chemical bonds at a limited number of locations along each chain. When the elastomer is stretched at constant temperature, the molecular chains are straightened out which leads to lower system entropy. Heating the elastomer increases its entropy, and, if the elastomer's length is maintained, its tension increases. The relation between the tension, length and heat applied to an elastomer is shown in Fig. 18.

If the elastomer begins at length L1 and temperature T1 and is then stretched adiabatically to L2, its temperature will rise to T2. Heat Q2 is rejected as the elastomer cools to temperature T3. The input work during this process is directly proportional to the area under the force-length curve, or

$$Work = \int_{L1}^{L2} F(l)dl.$$

where F is the tension in the elastomer at any length, l. The elastomer now at length L2 and temperature T3, is relaxed to length L1. During this adiabatic process, the elastomer cools to temperature T4 and produces work. Heat Q1

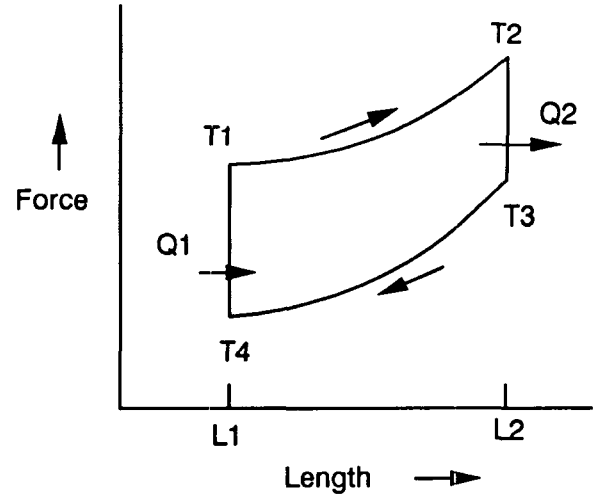

Fig. 18. Thermodynamic cycle with ideal elastomer.

is then added to the material returning its temperature once more to T1. Through this action, a heat pump cycle is created which uses mechanical energy to move heat Q1 at a lower temperature and rejecting heat Q2 at a higher temperature. By reversing the cycle direction, a heat engine which can produce work as heat flows from hot to cold is produced.

State Of Technology

The elastomeric heat pump is the creation of Elastek, Inc., and several patents covering the practical application and additional development of this concept have been issued. The working device is simply layers of an elastomer which are separated slightly to allow air to pass through. As the elastomer assembly is stretched, the air is either heated or cooled. A major Elastek contribution to the concept was the application of regenerator technology to increase the temperature lift. The regenerator is relatively simple, consisting of a second elastomer bed module arranged so that a mechanical input stretches one module while simultaneously relaxing the other. Experiments with this device have shown that the cold region reaches a low temperature several times the small (3.9°C, 7°F) adiabatic temperature change of a single typical elastomer.

Most of the performance information on the elastomeric heat pump was derived from Elastek's computer model of the system. Performance simulations conducted by Elastek predict a COP of about 6 for the device alone operating between 21° and 38°C (70 and 100°F), and a realistic COP greater than 3.5 when an electromechanical drive system is installed.

Characteristic Advantages

The system can be expected to have good heat transfer since the working elastomer is also part of the heat exchange device. Elastek estimates that 5 watts of cooling per gram of elastomer (7700 Btu/h per pound) can be produced, so that a relatively lightweight working material could be used. One must note that the support structure and mechanical power source will add to this number.

Development Issues

Perhaps the most critical issue is the lifetime of the elastomer over many cycles. Although some elastomers have been successfully formulated for biomedical applications (e.g. artificial hearts) which require extensive lifetimes, it is not clear that systems which derive useful work and heat from the stretching and relaxing of the elastomer could be expected to last. Other

issues focus around the connections of the elastomer to the support structure. Here, stress risers are certainly bound to be present which could be a problem. Early DOE work on an elastomeric energy storage device was hampered with problems associated with the connections at the ends of the elastomer. Still other issues deal with developing the balance-of-system under the tight cost and performance targets established by existing technologies. At this point, there is an insufficient body of data available to make further projections on future prospects for this interesting system.

Potential To Replace CFCs and HCFCs

Thermoelastic heat pumps represent at best a very long term possible alternative to replace vapor compression technology.

EVACUATED PANEL INSULATION

Introduction

Heat is transmitted by conduction, convection, and radiation. Plastic foam insulation transmits heat by conduction through the solid "struts" separating the walls and the gas in the cells and by radiation of heat directly through the foam. The development of foams with significantly smaller cell sizes are effective at reducing conduction through the solid portion of the foam; impermeable facers are used on foam boards to keep air from permeating into the cells, raising the gas thermal conductivity and increasing heat flow; and opacifiers such as carbon black are used to reduce radiative heat transfer. None of these developments, however, are directed toward reducing the conduction component of heat flow through the cells in the foam.

Technology Description

Evacuated panels are being developed for appliance insulation that can achieve extremely low thermal conductivities by virtually eliminating heat conduction through the gas in the insulation and maintain low conduction rates through the solid portion; attention may also be given to reducing or eliminating the radiative heat transfer. Typically, some sort of filler material is compressed into flat panels and encased in one or more gas impermeable membranes under a vacuum. Many different filler materials have been considered by developers including various powders, aerogels, glass fibers, and open cell polyurethane and polyisocyanurate foams. The barrier materials are selected depending on the degree of vacuum needed and include plastic polymers, glass, and metal. Center panel thermal conductivities range from 5 to 8 mW/mK (0.0347 to 0.555 Btu·in./h·ft^2·°F), though 2 mW/mK (0.0139 Btu·in./h·ft^2·°F) can be achieved; the size of the panels and thermal conductivity of the barrier material become important because the overall thermal conductivites are strongly influenced by the "edge effects."

Filler Materials

Evacuated powder panels: powders considered for use as fillers include precipitated silica, fumed silica, perlite, diatomaceous earth, and fly-ash. The powder is compacted and sealed in a barrier that can maintain an internal pressure of 0.1 to 10 Torr (13 to 1330 Pa). Extremely fine powders are used in order to reduce both conduction and convection of heat

through the panel; the powder size can be a health consideration in the manufacturing and also the disposal of panels. The weight of the insulating panels is an important consideration; conventional foam insulation weighs around 32 kg/m^3 (2 lbs/ft^3) and the panels weigh 150 to 450 kg/m^3 (10 to 28 lbs/ft^3). The panels can add significantly to the weight of a refrigerator, for example, and the cabinet design must be able to support that weight. Center of panel k-factors of 5 to 6 mW/mK (0.035 to 0.042 Btu-in./h-ft^2-°F) are possible with powder fillers.

Aerogel panels: aerogels are light weight solid materials with high porosity (> 90%), extremely small pores (2 to 100 nm), and extremely small ligaments (4 to 8 nm). This combination of properties gives them an extremely high surface areas, 600 to 1000 m^2/g (Ramamurthi). Conventional aerogels are known to be fragile and sensitive to moisture with involved and time consuming manufacturing processes. The thermal conductivity of "non-vacuumized" aerogel filled panels has been measured at 12 mW/mK (Nipkow 1993). Aerogel filled panels tolerate higher internal pressures than some other filler materials and panels have been manufactured that attain 7.5 mW/mK at 10 kPa (Hermann 1993). This higher pressure permits greater latitude in the selection of barrier materials for the enclosing membranes. Opacifiers may be used to reduce radiative heat transfer and further reduce the thermal conductivity. Aerogel panels have limited strength and will require additional support.

Evacuated fiberglass panels: evacuated fiberglass panels are similar in construction to powder and aerogel filled panels, though the thermal properties cover a broad range depending on the diameter of the glass fibers, degree of compacting, and barrier material. K-factors range from 2 to 14 mW/mK with internal pressures of 0.13 to 133 Pa (0.001 to 1.0 Torr) (DOE 1993). The thermal conductivity decreases as the degree of vacuum increases, although it remains relatively constant below 0.1 Torr (McGrath 1993). Fiberglass panels are also heavy, 300 kg/m^3 (19 lb/ft^3), and this will affect cabinet design and transportation requirements. Also like powder filled panels, there may be health considerations resulting from the very small glass fibers in the panels.

Open cell PUR/PIR foam filled panels: normally, closed cell polyurethane foams are used in appliance insulation and the low thermal conductivity of the gas contained in the cells contributes to the insulating ability of the foam. Through the use of open cell foams in vacuum panels, however, the blowing agent is removed altogether and heat transfer occurs only by conduction through the solid struts of the foam and by radiation. Efforts have been directed toward producing foams with a high proportion of open cells, so the gas can be removed, and with small cell structure to reduce conduction through the solid. Open cell foams are lightweight, inexpensive filler materials, but lower internal pressures are required to achieve the same insulating effect as panels using different fillers. This constraint on the pressure level requires a barrier envelope with lower permeability than can be used with something like aerogels. Panels using open cell foams also do not provide any support where they are used and some additional structure, such as blown foam, is needed.

State of the Technology

Sufficient prototype production and testing has been performed to establish the feasibility of evacuated panel insulation. Several variations on the technology have demonstrated different combinations of filler and barrier materials and internal pressures to meet thermal resistivity goals. Relatively small panels have been manufactured and it needs to be shown that size can be scaled up to what would be required for target applications, such as refrigerator/freezers. Small scale production plants exist and center of panel measurements indicate significantly lower thermal conductivities than HCFC, HFC, or hydrocarbon blown foam insulations.

Characteristic Advantages

Evacuated panels are characterized by their very low thermal conductivities and high value as insulation. Care must be exercised in evaluating the potential benefits of evacuated panel insulations, however, because most published data cite center-of-panel measurements for thermal conductivity (or resistivity). The barrier materials used to contain the filler material under vacuum have much higher thermal conductivities than the fillers themselves, and the heat flows across the surfaces and around the edges of the panels can greatly diminish the full-panel thermal conductivity from the center-of-panel measurements.

Development Issues

Product lifetime continues to be a concern; using a refrigerator/freezer as an example, the thermal performance of the cabinet will degrade substantially if the panel loses the required vacuum over the 15 to 20 year lifetime of the appliance. Market factors including the insulating value, cost, manufacturing and handling procedures, industrial safety, and proven lifetime will determine the combination of filler and barrier materials best suited for particular applications.

Potential To Replace CFCs and HCFCs

Evacuated panel insulation is one possible alternative to improving the thermal envelope of appliances, refrigerated shipping containers, and buildings. The extent to which it is used, or even whether it is used, will be dependent on not only on technical issues, but also on the relative costs of other energy saving alternatives (e.g. refrigerator door gaskets, compressor efficiency improvements, thicker layers of conventional insulation). If evacuated panels are

used in appliances, they will probably be used in conjunction with plastic foam insulation to seal around the panels and to provide structural support to the cabinet. These foams could be formed using HCFCs, HFCs, hydrocarbons, or CO_2. Assuming that the foam flow characteristics are acceptable so that voids do not occur in the insulation, the use of hydrocarbons or CO_2 could eliminate the use of fluorocarbons in this application, although fluorocarbon blown foams could be used and achieve lower overall energy use.

PART 2: ALTERNATIVE COMPRESSION TECHNOLOGIES

ADVANCED VAPOR COMPRESSION

Introduction

Vapor compression using fluorocarbons, hydrocarbons, or ammonia is the established technology for household, commercial, and industrial refrigeration and air conditioning. Other technologies such as absorption chillers and Stirling cycle cryocoolers share the market, but vapor compression is by far the dominant technology. As such, vapor compression refrigeration and air-conditioning equipment have benefitted from extensive research and development and are extremely efficient and reliable.

Technology Description

Vapor compression refrigeration systems all employ a compressor, a condenser, an expansion device, an evaporator, and a working fluid or refrigerant. Primarily advanced systems will be adaptations of current technology to use ozone safe refrigerants in efficient heating and cooling systems without adversely impacting the cost and safety of the end user.

State Of Technology

Efficiencies of vapor compression equipment have increased steadily in the past, driven by a competitive market, energy prices, and government policies. The historical trend of efficiencies for centrifugal chillers is shown in Fig. 19 (Trane 1994). There is an abrupt step up in efficiency between 1980 and 1981 when products were brought to market incorporating R&D efforts initiated in response to the energy crisis of the 1970s; efficiency started increasing at a faster rate in the early 1990s in response to competition between CFC alternatives. The curves in Fig. 20 for EER (before 1981) and SEER (after 1981) show the steady increase in efficiency for central air conditioners sold in the U.S. (ARI 1994). Similar curves could be developed for household refrigerators and supermarket refrigeration systems. In many instances, price has decreased (in real terms) and performance has improved as efficiency has gone up.

Many of the current R&D activities for vapor compression equipment are directed toward the design of products to use alternatives to HCFCs. These include modifications necessary for unitary equipment to use high pressure mixtures and zeotropic mixtures of HFCs. Efforts are also underway to identify alternatives to HCFC-123 for centrifugal chillers. In the late

1980s the early tests of alternatives to CFC-11 and CFC-12 in chillers showed losses in efficiency relative to the CFCs. The HVAC industry responded with aggressive development efforts and chillers on the market in 1994 using HFC-134a and HCFC-123 are more efficient than the chillers using CFCs that they replaced. The same pattern exists for household refrigerators using CFC-12 and CFC-11 foam insulation. In 1994, early testing indicates there could be some losses in efficiency using replacements for HCFC-22 in unitary equipment. Industry is confident that just as with chillers and refrigerators, by the time they bring new unitary equipment to the market it will be as efficient, or more efficient, than the products using HCFC-22 that are being replaced.

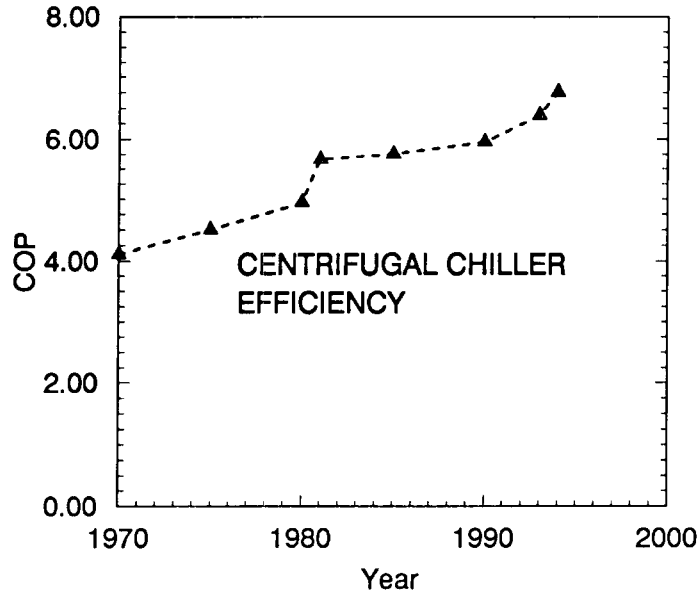

Fig. 19. Efficiency of centrifugal chiller at industry rating conditions.

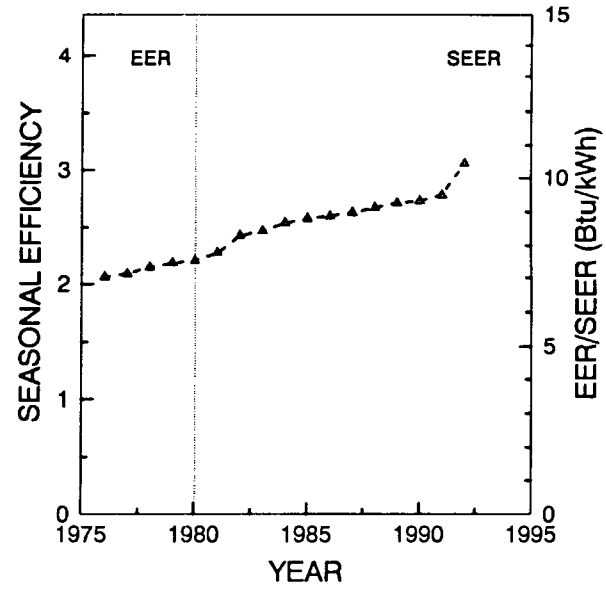

Fig. 20. Efficiency ratings of air conditioners sold in the U.S.

Characteristic Advantages

A major advantage of vapor compression technology is that advancements are building on proven performance and high efficiencies of well established products. Fewer questions exist about efficiency, reliability, costs, and market acceptance than there are for alternative not-in-kind technologies.

Development Issues

Foreseeable developments include the use of higher efficiency motors and compressors, heat exchanger modifications to take advantage of temperature glides of zeotropic refrigerant mixtures, and dual evaporators in two-temperature appliances like refrigerator/freezers. Regulations and costs are also reducing emissions by prohibiting venting of refrigerant charges during servicing and improving maintenance practices and procedures for eliminating leaks. Future developments are possible in the design of fittings used in large systems and in reducing refrigerant charge levels that will impact the amounts of refrigerant escaping to

the atmosphere (this includes consideration of hermetic, stand-alone display cases, multiple condensing units in close proximity to the display cases, and secondary refrigerant circuits for supermarket refrigeration).

Potential To Replace Conventional Technology

There is a very strong likelihood of success in developing vapor compression systems using HFCs, hydrocarbons, or ammonia that are comparable to or better than the products using CFCs or HCFCs that they are replacing. The successful transition away from CFCs has already been demonstrated in some products. While it is acknowledged that compression systems will continue to improve, it is not known how quickly or how great these improvements will be.

For the most part, the comparisons in this study are based on demonstrated efficiencies for both next generation compression equipment and the emerging not-in-kind technologies. Exceptions to the use of proven performance include the selection of refrigerant loss rates for automobile air conditioning and unitary air conditioners. Documented loss rates used in past studies are from when it was standard practice to simply vent the refrigerant charge during servicing. Such practices are now prohibited in the United States and elsewhere and it is clear that historical emission rates are not appropriate for current calculations. Projected future rates based on information from the industry are used in these cases.

HYDRAULIC REFRIGERATION

Introduction

The concept of hydraulic refrigeration is a straightforward application of using gravity to pressurize and condense a refrigerant which can subsequently be evaporated to deliver a cooling effect. This system relies upon the weight of a column of water to provide the compressive force and it uses relatively low technology pumps to move the fluids instead of finely tooled compressors machined to tight tolerances.

Technology Description

As shown in Fig. 21, there are two loops that operate simultaneously: one is a refrigerant loop and the other is a vertical water loop. The two loops share a common downpipe. Refrigerant vapor from the evaporator enters the downpipe (point 0) and is entrained by a falling stream of water. As the two-phase water and refrigerant vapor mixture descends, the hydrostatic pressure increases and compresses the bubbles of refrigerant. If the vertical pipe is long enough and smooth, the hydrostatic pressure at the bottom of the pipe (point 5) will exceed the saturation vapor pressure of the refrigerant, and the refrigerant will liquify. Liquid refrigerant and water, already immiscible, separate due to a difference in liquid densities. If the density of the condensed refrigerant exceeds that of water, the refrigerant descends to the bottom of the separation chamber as shown. The

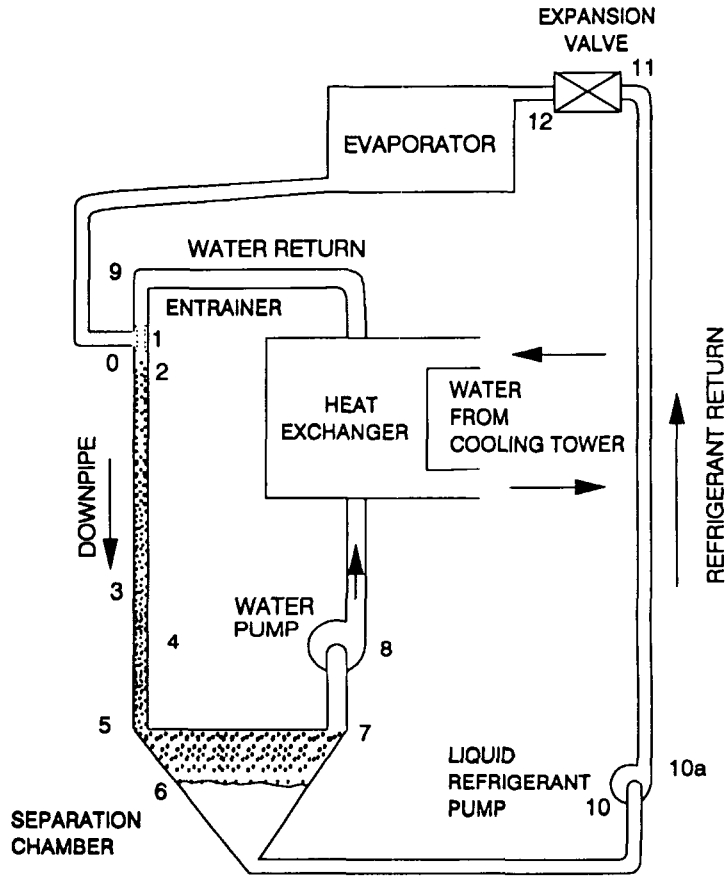

Fig. 21. Basic Hydraulic Refrigeration System (HRS).

refrigerant is pumped to the expansion valve (point 11) which drops its temperature and pressure. As in the case of a conventional vapor compression refrigeration cycle, the refrigerant enters the system evaporator where heat is absorbed, and the refrigerant reenters the water eductor system to complete the cycle. The water pumped through the downpipe entrains and compresses the refrigerant, and removes the heat of condensation from the refrigerant. The heat of condensation is transported by the water to a heat exchanger coupled to a cooling tower.

State Of Technology

Although systems utilizing hydraulic compression were developed early in the 20th century, models to provide an analytical understanding of the technology were not available until recently. Work by Hosterman and Rice resulted in four patents on the Hydraulic Refrigeration System (HRS) and predicted its operating characteristics using CFC-114, CFC-12, HCFC-22 and HC-600 (n-butane) (Hosterman 1979, Hosterman 1981, Rice 1984, and Rice 1991).

A significant research effort on the HRS concept is underway at Arizona State University where three units have been constructed. An early "proof of concept" version constructed in 1979 used CFC-114 as the refrigerant and operated at an evaporator temperature of 7.2°C (45°F) to meet ARI standard operating conditions. With a single refrigerant/water vertical entrainment tube that was 26 m (85 ft) long, the system worked reasonably well and provided about 7 kW (24,000 Btu/h) of refrigeration. A second system based on CFC-114 was designed, built and used to test the feasibility of a multi-stage arrangement to reduce the overall HRS height requirement, and to test better methods for separating water from the refrigerant at the exit of the vertical tube.

Experiments are currently underway on a three-stage system with higher capacities. The working fluid is HC-600 rather than the fluorocarbons used in earlier studies. The system design was based on a thermal/hydraulic model which predicted the capacity to be 31 kW (106,000 Btu/h) at an evaporator temperature of 7.2°C (45°F), and a projected COP of 2.93 (EER = 10 Btu/Wh). Measurements have shown that at this evaporator temperature, 21 kW (72,000 Btu/h) of refrigeration could be produced at a COP of 2.64 (EER = 9) with a condenser water temperature of 25°C (77°F). The performance of the HRS over a range of condensing temperatures and refrigeration capacities is shown in Fig. 22.

The principal performance limitations were found to be vapor compression loop water carry-over and pump inefficiencies. As the capacity (size) of the system increases, progressively higher water flow rates through the evaporator tube are needed to entrain the quantity of refrigerant required. Effective separation of the water from the refrigerant under these high flow rates has been a problem, and work is planned to study and develop more efficient separation technologies and to reduce pumping power requirements.

Although the n-butane HRS is still considered to be in the field test and evaluation stage, it is interesting to compare its measured performance with a conventional chiller. The HRS serves the same function as a chiller since both deliver cooling at the evaporator and both require a cooling tower or other heat rejection device. Standard ARI conditions for chiller rating are: (1) a leaving chilled water temperature of 6.7 °C (44 °F) and an entering condenser water temperature of 29 °C (85 °F). From the data shown in Fig. 22, the HRS delivers an approximate COP of 1.76 (EER=6) under these conditions. The HRS performance data cited was taken from only the third prototype unit made and the first of its kind which used HC-600 as the refrigerant. Better performance of the HRS is expected with continued research and development.

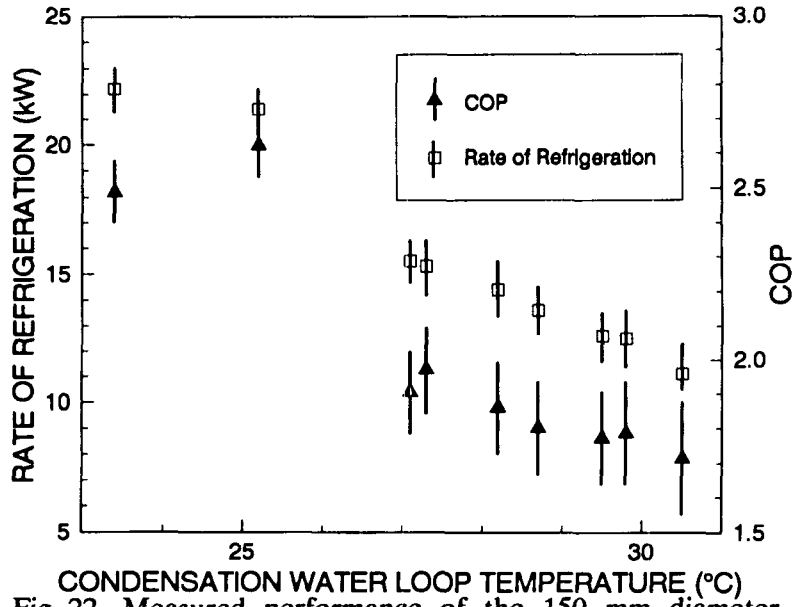

Fig. 22. Measured performance of the 150 mm diameter hydraulic refrigeration system.

CHARACTERISTICS AND ADVANTAGES

In principle, the HRS can operate with a variety of refrigerants including those with densities higher or lower than water. However, there are limitations that, in practice, reduce the number of refrigerants to a number smaller than would otherwise be expected. Refrigerants must:

- be immiscible with water so that the two are easily separated by gravity at the bottom of the tube and so there is little carryover of water into the refrigerant loop,
- be chemically stable in the presence of water (low hydrolysis rates),
- have suitable vapor pressure characteristics so that refrigerant condensation occurs over a small pressure range (to reduce the length of the vertical pipe) and under pressures that are low enough so that pipes with moderate pressure ratings can be used, and
- not form a gas hydrate (clathrate) under the temperature and pressure conditions present in the vertical pipe.

In addition, the refrigerant choice for the HRS must meet the same safety and environmental criteria as for other technologies. However, the fact that a large quantity of water is a necessary component of the system may provide a safety factor in applications using flammable refrigerants.

Characteristic Advantages

A potential advantage of the HRS is the fact that it does not require a compressor, only pumps and controllers. This means that an HRS design can be constructed on site by skilled workers using rather conventional components. Because of these features, it has been proposed as an attractive cooling technology for developing countries where space is available, and the cost of drilling a hole in the ground to accommodate the downpipe is not an especially limiting factor.

Development Issues

The HRS system must be air-free and hermetic for efficient operation. Any significant amount of non-condensible gas such as air interferes with condensation. Maintenance of hermetic conditions could be a problem with the rather extensive array of large piping and fittings.

It is necessary to use a large pipe to convey the large volume of gas and the much larger volume of water. A 150 mm (6 in.) diameter downpipe was successfully used for the 35 kW (120,000 Btu/h) system, and as higher capacity systems are proposed, the pipe diameter must be increased to minimize friction pressure drop and to maintain sufficient flow rates. The large pipe size, with increasing weight and cost makes single units greater than 70 to 100 kW (240,000 to 360,000 Btu/h) unlikely.

Another issue being addressed by the team at Arizona State University involves better separation of the refrigerant at the bottom of the downpipe from the water. The mutual solubility between "immiscible" liquids such as water and HC-600 makes moisture carryover into the expansion valve and evaporator inevitable. This carryover can reduce the evaporator performance and, under low evaporator conditions, can lead to freezing up of the expansion valve. Refrigerant carryover into the water loop also occurs but may be less of a problem.

The length of the downpipe is another issue for which there are a limited number of solutions. With HC-600, the vertical downpipe is about 30.5 m (100 ft) long, and unless there is a convenient location on site such as an elevator shaft in a building, a well will need to be drilled which could add significantly to the installation cost. Efforts are underway to reduce the height of the system by using a series of shorter pipes in a staged configuration. The

staged system has somewhat of a thermodynamic advantage since the water flow rate can be reduced in the latter stages and less pump power required. Nevertheless, each stage requires a separate pump which leads to increased cost and complexity.

Potential To Replace Conventional Compressor Technology

Overall, the HRS has been shown to work effectively and with a non-fluorocarbon refrigerant. The system's commercialization potential is limited by its size and weight. The system requires relatively simple components for construction and appears to be an attractive technology where installation and servicing skills are for conventional technologies not abundant. The safety aspects of using relatively large volumes of flammable or explosive refrigerants where there are limited technical skills would need to be addressed if a hydrocarbon is used as the working fluid. Further work is underway to improve the efficiency of the HRS through the use of better controls and more effective water/refrigerant separation techniques.

ACOUSTIC COMPRESSORS

Introduction

Acoustic compressors could be designed and used as drop in substitutes for the conventional positive displacement compressors used in refrigeration and air-conditioning equipment. As such they could be used with any refrigerant including fluorocarbons and hydrocarbons. Pressures are created by a resonant acoustic wave within the constant volume chamber of the acoustic compressor, and there are no moving parts exposed to refrigerant that require lubrication (Lucas 1993).

Technology Description

The drawing in Fig. 23 is a simplified schematic of an acoustic compressor. A linear motor is shown on the left consisting of a permanent magnet and coils to form an electromagnet. The refrigerant manifold with suction and discharge valves and plenums is on the right. A resonance tube, or resonator, separates the linear motor and refrigerant manifold. An alternating current is used to activate the coils attached to the metal flange at the end of the resonator, causing the entire resonator to oscillate mechanically on its cylindrical axis. The resonator is vibrated such that a standing sound wave is created in the refrigerant gas inside the tube. The tube geometry and vibrating frequency are chosen such that a pressure antinode is created at the valve location. The refrigerant gas in the resonance tube is drawn in through the suction valve and discharged as the compressor oscillates at its resonant frequency. An acoustic compressor typically operating at 300 Hz would have a resonance tube 27 cm (10.6

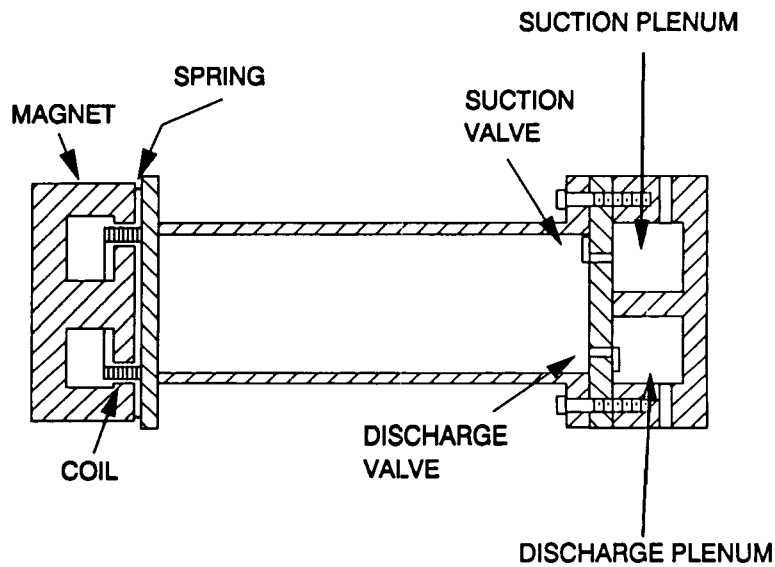

Fig. 23. Acoustic compressor.

in.) long if it used CFC-12 or HFC-134a (the length of the tube depends on the speed of sound in the refrigerant used).

State Of Technology

A prototype compressor has been built which has provided pressure lifts greater than 1400 kPa (200 psi) and mass flow rates of 2.7 to 5.4 kg/h (6 to 12 lb/h). Prototypes are being designed for compressors with capacities appropriate for home refrigerators (Lucas 1993).

Characteristic Advantages

The acoustic compressor has a number of potential advantages over conventional positive displacement compressors. All the issues associated with lubricants in refrigerants are irrelevent to acoustic compressors because there are no wear related moving parts in contact with the refrigerant that require lubrication. Consequently the compressor can operate essentially oil free. Compressors can also be designed for household refrigerators that require less space than conventional compressors so the inside shell of the appliance can be redesigned to increase the usable volume. The acoustic compressors should provide high efficiencies, COPs of 2.00 to 2.30 (EERs of 7 to 8 Btu/Wh) under the U.S. refrigerator compressor test conditions, and it is inherently simple to vary the refrigeration capacity of the acoustic compressor to follow load requirements (Lucas 1993).

Development Issues

Acoustic compressors are a very recent development and many technical issues must be solved before they can be used in consumer products. Areas that need attention include:

- energy losses due to non-linear acoustic effects in the resonator tube,
- high speed valves necessary because of the sonic frequencies required for small resonators,
- valve materials able to sustain up to 100 billion cycles during the lifetime of an appliance, that is 100 times the number of cycles for a valve on a positive displacement compressor
- noise abatement (mechanical oscillation at sonic frequencies generates a pure tone),
- pressure losses associated with high speed valves,
- 50 μm oscillation is sufficient to produce a pressure differential of 700 kPa (100 psi), but ΔP on the order of 1700 kPa (250 psi) is needed for most refrigerants being considered (fluorocarbons and hydrocarbons).
- improving the electrical to acoustic efficiency of the driver system, and

- electrical controls.

Potential To Replace Conventional Compressor Technology

Acoustic compressors have potential for application in domestic refrigeration, space conditioning, and automobile air conditioning and have attracted interest among OEMs. Acoustic compressors could result in improved vapor compression system efficiencies using HFCs, hydrocarbons, or other fluids. Efficiency levels that have been reported are low relative to conventional positive displacement compression efficiencies, but high efficiencies are possible with sufficient development. Commercial products could be available as early as 1998 (Lucas 1993), and there is a strong potential to displace conventional technology if efficiencies and reliability are equal or superior to positive displacement compressors (e.g. reciprocating, rotary, scroll), and if compressor noise can be reduced to comparable levels.

PART 3: APPLICATIONS

AUTOMOBILE AIR CONDITIONING

Introduction

Automobile air conditioning is an important application of fluorocarbons. Historically, CFC-12 has been used in belt-driven compression systems with flexible hoses connecting the compressor with the condenser in front of the radiator and the evaporator in the passenger compartment. Air conditioners are sized to provide quick "pull-down" from the inside temperatures reached while parked in full sun and also to provide adequate cooling during hot idle conditions experienced during stop and go traffic in high humidity climates. They must operate at extreme conditions, and although fuel economy is a factor in their design other considerations are of equal or greater significance. These include, among others:

- reliability,
- adequate capacity,
- size and weight of components,
- condenser frontal area, and
- service considerations.

Automobile air conditioners are not rated for efficiency and there are no established procedures for estimating fuel consumption short of exhaustive testing of alternative systems. It is clear, though, that power is required both to drive the compressor, fan, and blower and also for increased demands on the engine simply to transport the weight of the system.

Alternative Technologies

HFC Compression Systems

HFC-134a is the refrigerant of choice by the automobile manufacturers as an alternative to CFC-12 because it is non-flammable, non-toxic, and is similar to CFC-12 in performance. Most major manufacturers have either completed the change-over or are changing over to air conditioners using HFC-134a. Refrigerant charges of HFC-134a would be approximately 1100 grams in large vehicles typical of those used in North America and 730 grams for the smaller vehicles common in Japan and elsewhere.

Theoretical performance: the "ideal" (100% compressor isentropic efficiency) COP for HFC-134a is 3.54 at design conditions set for CFC-12 automobile air conditioners; 65.6°C (150°F) condensing, 4.4°C (40°F) evaporating, 11°C (20°F) subcooling, and 0°C (0°F)

superheat. These design temperatures, however, result in a condensing pressure using HFC-134a that is higher than that for CFC-12 air conditioners. Modifications to condensers to operate at approximately the same head pressures as CFC-12 result in a lower condensing temperature (60°C, 140°F) and a higher COP. The ideal COP of this enhanced system using HFC-134a is 3.92.

Compressors leak approximately 15 to 30 grams (½ to 1 oz) of refrigerant per year (Siewert 1983) with additional losses occuring from the hoses.

Actual performance: the COP for the enhanced HFC-134a is calculated to be 2.04 based on a 61% compressor efficiency; 15% losses for the drive belt and magnetic clutch, fan and blower powers. This COP is approximately 41% of Carnot efficiency at the specified conditions.

Engine and road vibrations, temperature and pressure fluctuations, accidents, and servicing will cause actual refrigerant losses to increase in time from the 15 to 30 grams (½ to 1 oz) per year at the compressor shaft. The loss rate has been reduced significantly from the historical level as a result of compliance with the Clean Air Act in the United States mandating refrigerant recovery and prohibiting venting of the charge and conscientious service practices in Europe and Japan. Estimates of annual refrigerant losses range from 55 to 110 g (2 to 4 ozs) per year (Hara 1994, Watkins 1994). It is assumed that 95% of the refrigerant charge is recovered when the system is scrapped.

The weight of the enhanced HFC-134a air conditioner is approximately 13.6 kg (30 lbs).

Commercial potential: most major manufacturers in Europe, Japan, and North America have either completed or are in the process of converting their model lines to using HFC-134a as the refrigerant in air conditioners for new cars. Air conditioners using HFC-134a are considered a highly viable consumer product.

Transcritical CO_2 Compression Systems

An automobile air conditioner is being studied in Norway that uses CO_2 as the refrigerant. The principal advantage of this system would be the use of a refrigerant with an extremely low GWP (GWP=1). Most of the published data for the prototype CO_2 air conditioner are presented relative to CFC-12, so many references to CFC-12 are made in this section even though it is no longer widely used in new equipment.

Theoretical performance: the compressor only COP of this cycle is 1.9 based on an isentropic efficiency of 70%, 33°C (60°F) of superheat, cooler discharge temperature of 46°C (115°F), 4.4°C (40°F) evaporating temperature, and 11.7 MPa (1700 psia) compressor discharge pressure (Sand 1994).

Actual performance: two different estimates were made of actual system performance, one consistent with the system COP used for the HFC-134a air conditioner and one derived from Pettersen and Lorentzen's data (1993). An ideal cycle COP of 2.1 is shown in Fig. 24 at conditions comparable to those used for HFC-134a. Applying a compressor efficiency of 72% from Fig. 10 (page 29) and 15% losses for the drive belt, fan, blower, etc. results in a COP of 1.28.

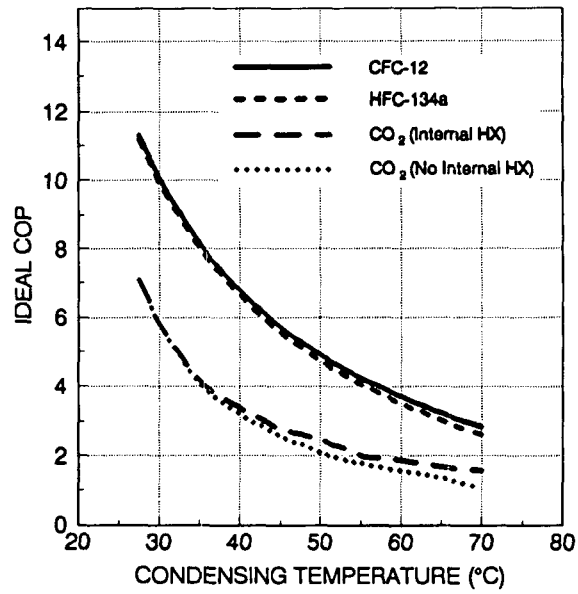

Fig. 24. Theoretical COPs for CFC-12, HFC-134a, and CO_2 air conditioners.

This estimate of the efficiency is significantly lower than the COPs in reported by Pettersen and Lorentzen for their prototype air conditioner. A second estimate was made using the data in Table 3 (see page 30, 31) assuming an ambient temperature of 25°C (77°F), 15% losses for the fan, blower, and drive mechanism, and that 30% of the operating time is at the low engine speed and 70% at the high speed. These assumptions result in a weighted average COP of 1.94 for the CO_2 air conditioning cycle (applying these assumptions for CFC-12 gives 1.90 which is consistent with the CFC-12 COP used in the Phase I report (Fischer 1991) and the COP for HFC-134a, 2.04).

There is a tremendous difference between the two COPs for the CO_2 air conditioner, and it is not known which is accurate. Some possible explanations are given on page 30. Both estimates are used to calculate TEWIs for this technology.

"The prototype CO_2 air conditioner was not designed with size and weight as important criteria. Both the compressor and receiver are quite large and heavy due to the available compressor parts, the desire to manufacture machined parts from steel, and the need to make the inside of the receiver accessible (Pettersen 1994)." The weights of the components listed in Table 6 show actual values for the prototype air conditioner and also the expected weights for a system designed for mass production. The use of lightweight materials and reductions in size, including smaller heat exchanger tubing and hoses possible with CO_2 make large weight reductions possible, particularly with regard to the compressor and receiver. The predicted weight of 11.7 kg (25.7 lbs) for the mass produced system is not greatly different from the 11.4 kg (25 lbs) for CFC-12 or 13.6 kg (30 lbs) for HFC-134a air conditioners.

Commercial potential: the concept of transcritical CO_2 compression for automobile air conditioning has been demonstrated with a laboratory prototype system, but has not been tested on a vehicle or undergone thorough field testing. The feasibility of consumer products based on this principal is uncertain and significant further developments in size and weight

reduction and reliability and safety testing are necessary.

Adsorption Air Conditioning

Adsorption air conditioning systems have been proposed for use in automobiles and trucks because of the opportunity to use waste heat to drive the system. Two different concepts are commonly mentioned as feasible systems, one using hydrogen and metal hydrides and the second using water and zeolites.

Theoretical performance: three different proposed concepts are discussed; a metal hydride adsorption air conditioner and two water/zeolite air conditioners. A private manufacturer in the U.S. has constructed and tested a prototype metal hydride automobile air conditioner that has a predicted COP of 2.65 with a 20°C (36°F) ΔT between supply and return air (DaCosta 1993). This temperature difference is comparable to the air temperatures in a car after the temperature in the passenger compartment has been pulled down to a comfortable level.

An air conditioner using water and zeolite with a cooling COP of 1.0 has been designed by Tchernev to operate at conditions appropriate for automobile air conditioning. The major disadvantage of this combination of refrigerant and sorbate is that it operates at a pressure significantly below atmospheric pressure, 0.8 kPa (0.12 psia), so that any leakage results in non-condensibles being drawn into the system causing it to shut down completely. No prototype has been constructed or tested for this system. An alternative water and zeolite air conditioner has been constructed which has a theoretical thermal COP of 0.55 and a 3.5 kW (12,000 Btu/h) cooling capacity (Schwarz 1993).

Table 5. Observed COPs for CFC-12 and CO_2 automobile air conditioners.

Engine Speed (rpm)	Ambient Temperature		COP	
	(°C)	(°F)	CFC-12	CO_2
800	25	77	2.90	3.07
800	43	109	1.67	2.04
800	55	131	1.09	1.50
2000	25	77	1.97	1.97
2000	43	109	1.35	1.38
2000	55	131	0.93	1.16

Table 6. Weight of laboratory prototype and predicted weight of CO_2 air conditioner

Component	Laboratory Prototype	Estimated Mass Produced System
Compressor (without clutch)	16.0 kg	4.0 kg
Evaporator	2.4	1.2
Gas Cooler	4.7	3.0
Receiver	16.1	2.0
Other (tubing, internal heat exchanger, etc.)	2.5	1.5
Total	41.7 kg	11.7 kg

Actual performance: the prototype metal hydride air conditioner is reported as having a measured COP of less than 2.5; some efficiency improvements are expected with system modifications. This system weighs 34 kg (75 lbs) and contains 12 kg (27 lbs) of metal alloy adsorbent. The prototype is quite bulky; 63 cm wide x 39 cm high x 46 cm deep (25 x 15.5 x 18 inches). Some improvement in size, weight, and performance is expected using alloys with higher heats of formation; significant reductions in size and weight are necessary for a commercial product to be viable.

The prototype water/zeolite air conditioner has measured COPs ranging from 0.27 to 0.31 during the desorption, cooling, process with a cycle average COP of 0.29 (Schwarz, Maier-Laxhuber, and Wörz 1993). The prototype uses two sorption beds with a combined inventory of 5.4 kg of zeolite. The total system weight is approximately 40 kg (90 lbs) (Schwarz and Maier-Laxhuber 1993). Schwarz stated that a supplemental gas burner will be required only 40% of the time for regenerating the system; 60% of the energy input would be provided through engine waste heat. A 125 W blower is used for circulating air. These COPs are based on a primary energy input and are not directly comparable to the COPs listed above which exclude the engine efficiency. The actual thermal COP of the HFC-134a and CO_2 air conditioners would be 0.51 and 0.49, respectively (using a 25% engine efficiency).

Commercial potential: several major obstacles must be overcome before adsorption air conditioners could be viable consumer products for automobile air conditioning. These include:

- reductions in size and weight to acceptable levels, present prototypes are acknowledged to be large and heavy even if "acceptable levels" is not stated precisely,

- demonstration of the vehicle operating conditions where adequate waste heat is available to drive the air conditioner and identification of conditions and energy input for when an auxiliary burner is required,

- demonstration of long term performance and capacity under frequent cycling of adsorbent beds (adsorption processes used for recovery of gases in manufacturering processes are known to degrade with use), and

- cost optimization to be competitive with compression technology.

Significant development and time are required before this technology could be commercialized.

Thermoelectric Cooling

Mei, Chen, and Kyle described a hypothetical thermoelectric air-conditioning system for an automobile (1992). Their system would consist of:

- a "high-voltage" dc generator (the voltage is unspecified and depends on the number of thermoelectric modules) in place of a compressor,
- a blower as used in a conventional automobile a/c system,
- a thermoelectric module acting as a cooler in place of the evaporator,
- a second thermoelectric module as a heat rejector behind the radiator where the condenser would normally be located, and
- a heat transfer loop using a blend of ethylene glycol and water to transfer heat from the cooler to the heat rejector.

Theoretical performance: analyses of hypothesized thermoelectric air conditioners have been done by Mei and Mathiprikasam. "Results show that the power required to deliver a cooling capacity of 4 kW (13,800 Btu/h) in a 38°C (100°F) environment will be 9.5 kW electric (Mathiprakasam 1991)." This corresponds to a 13°C (23°F) ΔT and a COP of 0.42. Mei saw this to be a limiting value stating that "[t]he maximum COP the TE system can achieve is around 0.42 (parasitic power not included) (Mei 1992)." Parasitic losses may be lower for a thermoelectric air conditioner than for a vapor compression system because of differences between the drive mechanism for a compressor and an alternator; alternator size will be larger than that for a vehicle with a vapor compression air conditioner. The COP of 0.42 is <u>not</u> a thermal efficiency because it does not account for the engine, belt, or alternator efficiency.

Actual performance: a prototype air conditioner has not been constructed, although some system design has been done and estimates of the size and weight are available. Mathiprikasam states that "[a] typical 127-couple TE module is approximately 30 mm square and 4 mm high, weighs about 25 g, and can pump about 10 W of heat against a temperature difference of 40°C (72°F) (1993)." Using the design cooling load of 4.17 kW (14,000 Btu/h) from Mei et al (1992), the cooler and the heat rejector would each weigh in excess of 10 kg (22 lbs). Data are not available for additional weight due to the heat transfer loop between the cooler and heat rejector or for any changes needed to the alternator due to increased electrical loads for the thermoelectric system. This weight has been assumed to be 2 kg (4.5 lbs), for a total system weight of approximately 22 kg (50 lbs).

Commercial potential: major breakthroughs in materials are necessary for this technology to be viable for automobile air conditioning and it is not likely to be commercialized on a wide scale in the foreseeable future.

Stirling Cycle Cooling

Stirling cycle refrigeration is occasionally mentioned as an alternative to vapor compression systems using a fluorocarbon refrigerant for automobile air conditioning. The Stirling cycle uses helium as the working fluid and would not have a direct effect on global warming due to refrigerant losses.

Theoretical performance: the theoretical COP of 8.0 for a Stirling cycle air conditioner corresponds to high- and low-side air temperatures of 35°C (95°F) and 11°C (52°F) with 5.6°C (10°F) ΔT's across the heat exchangers. While the theoretical efficiency of the Stirling cycle is very high, efficiencies of actual equipment are much lower.

Actual performance: engineering estimates of the COPs for a Stirling cycle air conditioner are reported to be 1.3 to 1.7, which is 70% to 90% of the efficiency of a comparable vapor compression system (STM); actual prototype test data are not available. The weight of a Stirling cycle air conditioner was estimated to be 31.8 kg (70 lbs) (STM).

Commercial potential: it is unlikely that a Stirling cycle air conditioner will be developed for automobiles. Significant developments must be made on bearings and seals, reliability and product lifetime, and costs.

Brayton (Air) Cycle Cooling

Equipment based on the Brayton, or air, cycle is used for air conditioning in military aircraft with recent developments in Europe for use in passenger planes and railroad passenger cars. At one time this technology was considered for use in automobile air conditioning, but it was never commercialized.

Theoretical performance: a theoretical efficiency has not been calculated for this technology.

Actual performance: estimated efficiencies for air cycle refrigeration range from 0.8 to 1.7 with a COP of 1.4 for automobile air conditioning operating conditions (AiResearch 1977). It is not known whether this includes parasitic energy consumption for the drive belt and magnetic clutch, fan, and blower. The estimated weight is 13.6 kg (30 lbs) (Bhatti 1991).

Commercial potential: this is not considered a viable technology for automobile air conditioning. Very finely machined, high efficiency components are needed to achieve COPs competitive with those of vapor compression air conditioners. Additional equipment would also be required to remove the latent cooling load in the passenger compartment.

TEWI Calculations

Greenhouse gas emissions for automobile air conditioners come from energy consumed to power the air conditioner, energy consumed from transporting the weight of the air conditioner, and from refrigerant losses. TEWI calculations are based on:

- a 7.0 kW (24,000 Btu/h) air conditioner,
- an engine efficiency of 25%
- 15% losses for fans, blowers, clutch and pulley drive mechanism,

- 2.32 kg CO_2/liter (8.79 kg CO_2/gallon) of gasoline,
- 0.243 kg CO_2/kWh heat content of fuel,
- 57 liters of gasoline/100 kg/10,000 km driven (10 gallons/100 lbs/10,000 miles), and
- an equipment lifetime of 11 years.

Carbon dioxide emissions from fuel consumption to power the air conditioner are estimated using (7):

$$CO_2 = \left[\frac{3.5\ \text{kW}}{\text{COP}}\right] \times \left[\frac{1}{0.25}\right] \times \left[\frac{\text{hours of A/C operation}}{\text{year}}\right] \times \left[\frac{0.243\ \text{kg}\ CO_2}{\text{kWh}_{\text{input}}}\right] \qquad (7)$$

National and regional average vehicle use are 16,400 km/y for North America (10,100 miles), 13,200 km/y for Europe (8180 miles/y), and 10,100 km/y (6280 miles/y) for Japan (Davis 1991). These data were used to derive regional averages of 107 hours of air conditioner operation in North America, 88 hours of operation in Europe, and 67 hours of operation in Japan which are consistent with the estimate of 47 liters of gasoline per 10,000 km driven (20 gallons/10,000 miles) used in the 1991 study (Fischer 1991).

Vapor Compression Air Conditioner: TEWI is calculated assuming a COP of 2.04, HFC-134a charge of 1.1 kg (39 ozs) in North America and Europe, 800 g (28 ozs) in Japan, a weight of 13.6 kg (30 lbs) annual emission rates of 55 and 110 grams, and that 95% of the refrigerant is recovered before the car is scrapped. The TEWI for North America is computed to be 5100 to 5900; for Europe, 4300 to 5100; and for Japan, 2500 to 3100 (the high and low ends of the range correspond to the 110 and 55 g/y HFC-134a loss rates).

CO_2 Automobile Air Conditioner: TEWI are calculated using two estimates of the system COP and with the weight of the laboratory prototype and an estimate of the weight of mass produced air conditioners. A COP of 1.94 was derived from laboratory measurements published by Pettersen and Lorentzen assuming 30% of the air conditioner use is at low and idle speeds and 70% at highway speeds and that there are 15% losses for the fan, blower, and belt drive mechanism (see page 91). A second estimated COP of 1.28 was also used based on a theoretical COP of 2.1 at 54.4°C (150°F) temperature and 15% losses for peripherals. System weights of 11.7 kg (26 lbs) and 41.7 kg (92 lbs) were used. Calculated TEWI are 4400 to 7200 for North America, 3600 to 5900 for Europe, and 1900 to 3200 for Japan.

Water/zeolite adsorption air conditioner: the major unknown in estimating the TEWI of this system is the percentage of the input power that can be provided effectively by waste heat. Calculations are reported assuming that an auxiliary burner is required to provide 70% of the driving energy and also 40%. The system is assumed to weigh 40 kg (90 lbs) and use a

125 W blower. The TEWI for North America is 3800 to 5900, 3200 to 4800 for Europe, and 1700 to 2600 for Japan.

Thermoelectric Air Conditioning: TEWI are calculated for the thermoelectric air conditioner using the COP of 0.42 reported by Mei and also for a COP of 1.2 (which corresponds to doubling the figure of merit, Z, of the thermoelectric couple). The weight is assumed to be 22 kg (48 lbs). The TEWI for North America is 8400 to 22,700; 6900 to 18,700 for Europe; and 3700 to 10,000 for Japan.

Stirling cycle air conditioner: TEWI are calculated for COPs of 1.3 and 1.7 and an air conditioner weight of 31.8 kg (70 lbs). The TEWI are 5500 to 6900 for North America, 4500 to 5700 for Europe, and 2400 to 3100 for Japan.

Conclusions

The TEWI for North America are displayed graphically in Fig. 25 and all of the results are listed in Table 7. In this figure the heavily shaded portion at the bottom of each bar corresponds to the CO_2 emissions from fuel consumption to power the air conditioner, fan, and blower (the adsorption systems have small segments representing energy use for the blower), a lightly shaded region from the weight contributions, and the HFC-134a bars have unshaded segments for the direct effects of refrigerant emissions. The conventional HFC-134a systems are shown with 10% and 5% refrigerant loss rates. Results for two different efficiency assumptions are shown for each of the alternative technologies (30% and 70% waste heat utilization for the adsorption system).

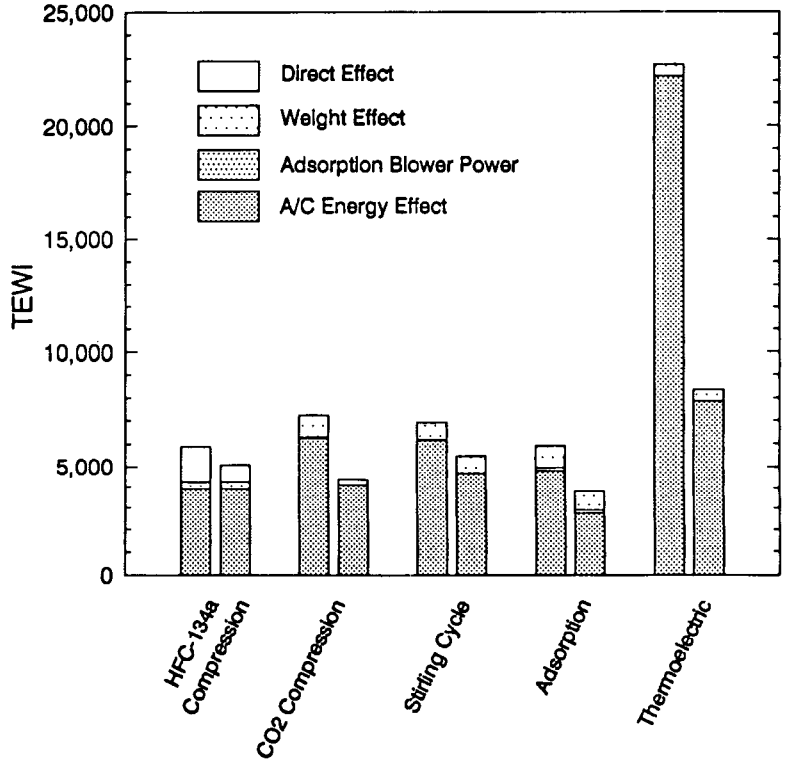

Fig. 25. TEWI for automobile air conditioners (North America).

Table 7. TEWI for alternative automobile air conditioners (North America).

Region Air Conditioner Technology	A/C & Blower Power	System Weight	Direct Effect	Total
North America				
HFC-134a Compression	1960	320	860-1640	3140-3920
Transcritical CO_2 Compression	2060-3130	280-990	1	2340-4120
Stirling Cycle A/C	2350-3080	760	0	3110-3840
Adsorption A/C	1550-2930	950	0	2500-3880
Thermoelectric A/C	3930-11,200	520	0	4450-11,740
Europe				
HFC-134a Compression	1610	260	860-1640	2730-3510
Transcritical CO_2 Compression	1700-2570	220-800	1	1920-3370
Stirling Cycle A/C	1940-2530	610	0	2550-3140
Adsorption A/C	1270-2410	770	0	2040-3180
Thermoelectric A/C	3230-9220	420	0	3650-9640
Japan				
HFC-134a Compression	1230	200	620-1200	2050-2630
Transcritical CO_2 Compression	1290-1960	170-610	1	1460-2570
Stirling Cycle A/C	1470-1960	470	0	1940-2400
Adsorption A/C	970-1830	590	0	1560-2420
Thermoelectric A/C	2460-7020	320	0	2780-7340

Transcritical CO_2 compression air conditioner could provide a significant drop in TEWI for automobile air conditioning if it works as well on a vehicle as has been reported for the lab prototype and if technical hurdles can be overcome. The TEWI derived from the theoretical COP of 2.1 could be higher than the TEWI for the HFC-134a air conditioner, even with 10% losses, depending on the weight.

TEWI for the Stirling cycle air conditioners (COPs of 1.3 and 1.7) are slightly higher than the TEWI for the HFC-134a system for the 10% and 5% loss scenarios. In each case, the Stirling system has a higher fuel consumption.

The TEWI for the adsorption air conditioner is comparable to the HFC-134a air conditioner with 10% losses if waste heat is adequate to power the system 30% of the time it is in use, it is higher than the HFC-134a system with 5% losses. Significant reductions in TEWI could occur if a greater portion of the input power could be provided by waste heat. It is unknown how effectively waste heat can be used to drive this system.

Thermoelectric air conditioning at the current stage of development would have a dramatically higher TEWI than any of the alternatives because of the very low COP. The second bar for this technology assumes a breakthrough in materials developed that resulted in doubling the thermoelectric figure of merit. This would be a major development, and yet the TEWI would still be higher than any of the other alternatives considered.

The transcritical CO_2 automobile air conditioner exists only as a single prototype system and has not been tested or proven under actual operating conditions. Concerns about the effects of the high operating pressures on lifetime and reliability have not been investigated thoroughly; and a rigorous safety and risk analysis should be performed. The weight effect is relatively small so uncertainties in the projected weight of a mass produced system are not terribly important. This technology is not close to commercial application and further development is needed before field testing can be performed. Acknowledging the long lead times associated with major changes by the automobile industry, it is unlikely that a commercial product could be in widespread use before 2003-2010.

The laboratory prototypes for adsorption automobile air conditioners have shown that this technology has potential for reducing global warming impacts from this application. Size and weight need to be reduced, efficiency and lifetime need to be demonstrated, and much more testing is needed. The TEWI for the water/zeolite system is comparable to that of HFC-134a if 30% to 40% of the drive energy is provided by waste heat, 50% to 60% waste heat utilization is needed to reduce TEWI 20% compared to HFC-134a. Actual values for waste heat utilization would need to be demonstrated to reduce this uncertainty. Product lifetime needs to be demonstrated through extensive field testing to study the long term effects of frequent cycling of the adsorption modules. Performance can deteriorate through "poisoning" of the sorbent beds. Adsorption cooling represents a dramatic change in technology relative to compression systems and the lead time to commercialization is probably at least as long as that for transcritical CO_2 compression, which is more of an evolutionary development from conventional equipment.

CHILLERS

Introduction

Air conditioning in large multi-story buildings and very large single-story buildings is commonly provided by centrifugal, reciprocating, or screw water chillers (water cooled evaporators). Cooling capacities for these systems range from 260 to 26,000 kW (75 to 7500 tons). Chillers using centrifugal compressors comprise approximately 70% of the worlds installed chiller capacity, with about 30% using reciprocating, screw, or scroll compressors (Fischer 1991). Historically, nearly two-thirds of the centrifugal chillers have been low pressure machines using CFC-11 or HCFC-123, about 12% high pressure systems using CFC-12 or HFC-134a, with the balance using HCFC-22, R-500, or CFC-114. The discussion in this section focuses on centrifugal chillers and alternative cooling technologies for large commercial and institutional buildings.

ALTERNATIVE TECHNOLOGIES

HCFC-123

In the past, the vast majority of centrifugal chillers have used CFC-11. HCFC-123 has been the refrigerant of choice for retrofitting these machines as well as for new low-pressure chillers because its thermodynamic properties are very similar to those of CFC-11. HCFC-123 has a very short atmospheric lifetime and consequently has an extremely low ozone depleting and global warming potentials. Very efficient chillers are commercially available that use HCFC-123.

Theoretical performance: steady-state efficiencies were computed using thermodynamic properties at an evaporator temperature of 4.4°C (40°F) and condensing temperature of 40°C (104°F) to compare performance with other refrigerants. These "ideal" COPs are 6.95 for 0°C (0°F) subcooling and superheat, 7.10 for 2.8°C (5°F) subcooling and superheat, and 7.24 for 5.6°C (10°F) subcooling and superheat (0.506 kW/ton, 0.495 kW/ton, and 0.486 kW/ton, respectively).

Actual performance: chiller efficiencies have improved significantly since the Montreal Protocol was first adopted and chillers in 1994 are more efficient than the products using CFCs that they are replacing. Chillers using HCFC-123 are commercially available with

COPs of 6.76 (0.52 kW/ton) at the ARI rating conditions. Significant progress has also been made in reducing refrigerant losses from purge units for low pressure chillers. At least one OEM is marketing chillers with refrigerant emissions of less than 0.5% of the charge per year (Smithart 1993); this is significantly less than the projected "future loss rate" of 4% used in the parametric analysis in the Phase I study. ARI (the Air Conditioning and Refrigeration Institute) estimated that the total refrigerant make-up rate resutling from losses in purge units, leakage, maintenance, and recovery upon equipment retirement would average 0.75% of the charge per year over the lifetime of the chiller (Hourahan 1994).

Commercial potential: chillers using HCFC-123 are commercially available and there is a very strong market for these machines.

HFC-134a

HFC-134a is the refrigerant of choice for the new generation of chillers that evolved from chillers designed to use CFC-12 and for retrofitting CFC-12 chillers. HFC-134a has no ozone depleting potential and a moderate global warming potential. HFC-134a chillers are commercially available.

Theoretical performance: steady-state efficiencies were computed using thermodynamic properties at an evaporator temperature of 4.4°C (40°F) and condensing temperature of 40°C (104°F) to compare performance with other refrigerants (Smith 1993). These "ideal" COPs are 6.48 for 0°C (0°F) subcooling and superheat, 6.68 for 2.8°C (5°F) subcooling and superheat, and 6.87 for 5.6°C (10°F) subcooling and superheat (0.543 kW/ton, 0.526 kW/ton, and 0.512 kW/ton, respectively).

Actual performance: efforts to remain competitive vis-à-vis HCFC-123 chillers have led to developments and innovations that have boosted HFC-134a chiller efficiencies to COPs of 6.76 (0.52 kW/ton) at ARI rating conditions. These machines use cycle modifications and turbine technology to recover some of the expansion losses to boost efficiencies and the COP of 6.76 is not directly comparable to the efficiencies given above which are theoretically possible based solely on evaporating and condensing temperatures.

Commercial potential: chillers using HFC-134a are commercially available and there is a very strong market for these machines.

Ammonia Chillers

Major chiller manufacturers have looked at using ammonia in chilled water systems, and while some machines are commercially available there is very little hard performance data on them because these are primarily customized units.

Theoretical performance: steady-state efficiencies were computed using thermodynamic properties at an evaporator temperature of 4.4°C (40°F) and condensing temperature of

40°C (104°F) to compare performance with other refrigerants (Smith 1993). These "ideal" COPs are 6.78 for 0°C (0°F) subcooling and superheat, 6.83 for 2.8°C (5°F) subcooling and superheat, and 6.87 for 5.6°C (10°F) subcooling and superheat (0.519 kW/ton, 0.515 kW/ton, and 0.512 kW/ton, respectively).

Actual performance: no hard data were located comparing operating efficiencies of ammonia and other chiller systems. Ammonia chillers are commercially available in Europe and some are reported to be more efficient than HFC-134a chillers (Mosemann 1993), but the absolute efficiencies were not identified (the comparison could have been to a very good, or a very poor, HFC-134a chiller). Ammonia and HCFC-22 are expected to have comparable efficiencies in chillers using screw compressors (Calm 1994).

Commercial potential: centrifugal chillers using ammonia are commercially available and could have good efficiencies, though development effort will be needed to remain competitive with the improving efficiencies of HCFC-123 and HFC-134a chillers. Building and fire codes that have evolved restricting the use of ammonia out of concerns for public welfare make it difficult to obtain the permits and licenses necessary to use ammonia chillers in urban areas of many countries. These institutional factors limit the potential for ammonia chillers to replace CFC, HCFC, and HFC chillers. An appraisal of the potential benefits from expanded use of ammonia as a refrigerant and the risks associated with liberalizing regulations affecting its use are beyond the scope of this study.

Absorption Chillers

Gas-fired absorption chillers are a commercially available alternative to vapor compression centrifugal chillers. Kohler reported on the current status of absorption chiller technology and the opportunities to substitute gas-fired chillers for electric driven centrifugal chillers using HCFCs or HFCs (1993):

> "Absorption water chillers are sold in the U.S. in capacities from 100 to 1500 tons (350 to 5300 kW).... The principal competition for absorption chillers is electrically driven vapor compression equipment. The decision as to which product to use is generally made based on life cycle costs. Since absorption chillers have higher first costs, significant operating cost savings or other incentives are required to justify their purchase.
>
> "Double effect machines represent the most efficient commercially available absorption technology.... As a rough guide, double effect chillers sell for a $250/ton ($71/kW) premium over electric chillers with COPs of 5.75 or better. [T]he incremental cost of electricity for the locality in question is a key factor in the life cycle costing [of electric and absorption chillers]"

Primary energy efficiencies are used in speaking of gas-fired absorption chillers and these COPs are not directly comparable to the COPs listed for electrically driven chillers. Gas-fired

COPs do not include electrical use for pumps, fans, or blowers and it is stated whether or not they include combustion losses.

Theoretical performance: theoretical efficiencies of single-effect absorption chillers are not given. Although gCOPs of 1.2 to 1.3 have actually been achieved for double-effect chillers, this serves as a reasonable theoretical limit because of the very large heat exchanger surfaces required; gCOPs of 2.0 have been reported, but these are based on 0.6 °C ΔT's (1 °F) in the heat exchangers instead of the 6 °C (10 °F) ΔT's for realistic surface areas (DeVault 1994). In speaking of the triple-effect absorption chiller, Kohler says:

> "Assuming 'practical' amounts of heat exchange surface and 'reasonable' maximum operating temperatures, various researchers have claimed full load COPs from under 1.3 to over 1.8. Whether or not machines with COPs over 1.8 are practical remains to be seen as several obstacles to their construction exist. These COPs do not account for combustion losses in the case of direct fired equipment." (1993)

Actual performance: actual COPs of single effect, indirect fired chillers typically peak around 0.7 and COPs for double-effect direct fired absorption chillers are on the order of 1.0 with COPs of 1.2 widely available for steam fired double effect chillers (Kohler 1993). The lower COP for the direct fired chillers result from losses in the combustion process.

Commercial potential: absorption chillers are a viable technology and under some economic circumstances compare favorably with centrifugal chillers using fluorocarbon refrigerants. The efficiencies which are ultimately achieved in commercially available equipment will depend on many market conditions and not simply on comparisons of CO_2 emissions from alternative technologies. Factors such as relative costs of natural gas and electricity, peak load charges, and equipment first costs enter into the chiller selection process. Absorption chillers are common in Japan where electricity costs reflect the expense of investing in additional generating capacity, and are used to a much lesser extent in the U.S., possibly in combination with one or more centrifugal chillers and controlled in such a way as to minimize operating costs.

Evaporative Coolers

Evaporative coolers are usually used as alternatives to unitary air conditioners in residential applications, though much larger sized units are commercially available for use in office buildings, hospitals, etc. This technology is described on page 38.

Theoretical performance: theoretical performance has not been computed for evaporative cooling; that calculation is highly site specific and beyond the scope of what can be performed in this study.

Actual performance: there are no government or industry rating systems to estimate energy use of evaporative cooling units. These systems require power input only for fans and small

water pumps so the input is low compared to vapor compression cooling systems, but that does not make it easy to perform a good comparison of the two cooling concepts. The energy loss mechanisms are significantly different between the two. Ventilation losses are normally 0 to 25% for vapor compression air conditioning with an additional loss of about 5% for thermal and volumetric losses in the ducts (Watts 1988). Direct evaporative cooling systems do not recirculate the cooled air as do vapor compression air conditioners and use 100% outdoor makeup air; consequently ventilation losses can run as high as 90% while duct losses are again about 5%.

The performance of an evaporative cooler depends on its design, the ambient dry bulb temperature, and also the coincident wet bulb temperature. Any analysis of energy use for evaporative air conditioning must account for each of these factors and consequently should be done on an hourly basis and is very site specific. Seasonal or annual performance calculations were not conducted for this study; instead results prepared by others have been used. Supple (1982) used a computer analysis with typical reference year weather tapes to evaluate energy use for cooling a commercial building with a specific design of an indirect evaporative cooling air conditioner. He reported results for four cities in the U.S.:

- Albuquerque, New Mexico; 69% of the cooling requirements could be satisfied with evaporative cooling at 0.09 kW/kW (0.31 kW/ton),
- Tucson, Arizona; 63% of the cooling requirements at 0.06 kW/kW (0.22 kW/ton),
- Chicago, Illinois; 42% of the cooling requirements at 0.23 kW/kW (0.81 kW/ton), and
- Savannah, Georgia; 15% of the cooling requirements at 0.25 kW/kW (0.89 kW/ton).

Improved efficiencies are possible in the arid western climates where 0.06 to 0.09 kW_{input}/kW_{output} (0.22 to 0.31 kW/ton) for evaporative cooling compares very favorably with 0.15 kW/kW (0.52 kW/ton) for a centrifugal chiller using HCFC-123 or HFC-134a. Comparison in the mid-west (Chicago) and south-east (Savannah) are very bad. Chiller performance remains essentially unchanged by climate at 0.15 kW/kW (0.52 kW/ton), while the evaporative units cannot meet all of the cooling requirements and use 0.23 to 0.25 kW/kW (0.81 to 0.89 kW/ton).

Commercial potential: it is unusual for evaporative cooling to be used in commercial buildings with large cooling demands, and while gains may be made in market share, it is unlikely that this technology can substitute entirely for centrifugal chillers in many applications. Evaporative cooling may be used economically in conjunction with desiccant cooling (particularly in supermarkets) or with vapor compression air conditioning to reduce equipment capacity, improve efficiency, and minimize life-cycle costs in some instances.

TEWI Calculations

TEWI for centrifugal and absorption chillers are considered extensively in the section covering sensitivity analyses beginning on page 153.

TEWI calculations for an indirect evaporative air conditioning system are shown in Table 8 in a commercial building in Tucson and Chicago and also for a centrifugal chiller in the same cities. Efficiency data cited earlier from Supple (1982) are used for the evaporative air conditioners while efficiency data from Glamm (1993), Smithart (1993), and Calm (1994) are used for the chiller. The refrigerant make-up rate is assumed to be 0.75% (ARI 1993).

Table 8. TEWI for evaporative air cooling and centrifugal chillers in Tucson and Chicago

Technology Characteristic	Tucson, Arizona		Chicago, Illinois	
	Evaporative Cooling	Centrifugal Chiller	Evaporative Cooling	Centrifugal Chiller
Efficiency (kW/kW)	0.063	0.15	0.23	0.15
Annual Energy Use (kWh)	116,000	276,000	423,000	276,000
Annual CO_2 emissions	75,000	179,000	275,000	179,000
Lifetime Indirect Effect	2.25×10^6	5.37×10^6	8.25×10^6	5.37×10^6
Refrigerant Charge (kg)	0	273	0	273
Loss Rate (kg/year)	0	0.75%	0	0.75%
Lifetime Losses (kg)	0	61	0	61
Direct Effect	0	0.005×10^6	0	0.005×10^6
TEWI (equivalent kg)	2.25×10^6	5.37×10^6	8.25×10^6	5.37×10^6

The TEWI for Tucson is less than 40% that of a centrifugal chiller using HCFCs to provide the same comfort conditions. TEWI in Chicago, however, is 55% higher for the evaporative cooler than it is for the chiller (note: Supple also concluded that evaporative cooling could not meet all of the loads in Chicago, this has not been taken into account). The direct effect of the refrigerant emissions is insignificant.

CONCLUSIONS

Evaporative cooling provides a viable alternative to vapor compression technologies in dry climates if adequate water is available at reasonable cost. Significantly lower TEWI are possible in those climates where evaporative cooling provides comparable comfort levels at high energy savings. Although applications of evaporative cooling can be pushed into more humid climates, differences in TEWI will diminish as the energy-savings advantage decreases. Energy savings, and lower TEWI (not calculated), are possible in most high humidity regions by using an evaporative system to precool condenser air for a vapor compression system (Foster 1990), but at the cost of purchasing and maintaining two mechanical systems.

REFRIGERATOR/FREEZERS

Introduction

The global warming impact of a refrigerator/freezer includes indirect emissions of CO_2 resulting from the combustion of fossil fuels to generate electricity to operate the refrigerator and also from the working fluid or refrigerant and the gas used to produce the foam insulation. This particular application has seen a great deal of work throughout the world to identify and develop alternatives to CFC-12 as the working fluid and CFC-11 as the foam blowing agent, so an abundance of performance data are available. The existence of a lot of data, however, does not necessarily make for an easy comparison of alternatives; this can be seen clearly in the Öko-Kühlschränke test data in Fig. 26. Each data point represents the normalized energy use that was measured for a commercially available 3-Star refrigerator in Europe (3-Star units have fresh-food and freezer compartments). The solid triangles are measurements for refrigerators using cyclopentane as the foam blowing agent and butane as the refrigerant. The open circles represent measurements for cyclopentane as the blowing agent and HFC-134a as the refrigerant; and the open squares represent models using HFC-134a as both the blowing agent and the refrigerant. Depending on which points are selected, it is possible to make comparisons where almost any combination of refrigerant and blowing agent is more efficient than refrigerators using one of the other combinations. The broad scatter makes it virtually impossible to make definitive conclusions about the application of each refrigerant or blowing agent.

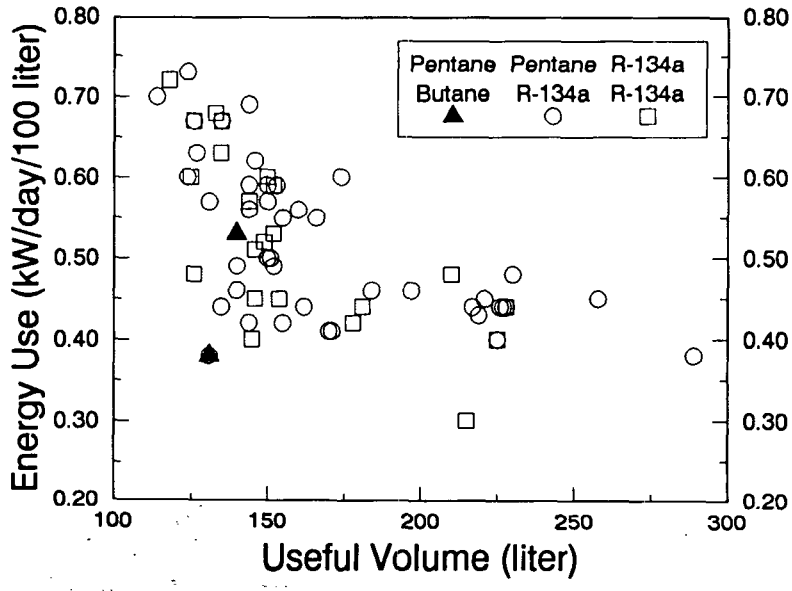

Fig. 26. Measured energy use for refrigerator/freezers using CFC alternatives.

The inconclusive nature of system operating data made it necessary to rely on computer models to estimate the performance of different combinations of refrigerants and blowing

agents to ensure that comparable systems are being evaluated. The Refrigerator Analysis (ERA) Program, Version 1.0, was used to simulate the performance of conventional (vapor compression) European and North American sized refrigerator/freezers for several combinations of alternative refrigerants and insulations. (The ERA Model has been a useful tool for comparing refrigerator design options, but it has limited absolute accuracy in predicting actual energy use of actual refrigerators (Swatkowski 1994)). Calculations were first performed for CFC reference cases in order to verify the simulated performance against the base cases established in the first cooperative project between AFEAS and the Department of Energy. The base cases were then modified to represent the more energy efficient appliances representative of commercial products in 1994.

Alternative Technologies

Refrigerants

HFC-134a is the refrigerant of choice to replace CFC-12 in refrigerator/freezers in most of the developed countries because it is non-flammable, non-toxic, and efficient. Propane, isobutane, and mixtures of propane and isobutane are used in other countries because they are comparable in efficiency to HFC-134a and have much lower GWPs.

Theoretical performance: a theoretical COP of 2.76 was calculated for both HFC-134a and propane using the Lee-Kesler-Plöcker equation-of-state based on 54°C (130°F) condensing, -23°C (-10°F) evaporating, and 32°C (90°F) liquid line and suction gas temperatures.

Actual performance: it is clear from Fig. 26 that each of these refrigerants have achieved efficiencies comparable to the other two.

Commercial potential: each of these refrigerants has been established in commercial products and future expanded use will depend on balancing economic and environmental considerations. German manufacturers have demonstrated that the flammable hydrocarbon refrigerants can be handled safely in the manufacturing process and consumers have been willing to accept appliances using hydrocarbons. Regulations affecting mandatory refrigerant recovery and recycling in the U.S. and elsewhere are demonstrating that fluorocarbon refrigerants can be contained and atmospheric emissions and environmental damage avoided.

Absorption Refrigeration

The Platen-Munters absorption refrigerator uses a non-condensible gas to equalize the pressure between the evaporator and the condenser and all solution flows are accomplished by gravity and bubble pumps. This results in a heat-actuated domestic refrigerator with no

moving parts (Marsala 1993). More than a million of these machines are manufactured and sold annually in world-wide markets and these products are commonly used in recreational vehicles because of their ability to use bottled gas and in hotel rooms because of their extremely quiet operation. These products are limited in size because of constraints imposed by the operating cycle (e.g. reliance on bubble pumps) and cannot automatically be scaled up to displace conventional compression products.

Theoretical performance: a theoretical efficiency was not calculated for this technology.

Actual performance: while many of absorption refrigerators are in use throughout the world, energy use information is not available which can be compared to data available for other technologies. The small units generally have low efficiencies compared to HFC or hydrocarbon compression refrigerators.

Commercial potential: absorption refrigerators will continue to fulfill niche requirements, but are very unlikely to be broadly adopted as replacements for vapor compression refrigerators.

Thermoelectric Refrigeration

Theoretical performance: data for commercially available thermoelectric modules shown in Fig. 9 have a COP of 0.32 at approximately the same conditions as those used for determining the efficiency of vapor compression refrigerators.

Actual performance: actual performance data are unavailable.

Commercial potential: thermoelectric refrigerators will continue to fulfill niche applications but are very unlikely to replace vapor compression refrigeration because of the extremely low efficiencies.

HCFC And HFC Blowing Agents

A literature survey was conducted to identify the most up-to-date information on plastic foam insulations using HCFCs, HFCs, water, and hydrocarbon as blowing agents. Data were gathered for fresh and aged foam thermal conductivities, foam densities, amounts of blowing agents, and information about auxiliary blowing agents. The thermal properties used in the calculations are summarized in Table 9 [Doerge 1991, De Vos 1992, Murphy 1993, and Stewart 1993]. Data shown in Table 9 are fresh foam k-factors, consistent with findings published by SPI for refrigerator/freezers [SPI 1988]. Other sources cite higher thermal conductivities for aged appliance insulations than those shown in Table 9, however the relative values are very similar (e.g. Swatkowski (1994) uses a thermal conductivity of aged cyclopentane blown foam that is 13.6% higher than that of HCFC-141b blown foam; the values in Table 9 show cyclopentane blown foam to be 12.8% higher).

Table 9. Thermal properties of plastic foam insulations.

Foam Blowing Agent	Thermal Conductivity		Thermal Resistivity	
	mW/mK	Btu-in. / h-ft^2-°F	m^2-°C/ W-cm	h-ft^2-°F / Btu-in.
HCFC-141b	16.8	0.117	0.595	8.6
HCFC-142b/HCFC-22	18.6	0.129	0.538	7.7
HFC-134a	19.1	0.133	0.524	7.5
HFC-365	18.1	0.126	0.552	7.9
CO_2	23.0	0.160	0.435	6.3
Cyclopentane	19.0	0.132	0.526	7.6

The choice of data for the computer calculations was compounded, to some extent, by recent efforts to improve all the properties of foams blown with HCFC-141b, not just the thermal behavior. The data selected in this case is based on using a small percentage of water in the foam formulation to give a reduced cell size and improved performance. More recent work on this blowing agent has focused on using perfluorinated pentane and other perfluorinated compounds (PFAs) and carbon black in order to give lower density foams than those formed using water as a co-blowing agent and also slightly better thermal properties. Although GWPs are not known for the PFAs, they are expected to be very high [Fisher 1994] and consequently will result in high TEWIs even at extremely low PFA concentrations (<0.5%) in the foam formulations. The high direct effect of PFAs on TEWI is evident in the results presented later comparing two formulations of HCFC-142b/HCFC-22 blends blowing agents. The data in Table 9 for 22/142b are based on the use of the careful selection of polyol-isocyanate formulation to form smaller cell sizes foaming process. The measurements in Table 9 for HFC-365 is for foam using a small percentage of HFC-134a as a co-blowing agent. Additional work is ongoing to optimize formulations of HFC blowing agents for improved thermal and mechanical properties.

Theoretical performance: the theoretical performance of blowing agents is not computed.

Actual performance: fresh-foam k-factors and thermal resitivities are shown in Table 9 for several fluorocarbon blowing agents.

Commercial potential: significant effort has been devoted toward developing foam formulations for use with HCFC-141b, HFC-134a, and HCFC-142b/HCFC-22 and these blowing agents are in commercial use. Further work is required for HFC-365 and other HFC blowing agents and their commercial potential cannot be evaluated in 1994.

Hydrocarbon Blowing Agents

Cyclopentane is used as a blowing agent for refrigerator/freezer production in Germany.

Theoretical performance: a theoretical efficiency is not calculated.

Actual performance: fresh-foam k-factor and thermal resisitivity is shown for cyclopentane blown foam in Table 9.

Commercial potential: cyclopentane is used in commercial production of refrigerators and may experience increased use as an alternative to fluorocarbon blowing agents depending on relative manufacturing costs and thermal properties.

Evacuated Panel Insulation

Computer simulations were also performed for refrigerators using composite wall construction of evacuated panels and plastic foam in the North American refrigerator. These panels can be formed using air and moisture tight metal or plastic barriers to contain a low conductivity filler material such as diatomaceous earth, aerogels, open cell polyurethane foam, or glass fibers. The actual selection of materials that appliance manufacturers make will depend on material and panel manufacturing costs, weight, and proven performance. No effort was directed toward selecting "the best" panel materials or construction; five levels of center panel conductivities are used between 3 and 7 mW/mK (R-20.6 to R-48 h-ft^2-°F/Btu-in.) were used [Nipkow 1993, Schilf 1993, Caps 1993, Reuter 1993, McGrath 1993]. Panels are assumed to be used in the doors and each wall of the cabinet, centered in each surface, and to extend to within 2 cm (0.79 in.) of each edge. This results in approximately 90% surface coverage for the refrigerator. ERA includes an algorithm that approximates the losses from lateral heat flow and the "edge effects" of vacuum insulation, so these losses are taken into account in the calculations.

Simulations for refrigerators with composite wall construction are based on constant overall wall thicknesses; the combined thickness of the vacuum panel and surrounding insulation for each wall of the refrigerator is the same as the foam thickness in the simulations for refrigerators without vacuum panels. One of the advantages of vacuum panel insulation is that thinner walls could be used while maintaining or improving the cabinet performance and also increasing usable internal volume; this was not done in these simulations. Finally, the manufacturer of very high resistivity evacuated panels has compared their own simulations with ERA with laboratory measurements and believes that ERA underestimates improvements in cabinet performance with high resistance walls [McGrath, 1993].

Theoretical performance: the theoretical performance of evacuated panels is not computed.

Actual performance: the actual performance of evacuated panel insulation is very strongly affected by the dimensions of the panels because of the edge effects. Data reported for

panels are usually in terms of "center of panel" measurements which can be significantly lower than the "total panel" values which determine how effective the panels are in a refrigerator cabinet. These total panel values are not reported.

Commercial potential: evacuated panels may be a viable option that is considered in future along with other energy saving design changes (e.g. higher efficiency compressors, better door gaskets). Evacuated panels could be implemented on a commercial scale if they are among the most cost-effective energy saving alternatives and have acceptable product lifetimes.

Methodology

North American Refrigerator/Freezer

The refrigerator/freezer simulations for a North American design are based on a 510 liter (18 ft^3) frost-free (automatic defrost) model. The compressor is assumed to have a 5.40 energy efficiency ratio (EER). The compressor capacity was adjusted in each of the simulations in order to give a calculated 50% run time; the compressor efficiency was not reduced at the low capacity levels that would be required for some of the cabinets with vacuum insulation. Below some threshold limit, low capacity compressors will not be as efficient as larger compressors because some friction and blow-by losses do not scale with size. This was not taken into account. Calculations were performed to simulate the standard rating test used in the U.S.; 32.2°C (90°F) room temperature, -15°C (5°F) freezer temperature, and 3.3°C (38°F) fresh food temperature. The freezer door and wall thicknesses are assumed to be 6.00 cm (2.36 in.) and 7.50 cm (2.93 in.), respectively, while the door and wall thicknesses for the fresh food section are assumed to be 4.00 cm (1.77 in.) and 6.00 cm (2.36 in.). The calculated energy use for the baseline refrigerator/freezer using CFC-11 blown foam and CFC-12 for the refrigerant is 654 kWh/year (1.79 kWh/day).

European Refrigerator/Freezer

The European refrigerator is assumed to be a 230 liter (8 ft^3) model with a small freezer compartment. The compressor is assumed to have an EER of 4.40. Temperatures in the simulations are set at 25°C (77°F), -18°C (-0.4°F), and 5°C (41°F) for the room, freezer compartment, and fresh food compartment. Calculations were performed only for HFC-134a and cyclopentane blown foams, these foams with vacuum panels, and with HFC-134a or the hydrocarbon blend as refrigerants. The freezer door and wall thicknesses are both assumed 6.75 cm (2.66 in.), while the door and wall thicknesses for the fresh food section are assumed to be 3.50 cm (1.38 in.) and 4.50 cm (1.77 in.). The calculated energy use for the baseline refrigerator/freezer using HFC-134a as both the refrigerant and blowing agent is 336 kWh/year (0.92 kWh/day) or 0.40 kWh/day/100 liters.

RESULTS

North American Refrigerator/Freezer

Calculated energy use for the North American refrigerator/freezers are listed in Table 10. These results and data for foam density, weight percent blowing agent, and refrigerator dimensions were used to compute the TEWI for each combination of insulation and refrigerants shown in Fig. 27; TEWI were not calculated for cabinets using HFC-365 blown foam insulation because estimates of the GWP for HFC-365 are not available. The TEWI are calculated using 100 year GWPs, a 15 year service life for the appliance and an average CO_2 emissions rate of 650 g CO_2/kWh. The direct impacts on global warming shown in Fig. 27 for the fluorocarbons range from 15 to 29% of the total equivalent warming impact. The refrigerators using HCFC-141b have 15% of the TEWI due to fluorocarbon emissions and those using HFC-134a as a blowing agent have 18% of the TEWI from chemical emissions. In each case, almost all of the direct effect is due to the foam blowing agent; assumptions on mandatory recovery of the refrigerant would result in marginal reductions in the TEWI.

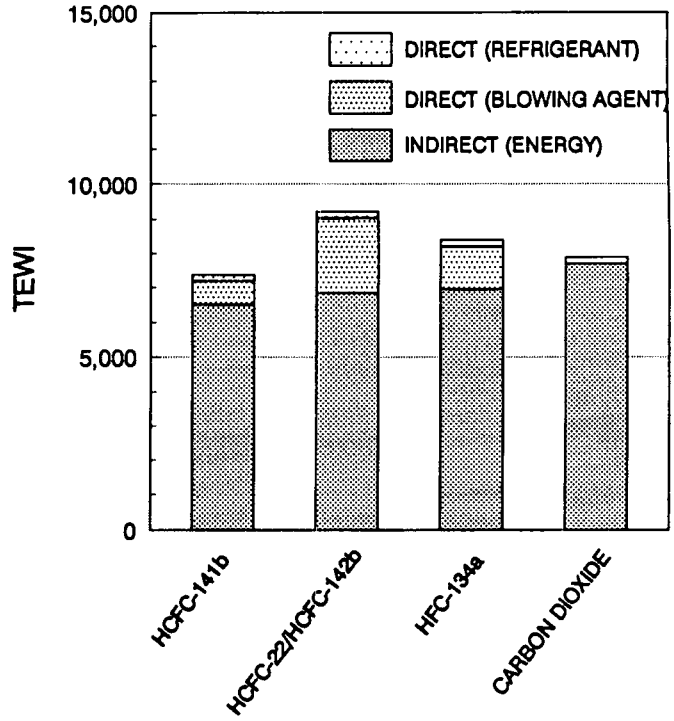

Fig. 27. TEWI for 510 l (18 ft^3) refrigerator/freezers using HFC-134a and foam insulations.

Perfluorinated Pentane as an Auxiliary Blowing Agent: Table 11 contains results for two different formulations of foams using HCFC-22/HCFC-142b blends as the blowing agent. Data in the second column are for a formulation that relies on the selection of polyol-isocyanate to form cells with reduced cell sizes, that in the third column is for a formulation using a perfluorinated compound as a nucleating agent to give finer cell size and to reduce frothing. There are several differences between the foams represented in Table 11, including the foam densities and weight percent blowing agents, but what is of principal interest in this discussion is the use of a PFA as a nucleating agent in the second foam. Even at a very low weight percent PFA, 0.3%, the perfluorinated pentane increases the direct effect of the chemical emissions by 162 kg equivalent of CO_2, almost 11%. A 2 to 3% reduction in k-factor for the foam would be needed to offset this increase in the direct effect due to using the PFA.

European Refrigerator/Freezer

The calculated energy use for the European style refrigerator/freezer are also contained in Table 10. There are no significant differences between the energy consumption for the refrigerators using HFC-134a as both refrigerant and blowing agent and those using propane/isobutane and cyclopentane because of the accuracy of the computer model the general assumptions about compressor and cycle performance. Measurements of actual compressor performance with the two refrigerants need to be used to make a more definitive comparison. The hydrocarbon refrigerators are estimated to use 1.1 to 2.6% less energy than a similar models using HFC-134a both as the refrigerant and foam blowing agent. Only small percentage of these differences are real, because of the higher thermal conductivity assumed for the HFC-134a blown foam (19.1 vs 19.0 mW/mK); differences in refrigerant performance are outside the accuracy of the compressor and cycle simulations performed.

Table 10. Calculated energy use for refrigerator/freezers (kWh/d).

	Refrigerant		
	230 liter		510 liter
Blowing Agent	HFC-134a	HC-290/HC-600a	HC-134a
HFC-134a	0.92	NC*	1.89
Cyclopentane	0.92	0.91	1.88
Carbon Dioxide	1.02	0.99	2.10
HCFC-141b	NC	NC	1.77
HCFC-22/HCFC-142b	NC	NC	1.86
HFC-365	NC	NC	1.84

* combination of refrigerant and blowing agent not considered

Table 11. TEWI for refrigerator/freezers using HCFC and PFA foam insulation.

Property	Polyol-Isocyanate	Perfluorinated Nucleating Agent
k-Factor (mW/mK)	17.1	18.6
Density (kg/m3)	33	34
Direct Effect		
HFC-134a	186	186
HCFC-22	282	516
HCFC-142b	451	825
PFA	0	162
Total	919	1689
Indirect Effect	9200	9700
TEWI	10,100	11,400

TEWI shown in Fig. 28 for the European refrigerator are calculated using a 15 year service life for the appliance and CO_2 emission rate for electricity generation of 513 g CO_2/kWh. These results show the same trends as those in Fig. 27 for the North American refrigerator/freezer, although the percentage of direct effects is higher, due primarily to the lower CO_2 emissions for the power generation mix in Europe (higher percentage of nuclear and hydroelectric power generation with virtually 0 CO_2 emissions). The lower rate of CO_2 emissions for power gen-

eration results in a relatively smaller indirect effect for the European refrigerator than the same unit would have in North America, a smaller overall TEWI, and hence a larger percentage direct effect than the North American refrigerators.

Results for refrigerators using HFC-134a as the refrigerant and foam blown with HCFC-141b or CO_2 are shown in Fig. 29 both with and without (NP) evacuated panels. The panels are assumed to have center k-factors from 7 to 3 mW/mK. The panels used in composite walls with HCFC-141b blown foam can reduce the TEWI from 12 to 18% relative to the refrigerator only with HCFC-141b blown foam. Evacuated panels used in conjunction with CO_2 blown foam could achieve the same TEWI as refrigerators with HCFC blown foam without panels, assuming that CO_2 blown foam characteristics are such that composite walls could be manufactured.

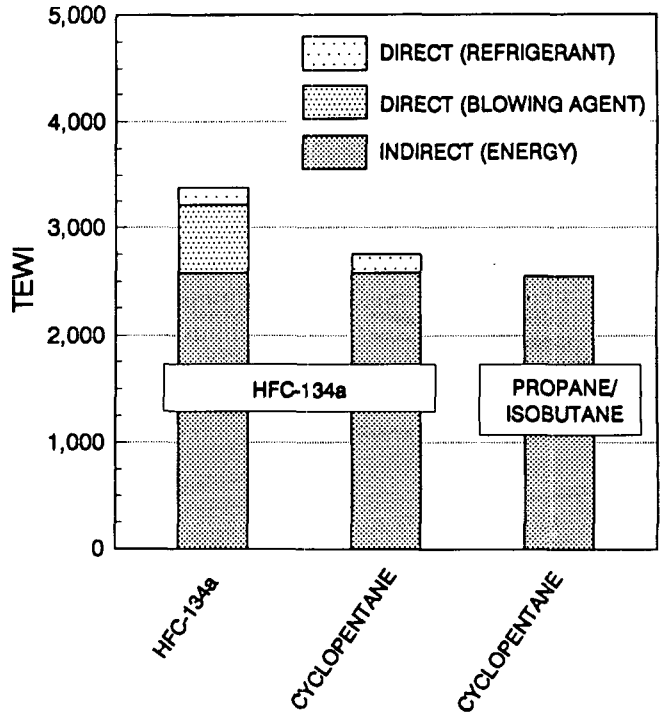

Fig. 28. Total equivalent warming impact for 230 l (8 ft^3) refrigerator/freezer.

An evaluation of energy requirements of a thermoelectric refrigerator was also performed using efficiencies from Fig. 9 (page 27) and ERA simulations of a 230 liter (8 ft^3) European style refrigerator. Energy consumption for the reference refrigerator using a blend of propane and isobutane was calculated to be 1.334 kWh/day; 1.045 kWh/day for the compressor (including 0.021 kWh/day for cycling losses) and 0.289 kWh/day for fans, lights, controls, etc. The calculated steady-state COP for the hydrocarbon compressor is 1.177. Data for commercially available thermoelectric modules shown in Fig. 9 have a COP of 0.32 at approximately the same conditions as those used with ERA for calculating the hydrocarbon performance in a refrigerator (25 °C ambient instead of 27 °C ambient temperatures and a 20 °C ΔT between the ambient and fresh food compartment, freezer energy consumption is not included in the thermoelectric calculations). Adjusting the cooling input power requirements to account for the differences in compressor and thermoelectric efficiencies, and adding the parasitic power consumption (but not the compressor cycling losses), yields an estimated energy consumption of 4.055 kWh/day for the thermoelectric refrigerator. The TEWI of the thermoelectric refrigerator is thus 11,400 (based on a 15 year useful life and 513 g CO_2/kWh_e) compared with a TEWI of 3300 for the corresponding HFC-134a refrigerator.

DISCUSSION

No attempt has been made to evaluate or balance the advantages or disadvantages of the fluorocarbon and hydrocarbon based refrigerators. There are no significant differences in the TEWI for the HCFC-141b/HFC-134a refrigerator and the cyclopentane/propane & isobutane refrigerator; one uses an HCFC (and hence can only be an interim product) and has an impact from direct emissions of blowing agent and refrigerant, the other has uses flammable materials and has a slightly higher energy consumption. North American sized refrigerators using HCFC-142b/HCFC-22 blends or HFC-134a as blowing agents have appreciably higher TEWI that of the refrigerator using HCFC-141b blown foam. This difference may diminish as foam formulations are optimized. The difference in TEWI between the European sized fluorocarbon and hydrocarbon fridges must be weighed against the risks associated with the hydrocarbon refrigerator (European and American risk analyses have questioned the safety of large, automatic defrost refrigerators using hydrocarbons). The projected energy use of refrigerators using HFC-365 blown foam is only 2% higher than that for a comparable hydrocarbon refrigerator. Further developments could result in similar or even lower energy consumption using HFC-365 blown foam; the big unknown in comparing products then is the GWP for HFC-365.

Finally, the actual performance of the appliance will depend on the thermal properties of the insulation at the operating temperatures. Generally the insulating ability of the foams increases at lower temperatures, however there is a range where the thermal conductivity goes up for foams blown with gaseous blowing agents (e.g. CFC-11, HCFC-141b, cyclopentane). As the temperature goes down, these blowing agents begin to condense in the cells of the foam, reducing the percentage of insulating gas in the cells and increasing the overall thermal conductivity. This phenomenon does not occur with foams formed with low boiling blowing agents (e.g., HFC-134a, HCFC-22/HCFC-142b) over the range of temperatures relevant to a household refrigerator/freezer. The variation of k-factor with temperature has not been taken into account in any of the calculations in this study.

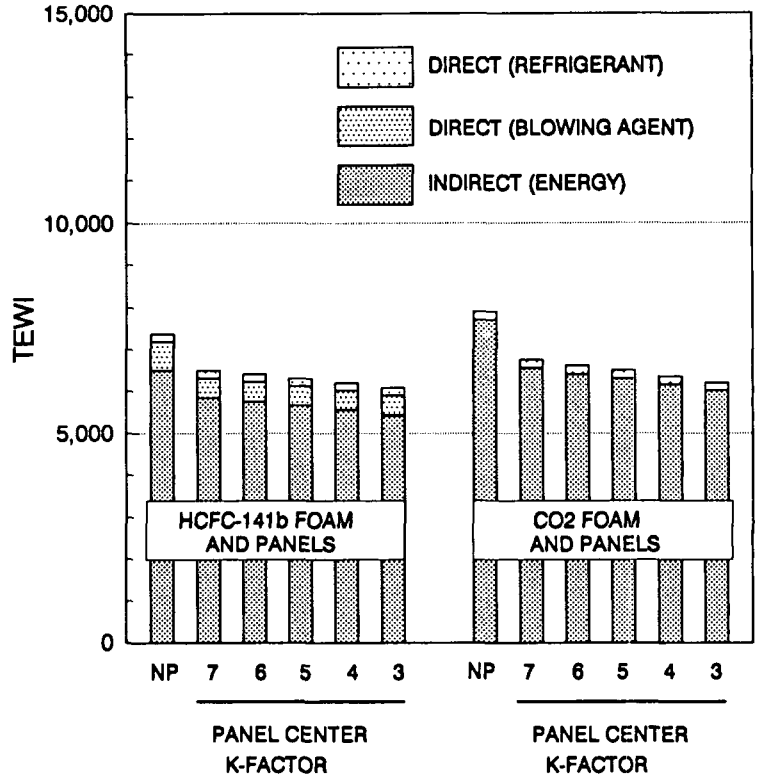

Fig. 29. TEWI for refrigerators with evacuated panel insulation.

SUPERMARKET REFRIGERATION SYSTEMS

Introduction

Refrigeration systems in large, modern supermarkets use racks of compressors in a machine room, with condensing units installed on the roof or outside at ground level, and refrigerant lines running beneath the floor to the evaporators in individual display cases on the sales floor. The high internal volumes of these systems necessitate large refrigerant charges while the multitude of joints and connections require vigilant maintenance to minimize leakage and to ensure efficient system performance and product quality.

Supermarket refrigeration is arbitrarily divided into three categories for discussion purposes based on the display case temperatures (equivalently the evaporating temperatures or evaporator inlet air temperatures). These categories are customarily referred to as low temperature, medium temperature, and high temperature refrigeration. Ice creams and frozen meats and foods clearly fall into the category of low temperature with evaporating temperatures as low as -40°C (-40°F) and evaporator exit temperatures between -18° and -32°C (0° to -25°F). Medium temperature display cases for meats and dairy products have evaporator exit air temperatures between -2° and 7°C (28° to 45°F). Fresh produce in high temperature cases have air temperatures around 10°C (50°F). Stores will also have food preparation areas, such as rooms for meat cutting, and walk-in freezers and coolers operated from the central refrigeration system.

Refrigerant selections in the past have been made on the basis of overall efficiency with compromises to reduce the total number of refrigerants used. Typically, two different refrigerants will be used in a store, one for low temperature cases and one for high temp, each selected based on their efficiencies, and one or the other of these two used for the medium temperature cases. Large stores have gravitated toward using CFC-12 for high temperature refrigeration, R-502 (a blend of HCFC-22 and CFC-115) for low temp, and then either CFC-12 or R-502 for medium temp. Government policies resulting from the original provisions of the Montreal Protocol led the manufacturers of compressors, as well as condensing units and display cases, to adapt their product lines to use HCFC-22 in medium and low temperature and to develop equipment to use HFC-134a in high temperature applications. Revisions to the protocol to phase-out HCFCs led to efforts by the fluorocarbon manufacturers to develop mixtures of HFCs (chlorine-free and ozone-safe refrigerants) that closely matched the performance of R-502 or HCFC-22 so further equipment modifications could be minimized.

The New York State Energy Research and Development Authority reported that the switch from R-502 to HCFC-22 in the early 1990's resulted in a 2.5 to 5.5% higher energy

consumption for low temperature refrigeration depending on whether reciprocating or screw compressors are used (Albrecht 1993). Recently tests are being reported for mixtures of refrigerants for replacing R-502. The data shows better performance than R-502 across some ranges of evaporating and condensing temperatures and poorer performance across other ranges of conditions.

Generally, efficiency drops at higher condensing temperatures. In looking at the changes for low temperature refrigeration, "energy efficiency [for one chain of supermarkets] was not a major factor in selecting the refrigerant because there is not that much difference; a bad mechanic can make a greater difference [than there is between the refrigerants] (Matthews 1993)."

Alternative Technologies

R-502 has been the refrigerant of choice for ice cream and frozen food display cases in supermarkets because of its high capacity, low energy consumption, and acceptable compressor discharge temperature at evaporating temperatures as low as -40°C (-40°F). R-502 is an azeotropic blend of CFC-115 and HCFC-22, and the scheduled phase-out of CFCs has resulted in the conversion of existing equipment and marketing of new equipment using HCFC-22. This switch required changes in compressor design to avoid unacceptably high discharge temperatures with HCFC-22; among these are both internally and externally compounded compressors with gas cooling before the second stage and liquid injection in single stage compression.

Revisions to the Montreal Protocol that provide a schedule for a total phase-out of HCFCs has resulted in the development of chlorine-free (ozone-safe) mixtures of refrigerants for low temperature refrigeration. The Air Conditioning and Refrigeration Institute (ARI) organized an effort by U.S., Japanese, and European compressor and refrigeration equipment manufacturers to evaluate the performance of some of these mixtures. Most results that were published prior to 1994 are for calorimeter testing of compressors which have not been fully optimized for the new refrigerants. Several supermarket chains are testing complete refrigeration systems using HFCs and HFC mixtures, although most of the information available in late 1993 are in the form of qualitative instead of quantitative comparisons.

Fluorocarbon Mixtures

Several different mixtures of fluorocarbons are being evaluated by grocery chains for use in low and medium temperature refrigeration; R-404A (HFC-125/HFC-143a/HFC-134a); R-407A (HFC-32/HFC-125/HFC-143a); R-507 (HFC-125/HFC-143a); and a mixture of HFC-32, HFC-125, and HFC-143a. HFC-134a is widely accepted as the leading candidate for replacing CFC-12 in high temperature refrigeration.

Theoretical performance: the efficiency of the low, medium, and high temperature

Table 12. Relative efficiency of supermarket refrigerants.

Refrigerant Designation	Components	Supermarket Temperature Range		
		Low (% R-502)	Medium (% R-502)	High (% CFC-12)
R-404A	HFC-125/HFC-143a/HFC-134a	96%	97%	
R-407A R-407B	HFC-32/HFC-125/HFC-134a HFC-32/HFC-125/HFC-134a	97% 96%	98% 97%	
R-507	HFC-125/HFC-143a	97%	97%	
none assigned	HFC-32/HFC-125/HFC-143a	97%	98%	
R-717	Ammonia with direct hx secondary loop with 2.8°C ΔT secondary loop with 5.6°C ΔT	93% 83% 75%	104% 99% 95%	92% 77% 66%
R-134a	HFC-134a			101%

alternatives are listed in Table 12 relative to R-502 (for medium and low temp) and CFC-12 (for high temp)[2]. Generally the COPs are two to four percentage points below the COP of the baseline refrigerant. The addition of an additional ΔT from a secondary loop to accomodate ammonia can result in a significant loss in efficiency.

Actual performance: R-404A has a 5 to 7% higher refrigeration capacity than R-502 (Menzer 1993 and Griffith 1993) and a lower discharge temperature and higher efficiency for low and medium temperature refrigeration (Hickman 1994). Calorimeter measurements have shown R-404A to have a 2 to 3% higher COP than R-502 at low temperature conditions (Menzer 1993), although a seasonal analysis by a compressor manufacturer reported that annual energy use would be 3% lower for the mixture than for R-502 at -29°C (-20°F) evaporating and a minimum condensing temperature of 21°C (70°F) (Griffith 1993). Annual energy consumption is virtually identical to that for R-502 for -29°C (-20°F) evaporating if the minimum condensing temperature is 27°C (80°F) (Griffith 1993).

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[2] Low temperature: -32°C (-25°F) evaporating, 43°C (110°F) condensing, -18°C (0°F) suction gas, 18°C (65°F) liquid line
Medium temperature: -7°C (20°F) evaporating, 43°C (110°F) condensing, 7°C (45°F) suction gas, 18°C (65°F) liquid line
High temperature: 7°C (45°F) evaporating, 43°C (110°F) condensing, 21°C (70°F) suction gas, 18°C (65°F) liquid line

Calorimeter testing of R-407A showed a 4% higher COP at -40°C (-40°F) evaporating and 32°C (90°F) condensing, but 2% lower COP at -29°C to -12°C (-20°F to 10°F) evaporating and 32°C (90°F) condensing (Menzer 1993). The COP is less than R-502 if subcooling is not used (Hickman 1994). Recent work by Snelson (1994) has shown that the refrigerant temperature glide with R-407A can be used to advantage to improve performance and that R-407A can have a higher COP than R-502 across a broad range of evaporating and condensing temperatures.

R-507 "has a volumetric cooling capacity about 10% higher than that of R-502 with similar COPs for evaporator temperatures of -40°C (-40°F) and higher" (Hickman 1994). Seasonal performance calculations for the azeotropic blend of HFC-125 and HFC-143a give a 2% improvement in seasonal performance with a minimum condensing temperature of 21°C (70°F) and a 1% increase in efficiency with a minimum condensing temperature of 27°C (80°F) (Griffith 1993). One refrigeration engineer reported that in a field installation this mixture has virtually the same energy use as R-404A and that they are both "very, very close to the efficiency of R-502 (Oas 1993)."

Commercial potential: these mixtures of HFCs have demonstrated high performance in supermarket refrigeration and have strong potential for commercialization as a replacement for R-502 and HCFC-22 as a refrigerants for low and medium temperature refrigeration. HFC-134a is a strong candidate to replace CFC-12 in high temperature applications.

Ammonia with Secondary Heat Transfer Loop

Ammonia is frequently mentioned as an alternative refrigerant for commercial refrigeration that does not damage stratospheric ozone or have any direct effect on global warming. It has been used safely in refrigerated warehouses and food processing plants throughout the world. The use of ammonia is restricted in North America and Japan because of its toxicity and flammability. Discussions of expanding the use of ammonia into retail refrigeration systems generally include the use of a secondary heat transfer loop which pumps chilled brine from the machine room to the individual display cases. An indirect loop of this nature isolates the ammonia from the sales floor, keeping it in the machine room and the condensing unit outside the store, providing greater protection for customers and employees. Such a system is more complicated than conventional refrigeration systems with direct heat transfer and consequently is more expensive and incurs losses in efficiency due to the additional ΔT between the ammonia and the brine and from the pumping power for the secondary heat transfer fluid.

Theoretical performance: the relative performance of ammonia is also shown in Table 12 for each of the supermarket refrigeration temperature ranges. Ammonia with direct expansion shows a degraded performance relative to R-502 and CFC-12 by as much as 8%. Performance can be down as much as 34% for indirect systems, accounting for energy use for pumping the secondary fluid would make this loss even higher.

Actual performance: American and European supermarkets have considered the use of ammonia and secondary heat transfer loops. Unfortunately the most current information compares an ammonia system using a secondary loop with a supermarket system using HCFC-22, not R-502 (when supermarkets converted from R-502 to HCFC-22 they found that energy use increased a couple of percent). Kruse reported on projects in Germany to design and test such systems comparing the performance of ammonia with HCFC-22 (1993). An engineering study found that compared to a refrigeration system using HCFC-22 with direct heat transfer, an indirect system using ammonia would have:

- 15% higher first costs,
- 7% higher energy consumption, and
- only 15% of the refrigerant charge of the HCFC-22 system (Kruse 1993).

This same study reported that an indirect system using HCFC-22 with an indirect heat transfer loop would have:

- 8% higher first costs than an equivalent system using HCFC-22 without the secondary heat transfer loop,
- 10% higher energy consumption, and
- only 18% of the refrigerant charge

and that an indirect system with HFCs would have:

- 10% higher first costs than the HCFC-22 system with direct heat transfer,
- 10% higher energy consumption, and
- only 17% of the refrigerant charge.

The possible benefits from the use of ammonia or the use of an HFC with a reduced charge in an indirect system need to be balanced with the higher first cost and increased energy consumption resulting from the secondary heat exchanger. The use of HFCs with low loss fittings and valves and rigorous routine system maintenance may be as effective, or more effective, at reducing TEWI at a more acceptable cost.

Commercial potential: it is likely that there will be some use of indirect systems with brine or glycol loops in Europe but not in the U.S. or Japan. This design has an increased cost and complexity with lower efficiency than other viable alternatives.

Stirling Cycle

Stirling cycle refrigeration (see page 19) has been proposed as a viable alternative to compression systems for supermarket refrigeration, particularly for low-temperature display cases. It uses helium as the refrigerant which is neither an ozone depleting substance nor a greenhouse gas.

Theoretical performance: Stirling enjoys a theoretical efficiency advantage over vapor compression at temperatures below -30°C (-22°F) (Miniatt 1993). The capacity and efficiency are also very flat across a broad range of ambient operating temperatures, unlike vapor compression systems, so annual energy use is expected to be lower.

Actual performance: prototype test results have not been reported so it is not possible to compare the actual performance of a Stirling cycle display case with a conventional vapor compression system.

Commercial potential: Stirling cycle equipment has not demonstrated either the long lifetimes or energy efficiencies required for this class of equipment. A clear demonstration of hardware performance is necessary before significant commercial interest will develop.

Magnetic Refrigeration

Magnetic refrigeration is also being developed for a possible application in low-temperature supermarket display cases (DeGregoria 1993). The proof-of-principal has been demonstrated, but construction and testing of prototype units have not been completed. This technology is expected to have substantially higher first costs with potential savings in operating costs. It is impossible to project TEWI for the technology at this time.

Theoretical performance: a theoretical efficiency has not been computed.

Actual performance: prototype performance has not been demonstrated or reported.

Commercial potential: commercial interest could develop in this technology if there are significant breakthroughs in high temperature superconducting materials.

TEWI for Supermarket Refrigeration

Data for supermarket energy consumption and particularly refrigeration system charge and emission rates are rarely published. Almost all information that is available is anecdotal. The 1991 report cited data obtained from a supermarket chain in the United States that has been on the forefront of evaluating energy saving designs and alternative refrigerants. The baseline system used in the analysis is for an existing store where energy consumption and refrigerant charge were measured for the low, medium, and high temperature refrigeration equipment. The refrigerant loss rates can vary widely from one store to another and this information is rarely documented in publications. Large supermarkets can have literally kilometers of refrigerant lines running between the machine room, condensing unit, and display cases on the sales floor. Consequently there are joints and connections where leakage can occur and loss rates can be very high. Refrigerant make-up rates of 20 to 40% of the charge per year are required for some supermarkets with losses possibly as high as 50%

(Borhanian 1993). The calculations of TEWI for supermarket systems with direct heat transfer assume that one-third of the charge is lost annually through leaks and servicing.

TEWI calculations are derived from a baseline refrigeration system using R-502 and CFC-12 since the only information available for the alternative refrigerants is stated relative to R-502 and CFC-12. The baseline store is assumed to use:

- 400 kg (880 lbs) of R-502 for low temperature refrigeration and 1120 kg (2500 lb) of CFC-12 for medium and high temperature cooling,
- 223,000 kWh per year for low temperature refrigeration and 316,000 and 80,000 kWh per year for medium and high temperature refrigeration, respectively.

The refrigeration system is assumed to have a useful lifetime of 20 years. Energy consumption is converted to indirect emissions of CO_2 from power generation using the North American average of 0.650 kg CO_2/kWh of electricity.

Alternative refrigeration systems are considered using several of the proposed mixtures of HFCs for low temperature refrigeration and HFC-134a for both medium and high temperature display cases, preparation areas, and walk-in coolers. Estimates for energy consumption are computed using the values for the baseline system and relative efficiency comparisons listed above. System charges of the alternative refrigerants are estimated based on the ratios of their liquid densities at room temperature to the densities of R-502 and CFC-12. The results for low, medium, and high temperature systems are shown in Fig. 30, Fig. 31, and Fig. 32. Each of the alternatives considered show a dramatic reduction in the direct effect from refrigerant emissions compared with the R-502 and CFC-12 refrigeration systems, but the direct effects are still large. This is a consequence of both the high leakage rate and the relatively high GWP of each of the components of the mixtures. The differences in the CO_2 emissions from energy consumption are virtually indistinguishable in these figures.

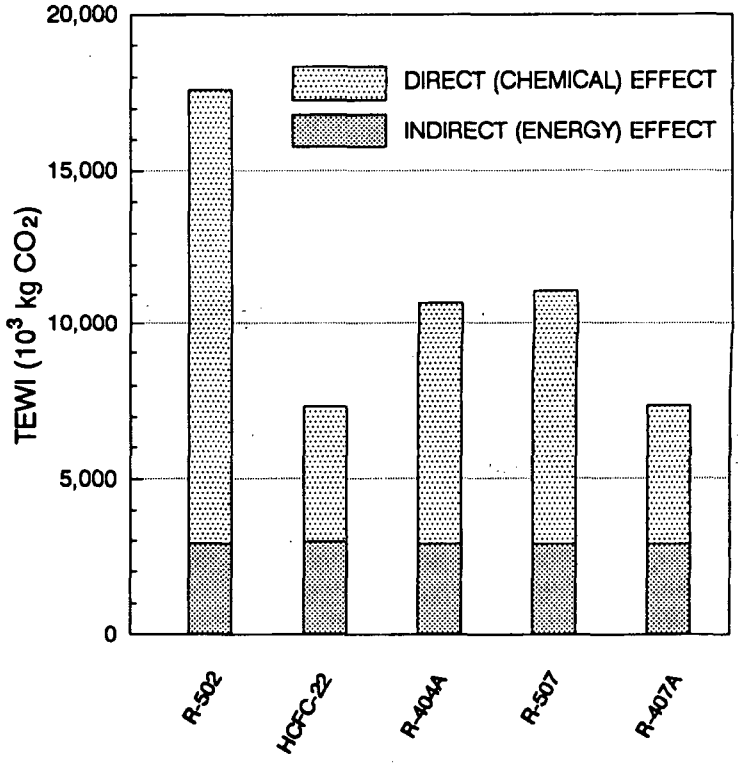

Fig. 30. TEWI for low-temperature supermarket refrigeration.

The increasing prices of refrigerants, raised environmental consciousness of retailers, and governmental regulations are all contributing toward efforts to control refrigerant losses. These efforts are resulting in the development and use of low loss fittings and valves, resistance heaters for evaporator defrost (in place of hot gas defrost which stresses fittings), and greater use of self-contained display cases which eliminate the need for long refrigerant lines and large charges. Some design engineers contend that a 5% loss rate is achievable. The significance of such developments on reducing TEWI is shown in Fig. 33 where the "historical" one-third loss rate and the target 5% loss rate are compared for low-temperature refrigeration. The use of self-contained display cases with hermetic compressors, improved component designs and increased awareness of the need to reduce losses will make significant contributions to reducing TEWI for supermarket refrigeration.

Conclusions

HFC mixtures have been developed and demonstrated that reduce the TEWI for conventional supermarket refrigeration systems without increasing the component from energy consumption. The alternative systems still have large refrigerant charges, lengthy refrigerant lines with many joints and connections, and high leakage rates. The direct contribution to global warming from refrigerant emissions continues to be high with both of the mixtures considered. Significant reductions in TEWI could be achieved using either of three approaches:

- the development and use of low loss fittings and valves in conjunction with a rigorous maintenance program,
- the use of self-contained, hermetic refrigeration systems in stand-alone display cases, and
- the use of HFCs or ammonia in refrigeration systems with secondary heat transfer loops.

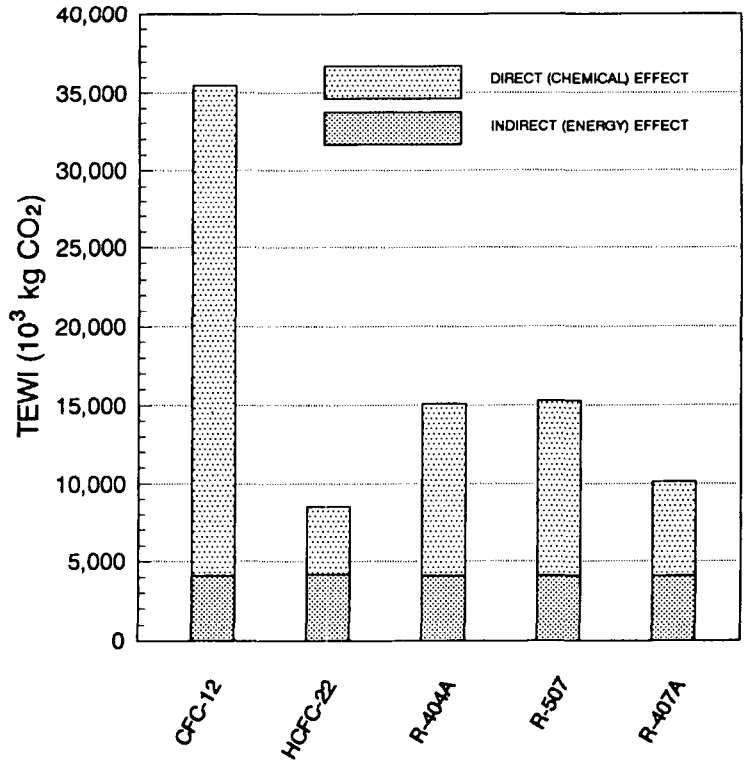

Fig. 31. TEWI for medium-temperature supermarket refrigeration.

Government regulations and policies and refrigerant prices are likely to force end-users to reduce refrigerant emissions from their historically high rates.

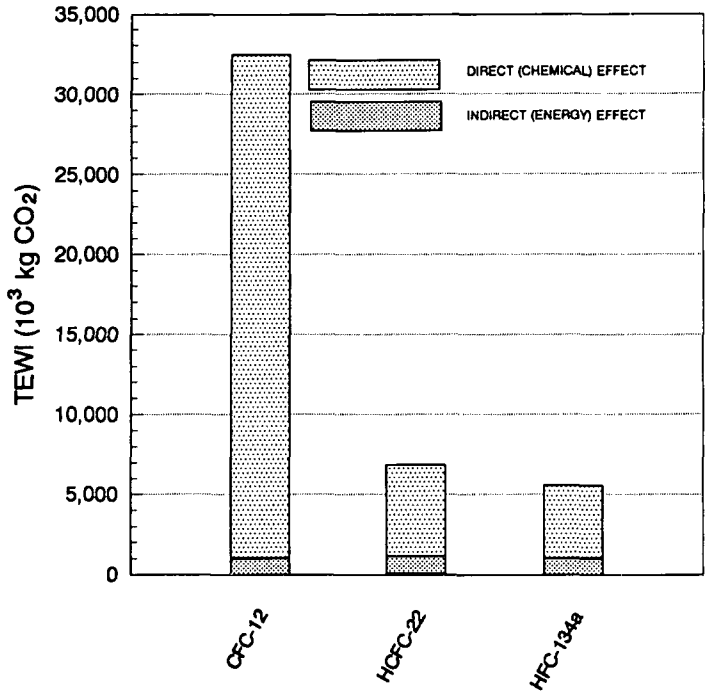

Fig. 32. TEWI for high-temperature supermarket refrigeration.

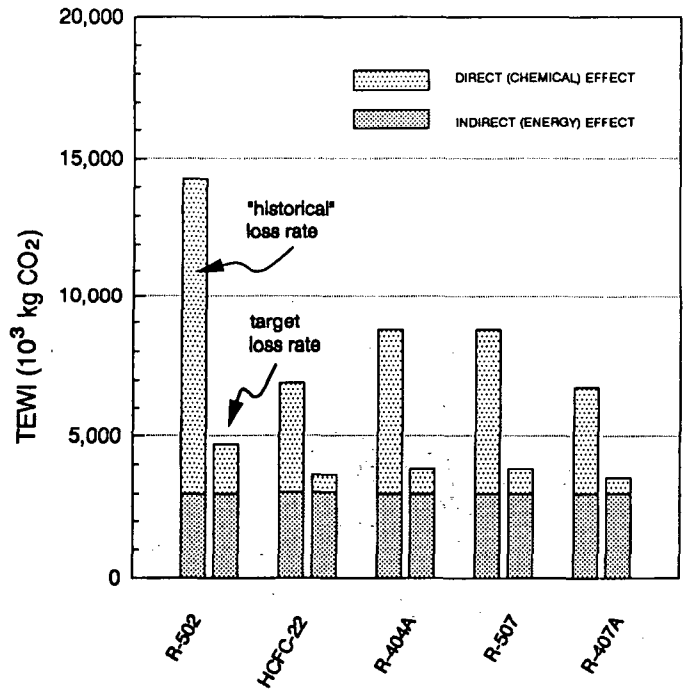

Fig. 33. TEWI for low-temp refrigeration with historical and target loss rates.

UNITARY EQUIPMENT

Introduction

Vapor compression residential and light commercial air conditioning systems using HCFC-22 are common throughout the United States and Japan and electric heat pumps using HCFC-22 are used in the mid-latitude regions of both countries where there is a favorable balance of heating and cooling requirements. Air conditioning is becoming more common in parts of Europe and in the developing countries. The efficiency of unitary equipment has improved steadily since at least 1975 as indicated in Fig. 20 on page 73. This trend in improving efficiency will continue even as new refrigerants are adopted. The market for unitary equipment is highly competitive and driven to a large degree by consumer costs. Alternative refrigerants with markedly lower cooling capacities than HCFC-22 are at a disadvantage to alternatives with similar or higher capacities because any loss in volumetric capacity will necessarily need to be made up with larger compressors and heat exchangers. That translates into a higher cost to the consumer. Alternatives that can deliver the same capacity at a lower cost will have an advantage in the marketplace.

Technology Description

A typical split system consists of an outdoor unit containing the compressor, one heat exchanger (condenser for cooling, evaporator for heating mode with a heat pump), and a fan, and one or more indoor heat exchangers and blowers (evaporator for cooling, condenser for heating mode with a heat pump). Industry selected HCFC-22 for these applications because of its efficiency and high cooling capacity.

Alternative Technologies

Although many pure refrigerants have been considered as alternatives to HCFC-22 in unitary equipment, only HFC-134a has been commercialized. Propane comes close to matching the thermophysical properties of HCFC-22 and some limited testing has been conducted, but it has not generated much interest in the United States or Japan because of its flammability. Propane is discussed more seriously in Europe, but the market for unitary equipment has been very small in Europe and developments for this application are driven by the needs in the U.S. and Japan. Ammonia is mentioned frequently by representatives of the

environmental movement and will be discussed below. Other fluorocarbons, such as HFC-32 and HFC-152a, are flammable and have not been pursued as pure refrigerants.

HFC Mixtures

HFC-134a is the only pure refrigerant that has received much attention as an alternative to HCFC-22 in unitary air conditioning; it has a significantly lower capacity than HCFC-22. Recent efforts toward replacing HCFC-22 have gravitated toward mixing two or more refrigerants in attempts to find mixtures that are ozone-safe, non-flammable, non-toxic, efficient, and have capacities similar to or better than HCFC-22. Much of this work has been done with computer analyses and laboratory testing to find mixtures that come close to matching the performance of HCFC-22 so few if any major equipment changes would need to be made. Several other mixtures have been identified that behave as pure refrigerants; they do not experience any change in temperature during condensation or evaporation at constant pressures. These "azeotropic" mixtures have substantially different operating pressures and capacities than HCFC-22 and relatively major equipment redesign and optimization is required to use them most efficiently.

Theoretical performance: ideal steady-state COPs for HFC mixtures considered as refrigerants for unitary equipment are shown in Table 13 at four test conditions[3]. While most of these show COPs a few percentage points lower than HCFC-22, there are a couple that have theoretical efficiencies higher than HCFC-22. These mixtures, however, are zeotropic blends of HFCs and experience large temperature glides in the evaporator and condenser; significant development effort will be needed to realize the higher efficiencies in actual system operation.

Actual performance: The HFC mixtures have been tested extensively in compressor calorimeters with limited "drop-in" testing or testing in "soft-optimized" heat pumps or air conditioners. Thirty-eight manufacturers from Europe, Japan, and North America have sponsored testing of HCFC-22 alternatives under the Alternative Refrigerant Evaluation Program (AREP) and the results have been reported by Hickman (1993, 1994) and others. Individual companies have independently published results of tests in heat pumps or air conditioners for mixtures of particular interest to them (Geiger 1993, Kondepudi 1993, Zheng 1993). The potential efficiency improvements for some blends shown in Table 13 have not yet been achieved in actual equipment, because of the unoptimized design and losses

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[3] Test A: 35°C (95°F) ambient cooling, 10°C (50°F) evaporating, 46°C (115°F) condensing, 12.8°C (55°F) return gas, 40.6°C (105°F) liquid;
Test B: 27.8°C (82°F) ambient cooling, 8.6°C (47.5°F) evaporating, 40°C (104°F) condensing, 12.8°C (55°F) return gas, 39°C (102°F) liquid;
Test C: 8.3°C (47°F) ambient heating, -1.1°C (30°F) evaporating, 33°C (92°F) condensing, 2°C (36°F) return gas, 33°C (92°F) liquid;
Test D: -8.3°C (17°F) ambient heating, -13°C (9°F) evaporating, 29°C (85°F) condensing, -11°C (12°F) return gas, 28°C (83°F) liquid

Table 13. Theoretical steady-state COPs for unitary air conditioners and heat pumps.

Refrigerant Designation	Components	Operating Test Conditions			
		"A"	"B"	"C"	"D"
R-22	HCFC-22	100%	100%	100%	100%
R-134a	HFC-134a	100%	100%	99%	99%
R-717	Ammonia secondary hx 2.8°C ΔT secondary hx 5.6°C ΔT	102% 86% 73%	102% 84% 71%	102% 87% 75%	100% 88% 78%
R-290	R-290 (Propane)	99%	99%	99%	99%
R-410A	HFC-32/HFC-125 (50/50)	97%	98%	98%	98%
unassigned	HFC-32/HFC-134a (25/75)	107%	109%	106%	103%
unassigned	HFC-32/HFC-227ea (35/65)	131%	139%	126%	116%
R-407C	HFC-32/HFC-125/HFC-134a (23/25/52)	101%	103%	101%	99%
unassigned	HFC-32/HFC-125/R-290/HFC-134a (20/55/5/20)	95%	96%	97%	96%

associated with large temperature glides in the cross-flow, air-to-air heat exchangers.

Published measurements for steady-state COP and capacity data were used with an unpublished computer program by Pannock at NIST to calculate seasonal cooling efficiencies (SEERs) for the HFC-32/HFC-125, HFC-32/HFC-125/HFC-134a, and HFC-32/HFC-134a mixtures. The SEERs for mixtures based on "drop-in" test results are particularly conservative. A seasonal cooling efficiency was estimated for HFC-32/HFC-125 using measurements for a "soft optimized" air conditioner that is 5% higher than that of HCFC-22; measurements for drop-in testing with HFC-32/HFC-125/HFC-134a yield a seasonal COP only 95% that of HCFC-22, and data from drop-in testing with HFC-32/HFC-134a give a seasonal COP 98% that of HCFC-22.

Further development of air conditioners to use these mixtures could lead to more favorable comparisons relative to HCFC-22. Domanski and Didion showed that design changes for each new refrigerant could improve performance substantially from what would be measured, or expected, in drop-in testing where only the flow control device is adjusted for each alternative (1993). In particular, Rice, Wright, and Bansal showed that HFC-32/-HFC-134a and HFC-32/HFC-125/HFC-134a could benefit from counterflow, or cross-counterflow, heat exchangers and have efficiencies exceeding HCFC-22 (1993).

Commercial potential: while none of the HFC mixtures is a drop-in replacement for the refrigerant in unitary equipment, there is strong commercial interest in several of them. HFC

mixtures can be used almost as efficiently as HCFC-22, and with optimization of equipment design will probably surpass the efficiency of HCFC-22. Refrigerant leakage from hermetic unitary equipment is very small and future service losses will be low because changes in maintenance and replacement practices mandating refrigeant recovery and recycling are in place or under consideration in many countries.

Propane

Propane (R-290) can be an excellent refrigerant which is attracting attention as an alternative to HCFC-22. The major disadvantage with propane, naturally, is that it is flammable; this is a major concern in some countries (e.g. the U.S. and Japan) and less of a concern in some European and developing countries.

Theoretical performance: the theoretical efficiency of propane is also listed in Table 13; the COP at each condition used in the calculations is only slightly below the theoretical COP for HCFC-22 at the same conditions. It should be noted that these calculated COPs do not account for any losses in efficiency that would be imposed as a result of changing system design to accomodate a flammable refrigerant.

Actual performance: one evaluation of propane conducted by an equipment manufacturer reported a slightly better efficiency and capacity for a 9 kW (2½ ton) air conditioner compared to an HCFC-22 system (Treadwell 1994). A prospective design was developed for a 12 kW (3½ ton) air conditioner as a packaged system (as opposed to a split-system) where the entire mechanical system is outside the conditioned space (the flammable refrigerant never enters the building). Underwriter's Laboratory performed a construction review of this single-package unit.

Commercial potential: there is a strong reluctance on the part of manufacturers in the United States and elsewhere to accept flammable refrigerants in residential and commercial applications which will make it difficult for propane to gain wide acceptance in this application. Concerns are related to legal liabilities rather than technical obstacles. The cost of a 12 kw (3½ ton) unitary air conditioner using propane was estimated to be 30% higher than a comparable system using HCFC-22 (Treadwell 1994). The increased costs are due to modifications necessary to handle a flammable refrigerant:

- completely sealing refrigerant tubing,
- sealing or relocation of potential indoor-side ignition sources from the blower motor, wiring, and motor capacitor,
- possibly increasing the wall thickness of heat exchanger return bends,
- a pressure relief valve, and
- a propane leak detector.

These are extra costs not associated with using a non-flammable refrigerant. The market for unitary equipment is very price sensitive and this estimated cost premium would put air conditioners using propane at a great disadvantage in free markets.

Ammonia

Ammonia is an excellent refrigerant and is likely to experience broader application as CFCs and HCFCs are phased out, but it is not an automatic choice for unitary equipment. The chemical properties of ammonia present material compatibility problems that do not exist with some of the fluorocarbon and hydrocarbon alternatives. While there are cost-effective ways of solving these problems in large industrial refrigeration systems, this may not be possible on smaller residential and light commercial equipment.

Residential and light commercial air conditioning systems are mass produced using copper for refrigerant tubing, both in the heat exchangers and connecting components, hermetic compressors using electric motors with copper wire windings, and direct heat exchange evaporators. Ammonia is incompatible with copper in the presence of water, and consequently designs or materials need to be changed to eliminate the use of copper to ensure acceptable equipment lifetimes. Aluminum or steel are typically used for piping and tubing in industrial ammonia refrigeration systems with accompanying materials and installation cost increases. Efforts are underway to develop electric motors for use in ammonia systems but have not yet resulted in hermetic compressors for unitary applications.

Direct heat transfer evaporators are not considered feasible with ammonia in residential applications; there is a desire or need to keep the ammonia out of the conditioned space because of its toxicity, so a secondary heat transfer loop and fluid (e.g. brine, glycol) would be needed. This additional heat transfer loop increases the cost and complexity of the system and reduces the efficiency as shown in Table 13. Ammonia also has high discharge temperatures that need to be handled to avoid damaging the system; this can be done economically on large refrigeration systems but the cost could be prohibitive on small systems.

Theoretical performance: a theoretical comparison of ammonia and HCFC-22 is also presented in Table 13. Steady-state COPs are comparable to slightly higher for ammonia than for HCFC-22 using the same refrigerant conditions, but are significantly lower when accounting for efficiency losses resulting from either a 2.8°C (5°F) or 5.6°C (10°F) ΔT due to using a secondary heat transfer loop (necessary to keep the ammonia out of the conditioned space). The COPs for systems with secondary loops do not include energy used for circulating the brine or glycol solution in the secondary loop.

Actual performance: actual system performance data is not available for ammonia used in unitary equipment.

Commercial potential: although ammonia has excellent thermodynamic properties, it has not been competitive with fluorcarbons for small equipment. Developments in motors, heat exchange surfaces, etc. that are pursued for use in large ammonia refrigeration systems could eventually be of use in unitary sized equipment and could create commercial interest in using ammonia in this application in the long term.

GAX Absorption Heat Pump

Residential and light commercial sized gas-fired absorption heat pumps using ammonia-water are under development to provide heating or cooling. One product using the GAX cycle has reached the proof-of-concept stage under a project sponsored by the U.S. Department of Energy and a major American manufacturer is currently evaluating hardware (EPA 1993). The GAX cycle (see page 16) is being used to boost the efficiency compared to single-effect systems to be competitive with electric driven heat pumps. A small number of ammonia-water GAX heat pumps have been manufactured in the U.S. for testing and optimistically a product could enter the market in 1997; a more realistic date for market entry is 1998 (Marsala 1993).

Theoretical performance: the GAX "heating efficiency exceeds the efficiency of a gas furnace by 20% to 80%... Conservative target, steady-state efficiencies...are 1.4 COP heating (gas-fired) at 47°F (8.3°C) and 0.9 COP cooling (gas-fired) at the DOE/B (82°F) test condition. Parasitic power consumption targets for the outdoor package are in the range of 150 to 190 Watts per ton of refrigeration capacity (42 to 54 W/kW) (Marsala 1993)." Similar efficiency data are cited by Erickson and Rane (1992) with cycle modifications described that could boost the cooling COP to 1.5. The next generation of GAX equipment will push the limits of single-stage efficiency and will require the development of cost-effective simultaneous heat and mass transfer surfaces for the absorber and generator which are required to maintain the desired temperature and concentration profiles needed to achieve full performance from the sealed absorber system (Marsala 1993).

Actual performance: tests performed on the prototype GAX heat pumps have measured gas COPs of 1.4 heating (8.3°C, 47°F and 28°C, 82°F) and 0.75 cooling, including flue losses but excluding electric parasitics (DeVault 1994).

Commercial potential: there are strong prospects for successful commercialization of the GAX concept for heat pump applications if the remaining technical obstacles are overcome. Marsala enumerated the technical issues that must be resolved in bringing the GAX heat pump to market, these include:

- developing a solution pump that has a long life, low cost, and can operate without a lubricant in a harsh environment,
- eliminating the production of non-condensible gases in the refrigerant/absorbent loop,
- reducing parasitic electric power consumption,

- developing cost effective manufacturing techniques and quality control.

Adsorption Heat Pumps

Jones (1993), Shelton (1993), and Ryan (1993) each describe adsorption heat pumps that can be applied to residential space conditioning. Each employs two or more pairs of sorption beds with regenerative systems to provide high efficiency and continuous heating or cooling without large dips in capacity as the system cycles between beds (see description on page 12). Ryan describes a heat pump using ammonia as the sorbate (refrigerant) and a salt sorbent. The salt offers two advantages over other solids; 1) the salt adsorbs the ammonia in an inactive system reducing emissions of ammonia in the event of an accident, and 2) the salts used adsorb ammonia in three distinct layers, or coordination spheres, at three different temperatures. The coordination spheres offer a flexibility in designing a regeneration system to reduce losses and boost efficiency that is not available with sorbents that function only at a single temperature. Shelton describes a heat pump using ammonia and activated carbon designed for optimal efficiency at -1°C (30°F) (1993). This system also employs multiple pairs of sorption reactors and a regenerator to improve efficiency. The Gas Research Institute in the U.S. has been sponsoring development of adsorption heat pumps.

Theoretical performance: much of the data on adsorption heat pumps is from computer simulations and not from "cycle" analyses similar to what is shown elsewhere in this report for vapor compression refrigerants or from actual system testing. Jones has reported efficiencies for the adsorption cycle using modifications under development at the Jet Propulsion Laboratory in California which could lead to substantial improvements in both heating and cooling steady-state performance (1993). Data from computer simulations for the baseline JPL adsorption heat pump and for the two system modifications are shown in Table 15.

Actual performance: Ryan (1993) reports efficiency data for a chemisorption heat pump. These data are shown in Table 15.

Commercial potential: this gas-fired heat pump could be viable in climates where home and office energy use for space conditioning is dominated by the cold winter weather. It is at an earlier stage of development than the GAX heat pump and affected by development issues

Table 14. Efficiency for simulated performance of enhanced adsorption heat pump.

Ambient Temperature		Steady-State COP (excluding parasitics)		
(°C)	(°F)	Standard System	Modification #1	Modification #2
35	95	1.0	1.3	1.7
27.8	82	1.3	1.6	2.1
8.3	47	1.9	2.1	2.4
-8.3	17	1.5	1.6	1.8

independent of other gas-fired technologies (e.g. "aging" of sorbent beds from frequent cycling).

Table 15. Performance data for chemisorption heat pump.

System Performance	Chemisorption Heat Pump
Cooling Mode	
COP	1.0
Equivalent SEER	3.4
Parasitic Power	300 W / ton of cooling capacity
Heating Mode	
COP	1.5
Parasitic Power	320 W / 20,000 Btu heating
Heating Capacity Required	20,000 Btu / ton of cooling capacity

Evaporative Coolers

Theoretical performance: a theoretical efficiency is not presented for evaporative coolers because the performance is so dependent on climate conditions.

Actual performance: Watt reported that the steady-state energy use of a residential size, direct evaporative cooling system will be only 20% that of a vapor compression air conditioning system with an energy efficiency ratio (EER) of 9.0 (1988). Foster reported that for four cities in the western U.S., energy savings with evaporative cooling could be 69 to 74% compared to vapor compression air conditioning. He also reported that "indirect systems with vapor compression second stages can provide adequate comfort in high humidity zones with a savings of 20 to 25% (1990)." The combined evaporative/vapor compression design to realize these savings requires two mechanical systems, higher first costs, and is consequently precludes residential applications.

The degree of data available does not permit the calculation of TEWI for residential evaporative air conditioning systems. Data on steady-state performance suggest that the comparison would be very similar to that of the commercial systems in Table 8 on page 106.

Commercial potential: evaporative coolers are an established product in suitable climates and compete with vapor compression systems for market share in those areas. It is unlikely that evaporative cooling could replace vapor compression equipment entirely in more humid climates where the vast majority of people live and work.

TEWI Calculations

Central Air Conditioning (Cooling Only)

Total equivalent warming impacts were calculated for a baseline 10.5 kW (36,000 Btu/h) baseline air conditioner with an SEER of 10 and a refrigerant charge of 4.45 kg (9.8 lbs) of

HCFC-22 in three locations in the U.S. Seasonal energy use is computed based on 450 operating hours per year for the air conditioner in Pittsburgh, Pennsylvania; 950 hours of operation in Atlanta, Georgia, and 2315 hours per year in Miami, Florida (Ballou 1981). Electricity consumption is converted to CO_2 emissions using the North American average of 0.650 kg CO_2/-kWh of electricity. It is assumed that 4% of the charge is lost annually through leaks and service and that 5% of the charge is lost when the system is scrapped after 20 years of use.

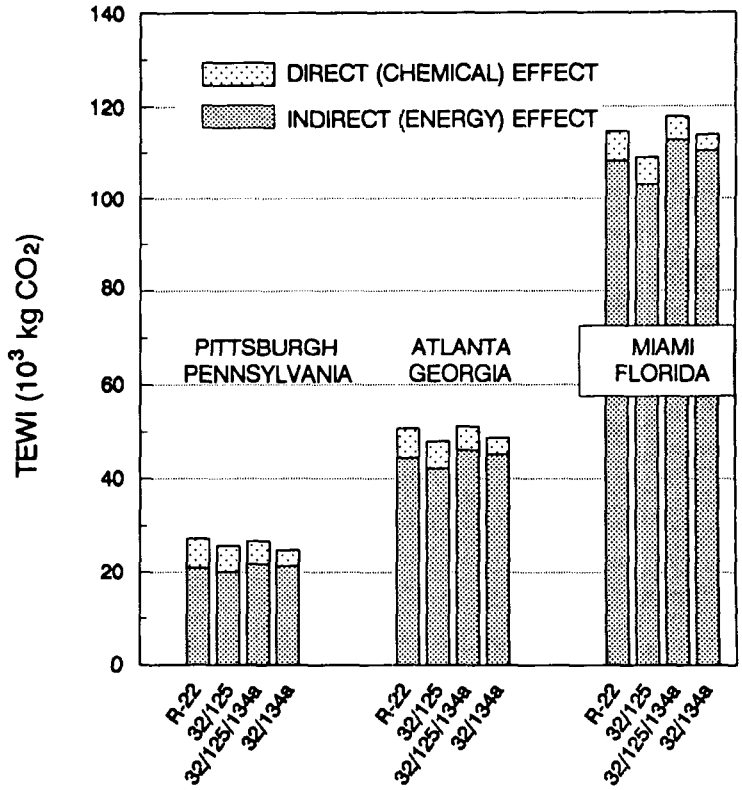


Fig. 34. TEWI for central air conditioners (SCOP=2.93, SEER=10).

The TEWIs are shown in Fig. 34 where the heavily shaded region represents the contribution from energy consumption and the lightly shaded portion is from refrigerant losses using the 100 year GWP value. TEWI were also calculated for three different mixtures of HFCs using the calculated relative seasonal efficiencies listed on page 133. Refrigerant charges are derived from the assumed value for the HCFC-22 baseline scaled by the ratio of the liquid refrigerant densities at 21°C (70°F). This procedure closely matched the optimum charge determined in one system test (Geiger 1993).

There are bars in Fig. 34 for the TEWI of the HFC mixtures as well as for HCFC-22. None of the systems represented were actually designed for optimal performance with a mixture and consequently the energy efficiencies reflect very early test results. In each case, the direct contribution of refrigerant emissions, using the 100 year GWPs represent as much as 20% of the TEWI in Pittsburgh where there are few cooling hours and as little as 3% in Miami where there is a large cooling load. Nearly 90% of the direct effect is due to the assumption on annual emissions from leakage, accidents, and maintenance practices; as regulatory procedures requiring conscientious maintenance and repairs of leaks and strict adherence to refrigerant recovery are adopted and followed, the direct effect will diminish in significance.

The efficiency of the HCFC-22 air conditioner is an important factor in the calculated TEWI. The bars shown in Fig. 35 are similar to those in Fig. 34 except for high efficiency air conditioners (baseline HCFC-22 SCOP=4.1, SEER=14). The TEWI are 22 to 28%

below the corresponding values shown in Fig. 34 because of the reductions in energy use.

TEWI are not computed for air conditioners using evaporative cooling, absorption, or adsorption cycles.

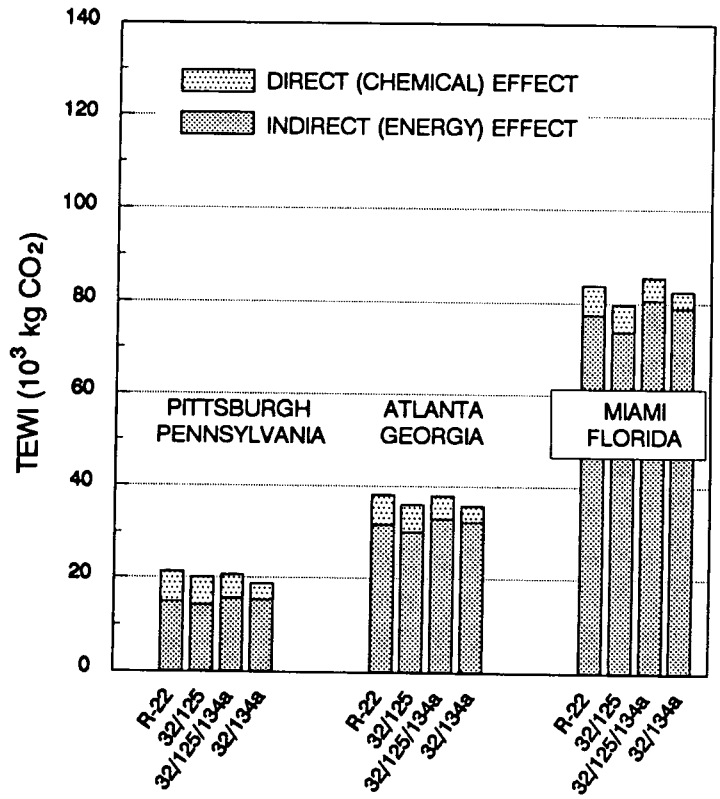


Fig. 35. TEWI for a high efficiency (SCOP=4.1, SEER=14) central air conditioner.

Residential Heat Pumps

The efficiency data for an adsorption heat pump in Table 15 were used to calculate gas and electricity consumption for (36,000 Btu/h) heat pumps in Pittsburgh, Atlanta, and Miami for comparison with an electric heat pump. Emissions of CO_2 from gas combustion are calculated on the basis of 53.9 g CO_2 per 1000 Btu of fuel input. The annual heating and cooling loads computed for an 1800 ft^3 home in Pittsburgh are 74.7 x 10^6 Btu and 16.1 x 10^6 Btu, respectively. The corresponding loads for Atlanta are 34.8 x 10^6 Btu heating and 33.8 x 10^6 Btu cooling (Ballou 1981). The loads for Miami are 0 Btu/y heating and 82.2 x 10^6 Btu/y cooling. The TEWI of an all electric system is calculated using performance data for a state-of-the-art continuously variable-speed electric heat pump using HCFC-22 . The electric heat pump has a heating seasonal performance factor (SPF) of 2.86 (HSPF=9.75) and a cooling SPF of 4.69 (SEER=16) (Rice 1994). Results are based on the North American average of 0.650 kg CO_2/kWh.

The TEWI for the adsorption heat pump and a variable-speed electric heat pump are shown in Fig. 36. The heavily shaded portion of each bar corresponds to CO_2 emissions during the heating season, the next segment of each bar corresponds to the cooling season, and the lightly shaded portion at the top of the electric heat pump bars represents the direct effects of refrigerant emissions. The effect of climate is dramatically illustrated with the differences between the TEWI for the three cities. The TEWI for the adsorption heat pump in Pittsburgh is much lower than that for the variable-speed electric heat pump.

The CO_2 emissions from the heating season are much smaller than those for the electric system and the differences in the cooling season emissions is not sufficient to offset those from heating. The TEWI for Atlanta are about the same, considering that the data for the adsorption system are laboratory measurements on a prototype heat pump. The adsorption

system has lower CO_2 emissions in heating that are nearly offset by the advantage the electric system maintains in cooling. In Miami the TEWI for the electric heat pump is much lower than that for the adsorption system, in spite of the direct effect of refrigerant emissions. Although these calculations are performed using HCFC-22 as the refrigerant, similar results would be expected for any of the mixtures considered on page 139 for unitary systems.

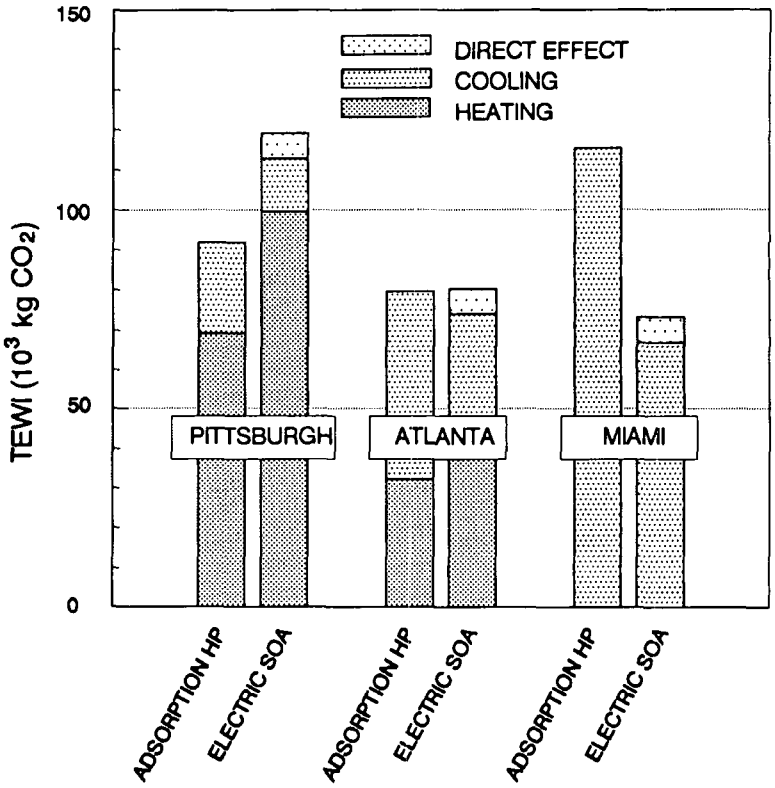


Fig. 36. TEWI for adsorption and variable-speed electric heat pumps.

Although TEWI are not calculated for a GAX absorption heat pump, the comparison between the GAX, HCFC-22, and HFC alternatives (by inference) would be very similar to the comparison in Fig. 36 for adsorption and HCFC-22 heat pumps. Both gas-fired systems have high heating efficiencies and lower TEWI in heating climates; both have higher overall emissions in predominantly cooling climates.

CONCLUSIONS

The direct effects of the TEWI for all four vapor compression systems presented are important, but probably manageable, fractions of the total in each case. These contributions should not be ignored, although rigorous procedures for handling refrigerants and accounting for refrigerant usage should be effective at reducing the direct effect from those shown in Fig. 34. Contributions to global warming from energy usage are either better or not significantly worse than those of current technology air conditioners; engineering optimization is likely to reduce CO_2 emissions with the mixtures from what is represented in Fig. 34.

In climates with a small cooling load and high heating load, the gas-fired heat pumps have a significantly smaller TEWI than electric heat pumps. This advantage diminishes and disappears as the balance shifts to lower heating and higher cooling loads. The comparisons presented in Fig. 36 are based on laboratory measurements for a prototype adsorption heat

pump and ratings data for a commercially available variable-speed electric heat pump. Keeping that in mind, the differences in TEWI for Pittsburgh and Miami are significant, but the difference shown for Atlanta are small enough to be insignificant. The adsorption cycle enhancements mentioned on page 137 could be effective in reducing TEWI further, but the long term performance of the basic or enhanced systems must be proven through field testing to establish the viability of the technology.

The GAX absorption heat pump could also be effective at reducing TEWI in heating-dominated climates. This technology has undergone extensive development and is close to commercialization. Both gas-fired alternatives could have comparable TEWI to conventional electric driven compression systems in climates with balanced heating and cooling loads, and have significantly higher TEWI in cooling-dominated climates.

BUILDING INSULATION

Introduction

Plastic foam insulation is frequently used in residential and commercial building construction either as a portion, or as the principal, insulating material in walls and roofs. The 1991 study examined a wide range of NIK insulation materials and alternative blowing agents for polyurethane, polyisocyanurate, and extruded polystyrene insulations in wood frame construction in North America and masonry and wood frame construction in Europe. Commercial building construction was looked at for both wall and roof insulation using a "composite" building in North America and Europe that reflected building usage for stores, offices, and warehouses and both masonry and metal panel construction.

Extensive effort by the insulation producers and their suppliers has been put into developing alternative materials that meet the needs that had been filled by CFC-11 and CFC-12 blown foam insulation. The most successful efforts to date have been evolutionary steps in identifying blowing agents that pose no, or at least a significantly smaller, threat to stratospheric ozone, optimizing the formulations of foams using these gases to obtain the necessary thermal and structural properties, and adapting equipment and manufacturing processes to use these blowing agents. These second generation blowing agents include HCFC-141b, cyclopentane, HFC-134a, and, HCFC-22/HCFC-142b; some of the best second generation blowing agents have small, but non-zero ODPs.

The polyurethane and polyisocyanurate industries are currently working on a number of third generation blowing agents with zero ODP and a minimum or no effect on global climate. Research is being carried out by raw material suppliers to the industry—blowing agents as well as other foam components—and by foam producers in various industries such as construction and appliance. An international workshop organized by PIMA was conducted in July 1994 bringing together a number of experts to discuss the needs of the foam insulation industry and to present an update on potential candidates or classes of blowing agents.

In 1994 it is too early to quantify the potential environmental benefits of these third generation blowing agents because the initial evaluations have shown that foam formulations must be optimized to accommodate zero ODP blowing agents. The thermal insulation properties of these new foams may still undergo considerable improvements before actual production would begin. Potential candidates under investigation cannot be considered as "drop-in" materials for current polyurethane or polyisocyanurate formulations and require further technological and chemical developments. Consequently, the status of chlorine-free

blowing agents is in a high state of flux, and the discussion which follows pertains almost exclusively to the second generation blowing agents.

Thermal Conductivities

The 1991 study included an effort to estimate average thermal conductivities of foam insulation over the lifetime of the building. Data on k-factors from thin-slice testing were used to project aging characteristics and 50 year time averaged k-factors for polyurethane foams blown with CFC-11, HCFC-123, HCFC-141b, and isopropylchloride and extruded polystyrene insulation using CFC-12 (Fischer 1991).

The polyurethane and polyisocyanurate industries have gravitated toward the use of HCFC-141b as an interim replacement for CFC-11; HCFC-123 and other blowing agents that were considered in 1991 have been rejected for one reason or another. At that time, the thermal conductivity of laminated boardstock blown with HCFC-141b was 13% higher than the thermal conductivity of boardstock blown with CFC-11. Optimization of formulations has led to the development of foams using HCFC-141b which have fresh foam k-factors only 5% to 8% higher than CFC-11 blown foam (Fishback 1992, Willoughby 1992, Yamamoto 1992, Thrun 1992). Only preliminary or incomplete data are available on the aging characteristics of these foams; they are referred to as aging no worse than CFC-11 blown foams.

Interest in hydrocarbon blowing agents is growing, those attracting the greatest interest in 1993 are cyclopentane and n-pentane. Cyclopentane has a lower vapor phase thermal conductivity than n-pentane and foams using cyclopentane will have lower k-factors than foams using n-pentane. Cyclopentane costs approximately three times as much as n-pentane (Heilig 1993a), however, and the price sensitive market for building materials will probably favor the use of n-pentane. Early, unoptimized formulations of boardstock using n-pentane have k-factors 10% to 20% higher than CFC-11 blown foam (Heilig 1993). Heilig reported on aging tests of pentane blown foam and states "the aging behavior...is excellent and at least equivalent to standard CFC-11 foams (1993)."

The extruded polystyrene insulation industry moved aggressively starting in 1989 to eliminate CFCs in their products and much, if not all, of the production has shifted to using HCFC-142b or HCFC-142b with HCFC-22 to achieve the desired density (Abbott 1993). The industry has not identified acceptable chlorine-free blowing agents, although HFC-134a, HFC-152a, hydrocarbons, alcohols, and carbon dioxide have all been considered (Abbott 1993). Cost has been a factor in eliminating some potential blowing agents but the choice is also constrained by the manufacturing process, permeability of the gases in the foam, solubility in the polymers, and flammability. Boardstock has been produced using HCFC-142b that has equivalent thermal characteristics to the CFC-12 blown foam considered in the 1991 project (Dow 1994).

The thermal conductivity of foam insulations are known to vary with the thickness of the insulation as well as with age because of the longer mean path for permeation of air into the foam and the eventual escape of the blowing agent. Fifty and 15 year time averaged k-factors for polyurethane and extruded polystyrene insulation are presented in Fig. 37 and Fig. 38 as they vary with the thickness of the foam; the 50-year values in Fig. 37 are for laminated boardstock for wall insulation and the 15 year data in Fig. 38 are for boardstock for roofing insulation (either unfaced or with a gas-permeable facing material). The curve for HCFC-141b blown foam is an assumed 6% above the CFC-11 data developed as part of the 1991 study and the n-pentane blown foam is 15% higher than CFC-11 which is consistent with reported measurements on fresh foam thermal conductivity and general aging behavior.

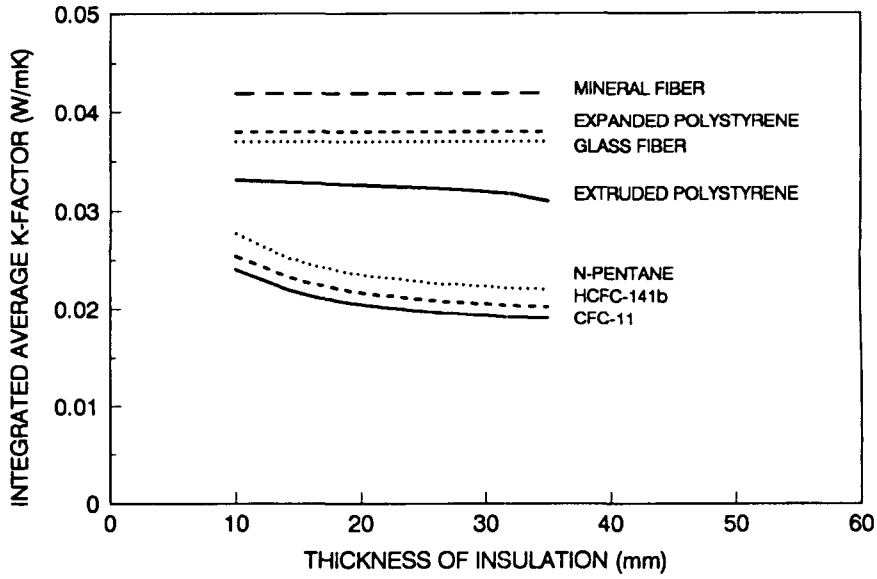


Fig. 37. Thermal conductivity for wall sheathing materials.

CALCULATION OF TEWI

Two homes are considered in calculating the global warming impacts of alternative building insulations. The first is a single-family dwelling in a multi-unit building in the U.K. and the second is a detached single family dwelling in the U.S.

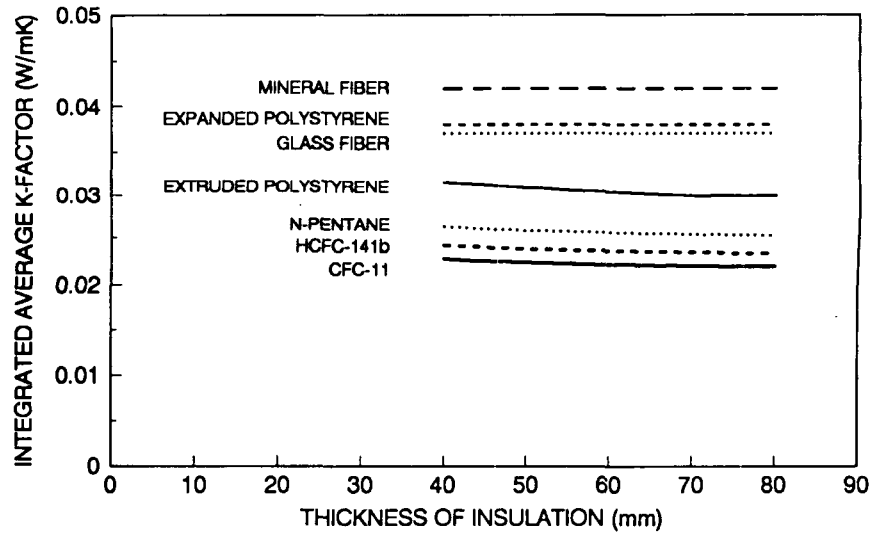


Fig. 38. Thermal conductivities for roof insulation.

European Home

The home in England is assumed to be of masonry construction as shown in Fig. 39 using insulation between a brick facade and the block wall of the building with gypsum wallboard on the inside. It is assumed to have a hot water heating system using a 65% efficient natural

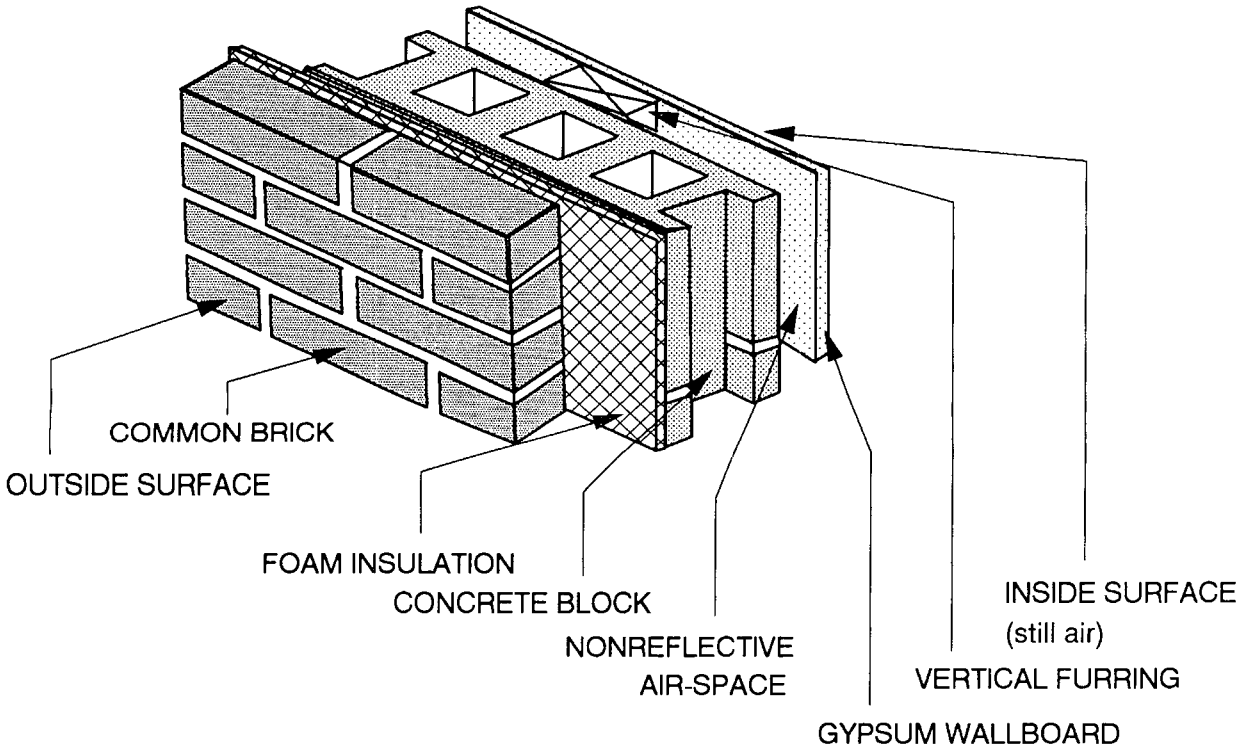


Fig. 39. Masonry wall construction for residential building in the U.K.

gas fired boiler. It is also assumed to be the end unit of a multi-unit row house so there are three exterior walls. It has 130 m^2 (1400 ft^2) of floor space. Weather data from Mildenhall Royal Air Station is used to calculate the building heating loads. The alternative insulations considered are 28 mm (1.10 in.) of polyurethane using CFC-11, HCFC-141b, or n-pentane as the blowing agent or 43 mm (1.69 in.) of extruded polystyrene with either CFC-12 or HCFC-142b. The CFC-11 polyurethane has a k-factor of 19.4 mW/mK (0.135 Btu·in./h·ft^2·°F) and the thermal conductivities of the foams using HCFC-141b and n-pentane are 6% and 15% higher. The CFC-12 and HCFC-142b extruded polystyrenes have k-factors of 29.8 mW/mK (0.207 Btu·in./h·ft^2·°F).

The annual heating load of this home is computed to be 6100 MJ (5.79×10^6 Btu) for either the CFC-11 or CFC-12 baselines or the HCFC-142b blown extruded polystyrene. The HCFC-141b and n-pentane alternatives are estimated to have heating loads of 6345 and

6670 MJ (6.02×10^6 and 6.33×10^6 Btu), respectively. It is assumed that only 30% of the heating loads are due to losses through the insulated wall surfaces; this is consistent with the assumptions used in the 1991 study. Applying the assumed boiler efficiency and 51.1 g of CO_2/MJ of energy consumed results in annual CO_2 emissions of 144 kg for CFC-11, CFC-12, or HCFC-142b insulations (at the appropriate thicknesses) and 150 kg for HCFC-141b and 157 kg for n-pentane blown polyurethane insulation (at the same thickness as selected for CFC-11 polyurethane).

Table 16. TEWI for masonry construction in Great Britain.

	CFC-11	HCFC-141b	n-pentane	CFC-12	HCFC-142b
Heating Load (MJ)	6112	6345	6670	6112	6112
Boiler Efficiency	0.65	0.65	0.65	0.65	0.65
Energy Input (MJ)	9400	9800	10,400	9400	9400
Fraction Losses through walls	0.30	0.30	0.30	0.30	0.30
Energy Loss through Walls	2800	2900	3100	2800	2800
Heating Value (kJ/m^3)	38,200	38,200	38,200	38,200	38,200
Natural Gas Consumption (m^3/y)	74	76.6	80.6	74	74
CO_2 Emission (g/MJ)	51.1	51.1	51.1	51.1	51.1
CO_2 Emissions (kg/y)	144	150	157	144	144
Lifetime (y)	50	50	50	50	50
CO_2 Lifetime (kg)	7200	7500	7900	7200	7200
Direct Effect (kg CO_2)	30,500	3400	0	42,300	9300
TEWI	37,700	10,900	7900	49,500	16,500

It is estimated that 77.5 m^2 (835 ft^2) of insulation are needed to fill the gap between the brick facade and block walls. Using the insulation thicknesses stated earlier with the appropriate densities and weight percent blowing agent for each gas results in an estimated 8.7 kg of CFC-11, 7.6 kg of HCFC-141b, 4.2 kg of n-pentane, 5.8 kg of CFC-12, or 5.8 kg of HCFC-142b contained in the foam. The direct effects from these masses of blowing agents are combined with the estimated lifetime emissions of CO_2 for each dwelling in Table 16.

North American Home

The home in North America is assumed to be of wood frame construction with fiberglass

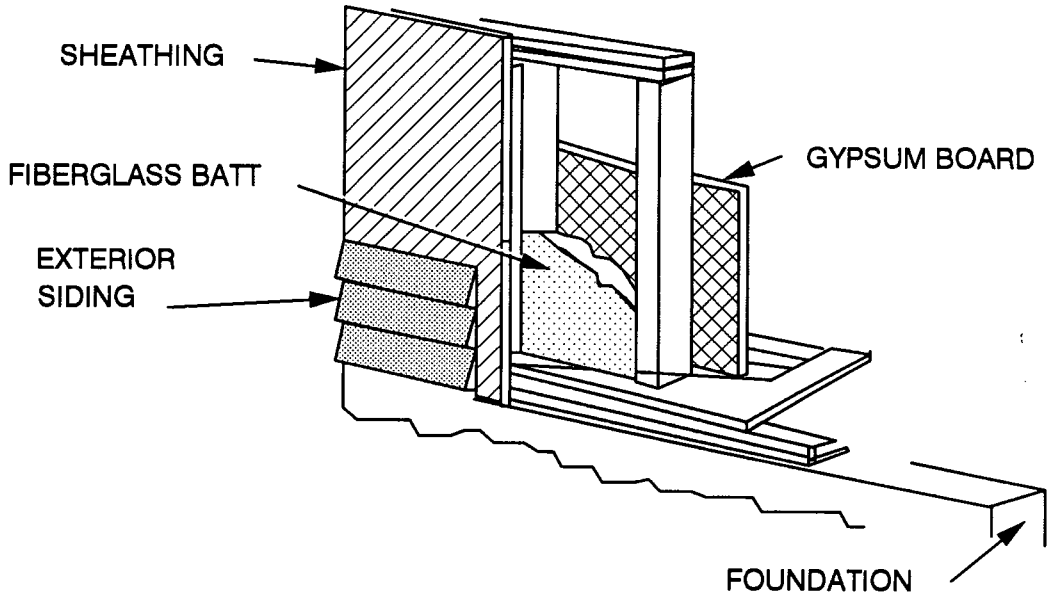


Fig. 40. Wood frame construction for residential building in the U.S.

insulation batts filling the wall cavity. Typical construction is shown in Fig. 40. The house is detached from other buildings and is assumed to have 186 m^2 (2000 ft^2) of living space. It has an 85% efficient forced air gas furnace. Weather data for Chicago, Illinois, U.S. are used for computing the heating load. The baseline house is assumed to use 21 mm (0.83 in.) of polyurethane wall sheathing with a thermal conductivity of 20.5 mW/mK (0.142 Btu-in./h·ft^2·°F) or 36 mm (1.44 in.) of extruded polystyrene with a k-factor of 35 mW/mK (0.247 Btu-in./h·ft^2·°F). The alternatives considered are the same thicknesses of plastic sheathing material assumed to be blown with:

- HCFC-141b with a 6% increase in k-factor,
- n-pentane with a 15% increase in k-factor, or
- HCFC-142b with the same k-factor as CFC-12 blown extruded polystyrene.

It is estimated that 3.38 m^3 (119 ft^3) of polyurethane or 5.87 m^3 (207 ft^3) of extruded polystyrene foam are required to insulate this building. Accounting for differences in density between polyurethane and extruded polystyrene, and differences in the weight percents of blowing agents (including blowing efficiencies), results in an estimated 13.5 kg of CFC-11, 11.5 kg of HCFC-141b, 6.5 kg of n-pentane, 10.2 kg of CFC-12, and 8.5 kg of HCFC-142b contained in the foam insulation.

Table 17. TEWI for wood frame home in Chicago with 85% gas furnace.

	CFC-11	HCFC-141b	n-pentane	CFC-12	HCFC-142b
Heating Load (MJ/y)	29,600	30,200	31,000	29,600	29,600
Furnace Efficiency	85%	85%	85%	85%	85%
Energy Input (MJ/y)	34,900	35,600	36,500	34,900	34,900
Fraction Through Walls	0.30	0.30	0.30	0.30	0.30
Losses through walls (MJ/y)	10,500	10,700	11,000	10,500	10,500
Natural Gas (m^3/y)	274	279	286	274	274
CO_2 Emissions (kg/y)	535	545	560	535	535
Lifetime (y)	50	50	50	50	50
CO_2 Emissions (lifetime)	26,800	27,200	28,000	26,500	26,800
Direct Effect (kg CO_2)	47,200	5,100	0	74,500	13,600
TEWI (kg CO_2)	74,000	32,300	28,000	101,000	40,400

The baseline home heating load using either CFC-11 or CFC-12 blown foam (in the appropriate thickness) is estimated to be 29,650 MJ/year (28.1×10^6 Btu/y), of which 30%, or 8,900 MJ/y (8.43×10^6 Btu/y) is estimated to be lost through the insulated wall surfaces with the balance due to losses through the roof and foundation, windows, air infiltration, etc. The corresponding wall thermal losses for HCFC-141b, n-pentane, and HCFC-142b sheathing materials are estimated to be 9075 MJ/y (8.61×10^6 Btu/y), 9315 MJ/y (8.83×10^6 Btu/y), and 8900 MJ/y (8.43×10^6 Btu/y). Applying the furnace efficiency and an assumed 51.1 g CO_2/MJ of energy consumed yields annual CO_2 emissions of 535 kg for CFC-11, CFC-12, and HCFC-142b blown polyurethane or extruded polystyrene with slightly higher values of 545 kg/y for HCFC-141b and 560 kg/y for n-pentane blown polyurethane. Table 17 shows the TEWIs for this home and the alternative insulations using the derived CO_2 emission rates, the assumed 50 year lifetime for the home, and the direct effects of the blowing agent in the foams.

CONCLUSIONS

Several conclusions can be drawn from the TEWI in Table 16 and Table 17:

- the use of HCFC-141b in place of CFC-11 reduces TEWI by 50 to 70%, depending on the home, but the direct effect will be between 16 and 30% of the total using 100 year GWP values,

- the use of n-pentane blown polyurethane reduces TEWI 14% to 28% below that of HCFC-141b even though its use requires about 5% more energy per year, and

- HCFC-142b extruded polystyrene has TEWIs that are only 33% to 40% those of CFC-12 blown extruded polystyrene, although the direct effect remains a significant fraction of the total.

Effort continues among foam producers and their suppliers to optimize the formulations used with each of the blowing agents to improve overall product performance as well as to reduce the k-factors as much as possible. These conclusions are based on applying equal thicknesses of similar materials, which is a choice made only for the purposes of comparing the alternative blowing agents. In practice the building materials used will be selected based on cost and product availability and the thickness will be determined by local building codes and practices or the architect's specifications.

Building insulation is one application where the differences between alternatives could result in a policy dilemma. At the thermal properties currently achievable, the choice of polyurethanes with the lowest TEWI leads to higher CO_2 emissions and that with the lowest CO_2 emissions has a higher TEWI. No attempt has been made in this project to resolve the competing goals of reducing overall greenhouse gas emissions at the same time as reducing CO_2 emissions. The optimization of foam formulations using third generation blowing agents could resolve the dilemma by resulting in insulation products with thermal properties better than the CFC and HCFC foams while using blowing agents with zero ODP.

PART 4: SENSITIVITY TO ASSUMPTIONS

SENSITIVITY TO ASSUMPTIONS

Introduction

The TEWI (total equivalent warming impact) concept was developed to provide a simple indicator of global warming impact of CFC alternative technologies that considered both direct (chemical) emissions and indirect (energy) emissions over the end-use service life. As an example, in chiller applications there may be direct chemical emissions as a result of loss of refrigerant through leakage and service as well as indirect emissions associated with the energy required to operate the chiller over its service life. As with any other simple indicator, calculated values of TEWI are sensitive to the underlying assumptions.

This section explores the sensitivity of TEWI to its underlying assumptions. Of all of the CFC alternative end-uses treated in the original AFEAS/DOE study (Fischer 1991), TEWI has received the greatest attention in the context of the transition away from CFC-based chillers. For this reason the water-cooled chiller application was selected for this sensitivity analysis. This is an interesting case because NIK (not-in-kind) CFC alternative technologies are already commercially available and they are primarily fueled by a different energy form (natural gas). With the entire world-wide installed base of chillers at stake, eloquent spokesmen for various alternative fluorocarbon and NIK refrigerants, electric or gas engine chillers based on those refrigerants, and NIK chillers have been participating in a lively debate. The sensitivity analysis provides objective information that may result in more appropriate future use of TEWI in this debate.

Sensitivities of TEWI to refrigerant charge loss rate, CO_2 emissions associated with electric power generation, electric chiller seasonal efficiency, and heating/cooling load ratio are investigated. The relative importance of global warming (and TEWI) as an issue in chiller selection is also put in perspective.

Background

The initial AFEAS/DOE study addressed the energy and global warming impacts of the CFC alternative technologies being considered in the seven major CFC end-uses world-wide. It focused on CFC alternative technologies that were actively being pursued by the affected end-user industries at the time. In the case of chillers, alternative fluorocarbon refrigerants were primarily evaluated, with only preliminary attention given to NIK or next generation technologies. Absorption chillers were given brief consideration, but using an analysis

framework best suited for the primary emphasis of the study, i.e., comparing alternative refrigerants in vapor compression machines. Ammonia vapor compression chillers were also briefly considered but hard data were unavailable, a situation that continues to this day. Engine-driven vapor compression machines were not considered at all.

Comparisons of absorption and vapor compression chillers were made at equal chiller manufacturing cost in the original AFEAS/DOE study because these were the only data obtainable from industry. While the approach of comparing performance for equally priced equipment was perfectly valid for the focus of the study, i.e., comparing different versions of equipment within the same equipment class (e.g., CFC-11 vs. HCFC-123 centrifugal chillers), it is admittedly less valid when making comparisons across equipment classes.

For example it is a well known fact that the unit equipment cost of absorption chillers is higher than that of electric vapor compression chillers. However, this premium is supported in certain applications due to savings to the building owner in other areas such as elimination of the need for a separate boiler; the use of smaller transformers, switchgear, and wiring within the building; utility bill savings; and rebates or financing from electric utilities benefiting from peak load shaving and/or gas utilities benefiting from valley filling load additions (AGCC 1994). The actual cost premium per unit capacity is also diminished in certain applications by installing a mix of absorption and electric chillers that achieves most of the potential bill savings of an all-gas chiller plant at a fraction of the cost (Byars 1992). In general, building owners should make their space conditioning decisions based on all of the capital costs and all of the operating costs. Chiller cost and performance is an important part of the cost equation, but not the only factor.

The original study attempted to include some of these "real world" effects while staying within the constraints of the analysis framework. The effort received mixed reviews from several directions. While one chiller manufacturer termed it "a labored effort to say something helpful about absorption systems" (Hickman 1991), the natural gas industry's reaction was that the analysis was "flawed" (Baly 1992) and that the same-chiller-cost electric vapor compression chiller efficiencies used to compare with absorption were a "thermodynamic dream" (Dolan 1993; 0.39 kW/RT for an HCFC-22 electric chiller in the 1000 RT water-cooled category).

The sensitivity analyses presented here attempt to address these reviews, and improve the context for use of TEWI in the chiller selection debate, and also place in perspective the relative importance of TEWI in that debate. Although specific to chillers, the analysis provides an indication of how TEWI for any application might vary in relation to the underlying assumptions.

Analysis Framework

The analysis algorithms are described in detail in the original AFEAS/DOE study. Only changes from the original approach are discussed here. The analysis focuses exclusively on North America where about two thirds of the world's installed chiller capacity resides. However, the sensitivity analysis does provide insight into the chiller selection issue in other parts of the world. Major changes from the original approach that are used in this sensitivity analysis are described below:

- The analyses focus on water-cooled machines because this configuration dominates the installed capacity base. It no longer distinguishes between chiller size ranges because the 1050 kW (300 ton) and 3500 kW (1000 ton) categories used in the previous study are representative of most of the installed capacity base, and the CFC alternatives are the same for both.

- A more refined consideration of absorption chillers, engine-driven vapor compression machines, and ammonia vapor compression chillers is made than in the original study, although hard data for the latter still remain largely unavailable.

- Recognizing that buildings have other energy needs, such as space heating and water heating, the analyses consider total building TEWI by including a range of ratios that account for differing heating and cooling loads in large office buildings.

- Direct (chemical) emissions are estimated the same as in the original study except that the GWP integration time horizon is 100 rather than 500 years. The indirect (energy) emissions are calculated differently, as explained in the sections immediately following.

Values used for analyses of refrigerant charge loss, CO_2 emissions associated with electric power generation, electric chiller seasonal efficiency, and heating/cooling load ratio span ranges illustrative of "best available" (low) to "worst case" (high) performance. A chiller load factor of 0.20 was used in all analyses. The ranges of variable values used in the sensitivity analyses are presented in Table 18.

Refrigerant Charge Make-Up Rate

The sensitivity analysis range selected for this parameter is 0% to 16% of the charge per year. The low value corresponds to zero losses from system purge operation, equipment servicing, and end-of-life equipment retirement. The high value represents a loss rate that is slightly above the legal limit in the United States, and approximates both past U.S. performance and what may exist in developing countries. This range is identical to the range used in the original AFEAS/DOE study. A make-up rate of 0.75% per year is possible for new equipment (Hourahan 1994); purge losses and leakage for new equipment can be as

Table 18. Ranges of values for sensitivity analysis variables.

Variable Analyzed	Best Value	Value 2	Value 3	Value 4	Worst Value	Representative Value
Refrigerant Make-Up Rate (% of charge/year)	0%	4%	8%	12%	16%	0.75% new equipment 4% existing equipment
CO_2 Emission for Power Generation (kg CO_2/kWh)	0.4	0.6	0.8	1.0	1.2	0.51 Europe 0.58 Japan 0.65 North America
Electric Chiller Rated (COP) Efficiency	6.34	5.67	4.69	4.14	3.70	6.34 new equipment 5.67 average installed equipment
(kW/ton)	0.55	0.65	0.75	0.85	0.95	0.52 new equipment 0.65 average installed equipment
Building Heating/Cooling Load Ratio	0.0	0.1	0.2	0.3	0.4	unknown

low as 0.5% per year (Smithart 1993), while purge, leakage, and service losses for installed equipment can be 15% or higher (Whalen 1994).

CO_2 Emissions from Electric Generation

The sensitivity analysis range for this parameter is 0.4 to 1.2 kg CO_2/kWh. Regional averages for Europe, Japan, and North America are 0.51, 0.58, and 0.65 kg CO_2/kWh. The low value represents the emissions of a gas combined cycle power plant operating continuously at full load at an ambient temperature of 15°C (60°F). The high value is equivalent to an older coal steam plant in load-following duty at 35°C (95°F). Both values assume electric transmission and distribution (T&D) losses of 8%.

Electric Chiller Seasonal Efficiency

The sensitivity analysis range selected for COPs is 3.70 to 6.39 (0.55 to 0.95 kW/ton). An average for chillers installed in the U.S. is approximately 5.41 (0.65 kW/ton) while the most efficient chillers available in 1993 was 6.39 (0.55 kW/ton); chillers with rated COPs of 6.76 (0.52 kW/ton) became available in 1994. The low efficiency value (3.70 or 0.95 kW/ton) corresponds to monitored performance of a sample of 21 centrifugal chillers in ten buildings of various types (offices, hotels, schools, retail) that were monitored by the San Diego Gas & Electric Company (SDGE) in 1993. Since efficiency differences attributable to refrigerants over this sensitivity range are minor, seasonal efficiencies are assumed to be independent of fluorocarbon refrigerant.

Theoretically, ammonia can have an efficiency advantage, but on a same-cost basis this advantage may disappear because easily worked copper tubes must be replaced with mild steel in the heat exchangers. This analysis assumes that ammonia and HCFC-123 operate at approximately the same efficiency at the same costs. Since ammonia same-cost performance uncertainty is greater than the direct (chemical) emissions difference between ammonia (GWP=0) and HCFC-123 (GWP=93), ammonia (R-717) and HCFC-123 are shown together on the sensitivity graphs.

Building Heating to Cooling Load Ratio

The sensitivity analysis range for the heating to cooling load ratio is 0.0 to 0.4. The low value corresponds to zero heating load where: (a) total building TEWI equals chiller TEWI; or (b) heating loads exist and gas is used for heating, whether or not gas is used for cooling. The high value corresponds to a large office building in the midwestern United States that has relatively high heating needs (MacDonald 1991). The assumed relationships between gCOPs for gas-fired equipment and building heating loads is explained in the section on "Gas Chiller Performance".

The original AFEAS/DOE study and its sensitivity analysis focused on chiller TEWI. In addition to cooling, buildings have other energy service needs such as space heating and water heating. Many commercial and institutional buildings such as large offices have modest heating requirements and insignificant water heating requirements. These loads, traditionally served by natural gas where it is available, may not be large enough to justify gas grid line extensions in cases where the chiller plant is all electric. In such cases, heating requirements are typically met with electric resistance heat integrated into air delivery terminal units or configured as separate perimeter systems.

Building TEWI is defined as the sum of chiller TEWI and the CO_2 associated with meeting the space heating load of the building over the life of the chiller. This sensitivity analsysis address total building TEWI rather than just chiller TEWI for a large office that would be an all-electric building if the chiller plant is all-electric, but would use gas for space heating if some gas chillers were used. Electric resistance heat at 100% end-use efficiency is assumed for electric chillers. Gas heat at 80% end-use efficiency is assumed for gas chillers. The chiller plants are assumed to be either all electric or all gas.

Gas Chiller Performance

The gas chillers considered in the sensitivity analyses include direct-fired double-effect absorption machines, engine-driven vapor compression machines with and without heat recovery, and direct-fired triple-effect absorption machines. These units are commercially

available except for the triple-effect, which is expected to be introduced in 1996 (Byars 1994). The seasonal performance factors for these units are stated in terms of gas-only COP. The mean values of gCOP used in this analysis are drawn from 1994 data provided by the American Gas Cooling Center (AGCC):

- direct-fired absorption, double-effect, gCOP=1.0
- direct-fired absorption, triple-effect, gCOP=1.5
- engine-driven vapor compression, gCOP=1.6
- engine-driven vapor compression with heat recovery, gCOP=2.0

Engine heat recovery can be used for heating, water heating, or to drive indirect-fired single-effect absorption chillers; therefore gas engine-driven chiller performance is examined with and without heat recovery. In the heat recovery case, gas savings are estimated assuming that 25 percent of the available waste heat (from engine coolant and exhaust gas) is utilized to replace 80 percent efficient gas boiler heat. These gas savings are then deducted from gas consumption for cooling to derive an effective cooling gCOP.

In addition to gas, absorption chillers have on-board electrical requirements for solution pumps and other auxiliaries. These were assumed to be 0.010 kW_{input}/kW_{output} (0.035 kW/ton) for double-effect (same as original AFEAS/DOE study) and 0.013 kW/kW (0.045 kW/ton) for triple-effect. All of the gas chillers have incremental cooling tower heat rejection requirements above those for electric chillers on a per cooling delivered basis. The extra condenser-side electric auxiliaries are estimated using the same methodology as in the original study.

Commercially available gas-engine-driven vapor compression chillers use high-pressure refrigerants (typically HCFC-22) and must use open drive compressors with rotary seals. Most electric-motor-driven chillers have both the motor and compressor within a hermetic seal, but may use high-pressure (HCFC-22, HFC-134a) or low-pressure (HCFC-123) refrigerants. Credible data addressing the relative annual refrigerant loss rates of open versus hermetic configurations, or high- versus low-pressure configurations were not available, and therefore the same loss rates were assumed in all cases.

Sensitivity Analysis Results

The sensitivities of chiller TEWI to refrigerant charge loss rate, CO_2 emissions associated with electric power generation, and electric chiller seasonal efficiency; and of building TEWI to heating to cooling load ratio are shown in Fig. 41, Fig. 42, Fig. 43, and Fig. 44. The results on each figure are discussed consecutively below.

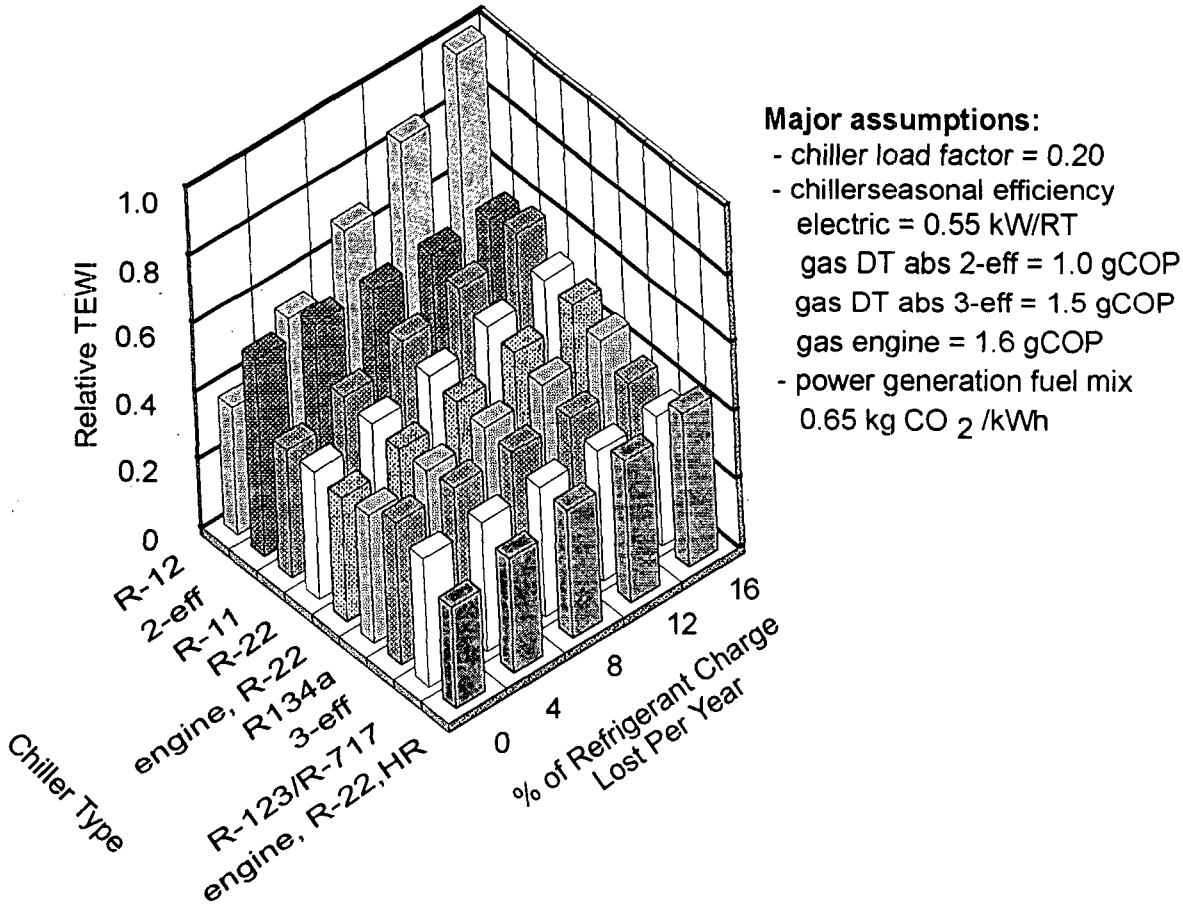


Fig. 41. Sensitivity of chiller TEWI to refrigerant charge leakage rate.

Refrigerant Make-Up Rate

The sensitivity of chiller TEWI to refrigerant charge loss rate is presented in Fig. 41. The major assumptions are listed on the figure. Make-up rates of less than 4% per year are achievable for new centrifugal chillers and could be as low as 0.75% per year (Hourahan 1994). The chiller types are listed left to right in descending order according to chiller TEWI at the mid-point loss rate. It is evident from Fig. 41 that a large part of the warming problem associated with refrigerant loss goes away as CFCs are retired. Nonetheless, HCFC-22 and HFC-134a machines need low loss rates for warming competitiveness and HCFC-22 and HCFC-123 machines need low loss rates for ODP competitiveness. Engine-driven machines need low loss seals with their open compressor designs.

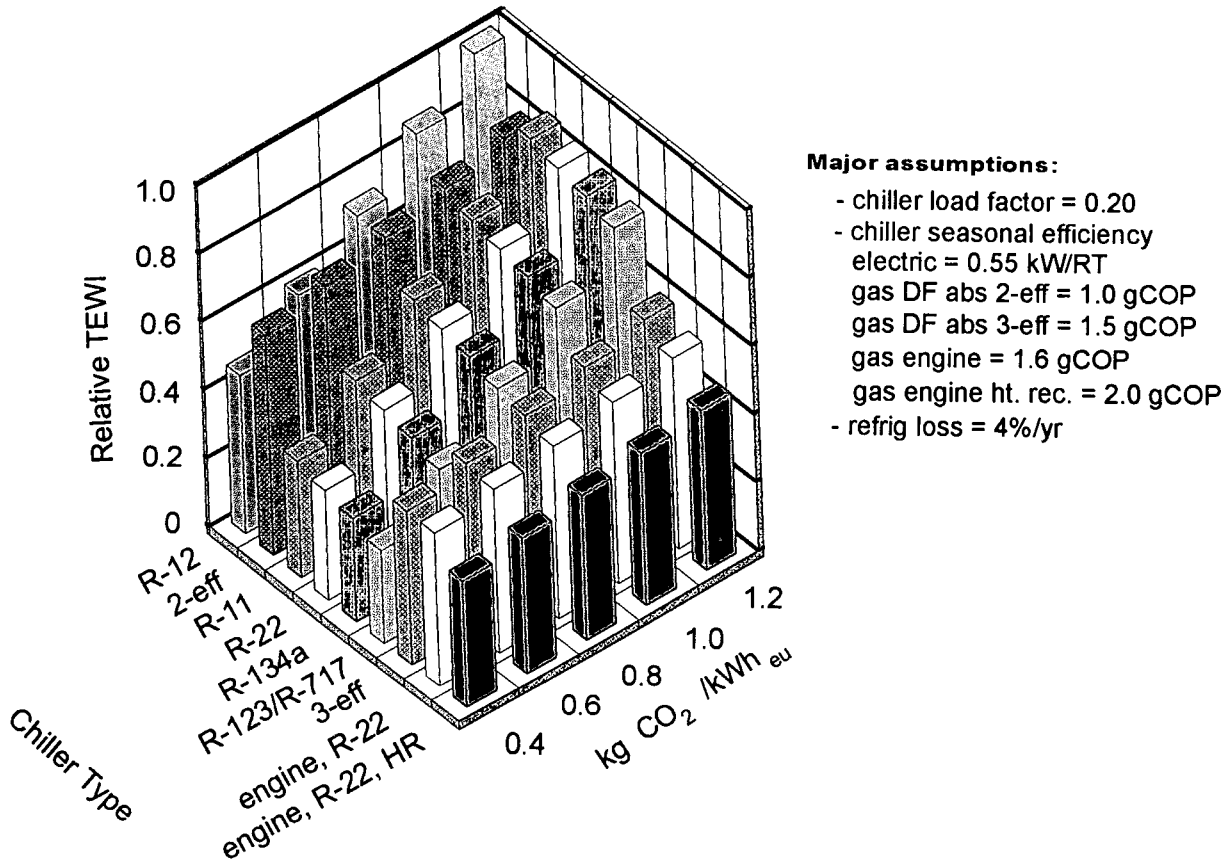


Fig. 42. Sensitivity of chiller TEWI to CO_2 emissions to generate electric power.

CO_2 Emissions from Electric Generation

The sensitivity of chiller TEWI to CO_2 emissions associated with electric power generation is presented in Fig. 42. If North America was experiencing a large need for additional electric generating capacity and this need were to be met with gas combined cycle power plants (about 0.4 kg CO_2/kWh operating at full load at mild temperatures), one could argue that a chiller early retirement program should only consider the placement of electric vapor compression machines. However given the actual circumstances a case can also be made for efficient gas chillers. The chiller TEWI values presented in Fig. 42 assume all chillers in a building are of the stated type. The cost of a chiller early retirement program would be minimized by making the upgrade as financially attractive to building owners as possible so that utility bill savings finance most of the costs. In some areas the lowest cost option for reducing CO_2 emissions may be gas chillers or a mix of electric and gas chillers. Chiller

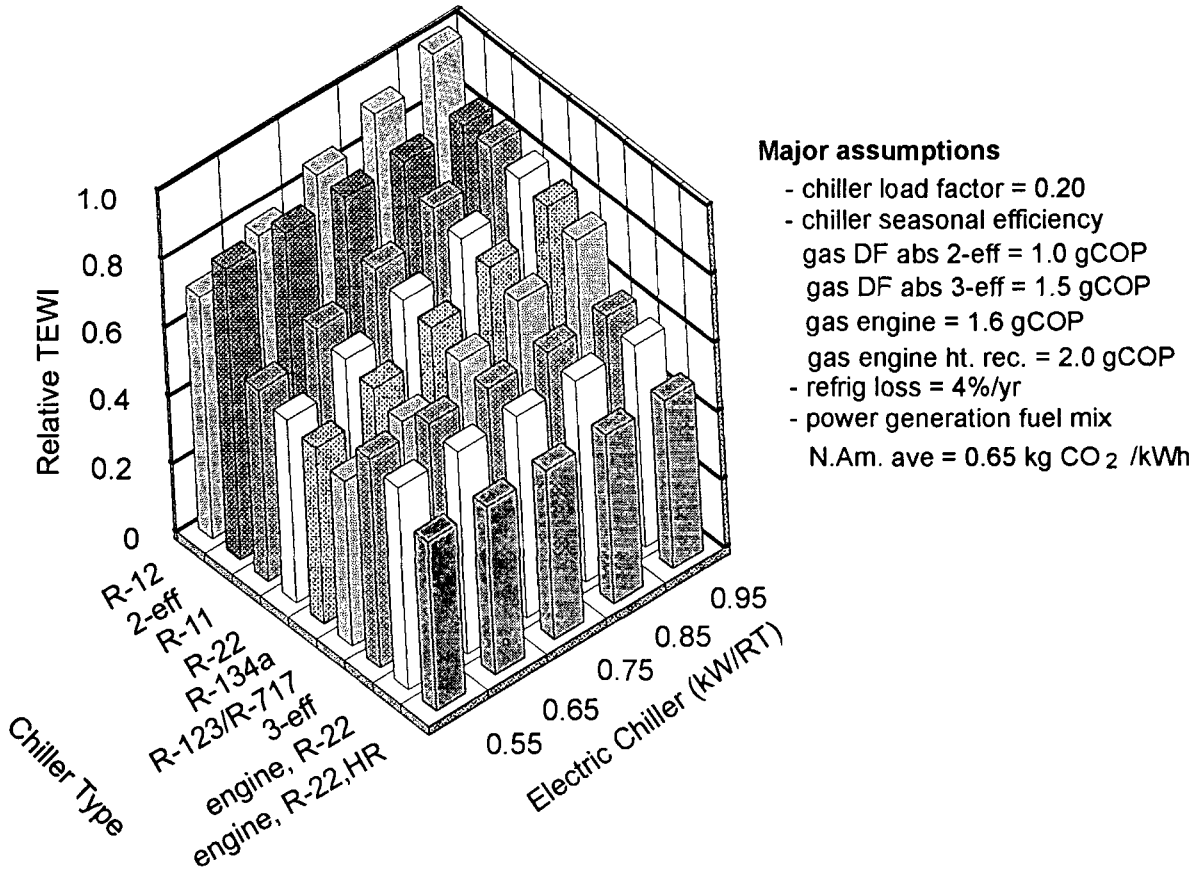
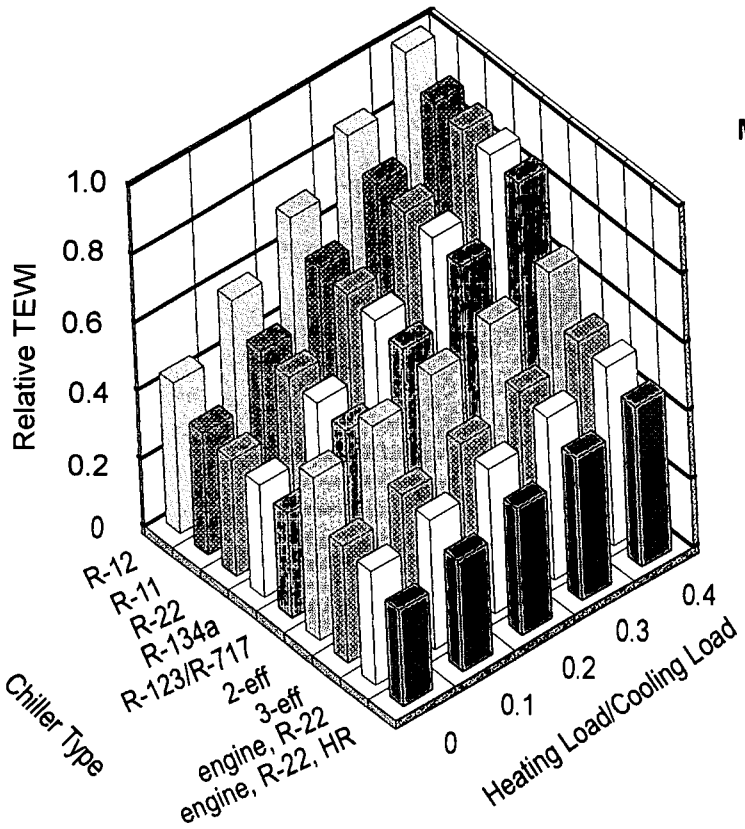


Fig. 43. Sensitivity of chiller TEWI to electric chiller seasonal efficiency at end-use.

TEWI for 50/50 or other mixed plant options can be generally estimated from Fig. 42.

It is acknowledged that the general approach used in Fig. 42 is a simplification of emissions from electric power generation. All of the developed countries and many of the developing ones operate on grids distributing power from many different plants and even from different forms of power generation (e.g. fossil fuel steam plants, nuclear, and hydroelectric power plants). Grid averages for CO_2 emissions vary geographically because some regions have a preponderance of hydro capacity while others rely heavily on coal-fired plants, and even by time of year. Most regional averages fall within the range selected for Fig. 42 (individual countries which rely almost exclusively on hydroelectric power, such as Norway, may fall below the lower limit in the figure).



Major assumptions:

- chiller load factor = 0.20
- chiller seasonal efficiency
  gas DF abs 2-eff = 1.0 gCOP
  gas DF abs 3-eff = 1.5 gCOP
  gas engine = 1.6 gCOP
  gas engine ht. rec.= 2.0 gCOP
- refrig loss = 4%/yr
- power generation fuel mix
  N.Am. ave = 0.65kg CO_2 /kWh
  heating is elec.res. or 80% eff. gas

Fig. 44. Sensitivity of building TEWI to ratio of heating to cooling load.

Electric Chiller Seasonal Efficiency

The sensitivity of chiller TEWI to electric chiller seasonal efficiency is presented in Fig. 43. At the North American average electric generation fuel mix, the 6.39 rated efficiency (0.55 kW/ton) of the most efficient commercially available electric chiller (1993) would imply an advantage over other chillers. The values at the inefficient end of the spectrum (0.9 kW/ton) represent very old or poorly operated electric chillers running at COPs of 3.91 (0.9 kW/ton). For new centrifugal chillers, even the least efficient centrifugal chiller that can be purchased has a COP of 4.40 to 4.69 (0.75 to 0.80 kW/ton). Rated performance and actual in-service chiller performance may differ. Whether these differences are the same across all classes of chiller equipment is not clear. CO_2 emissions are related to actual in-service performance.

Under most conditions, high efficiency electric centrifugal chillers using either HCFC-123 or HFC-134a have lower TEWIs than any of the commercially available gas-fired alternatives

or the triple-effect absorption chiller that is being developed. The triple-effect chiller may have a lower TEWI in some circumstances; for example:

- compared to centrifugal chillers with low installed efficiencies, COP less than 4.69 (greater than 0.75 kW/ton),
- compared to moderately efficient centrifugal chillers (5.41, 0.65 kW/ton) with high CO_2 emissions for power generation (greater than 0.80 kg/kWh), or
- relatively high heating to cooling load ratios.

Heating to Cooling Load Ratio

The sensitivity of building (not chiller) TEWI to heating to cooling load ratio is presented in Fig. 44. For large offices the heating to cooling load ratio in colder climates is about 0.4, as represented by South Bend, Indiana in the U.S. This ratio in warmer climates is about 0.1, as represented by Houston, Texas (MacDonald 1991). If excluding gas from the chiller plant leaves insufficient economic justification for the building to be connected to the gas grid, the all-electric building may cause more CO_2 emissions than a mixed fuel building when both heating and cooling are considered. If gas would be used for heating whether or not gas is used for cooling, heating has no incremental effect (i.e., the effective heating to cooling load ratio is 0.0).

Summary

TEWI was developed to provide a simple indicator of global warming impact of CFC alternative technologies that considered both direct (chemical) emissions and indirect (energy) emissions over the end-use service life. As with any other simple indicator, calculated values of TEWI are sensitive to the underlying assumptions that must address future technologies, markets, and economic conditions. since achieving full consensus on such assumptions is difficult, this sensitivity analysis was designed to accomodate a wide range of performance values for key variables that most affect chiller-related CO_2 emissions.

The analysis also recognizes that global warming is only one issue that may affect chiller selection. Other important factors as cost, convenience, and reliability must be considered and integrated into the selection process. This analysis also provides an indication of how TEWI for any application might vary in relation to the underlying assumptions.

CONCLUSIONS

The TEWI of technology options have been evaluated for various end-use applications using 100 year global warming potentials for each of the relevant greenhouse gases. This choice of time horizon primarily affects just the next generation alternatives based on fluorocarbon refrigerants and blowing agents and to a lesser extent hydrocarbon refrigerants and blowing agents. This is a significant departure from the 500 year time horizon used in the 1991 study and represents a shift in the balance between the total warming impacts of refrigeration, insulation, and air conditioning and the potential impacts over the next couple of decades. Ultimately, the choice of time horizon is a political issue beyond the scope of this report.

Scientifically, there are arguments for using infinite time horizons or approximating them by using a 500 year time horizon. This was the basis of the 1991 AFEAS/DOE project. Scientists have also stated that while long term change reflects the cumulative effects, there is also a real concern that the continued high release rates of greenhouse gases could affect the rate of climate change in the next several decades. For these reasons, policy makers typically use a 100 year integration time horizon. This second study of AFEAS and DOE presents the principle comparison data on a 100 year time horizon basis.

A consistent conclusion drawn from using the 100 year or 500 year time horizon is that improving energy efficiency is a powerful tool to mitigate future potential climate change since it is directly connected to energy related CO_2 emissions. Furthermore, emissions of fluorocarbons need to be, and will be, minimized wherever practical or their direct contribution to global warming can offset or outweigh the efficiency benefit derived from the use of fluorocarbons as refrigerants and blowing agents when looking at TEWI values for most applications.

GENERAL CONCLUSIONS

- while use of non-fluorocarbon, not-in-kind technologies may expand in already established niche markets and other NIK technologies may find new market applications, it appears unlikely that conventional and fluorocarbon technologies can be displaced to any large degree in the foreseeable future. In many cases, there is a greater potential for complementary NIK technology use in "hybrid" or combined NIK/fluorocarbon systems. Examples of hybrid systems would be desiccant dehumidification with fluorocarbon compression for supermarket air conditioning, evacuated panels in combination with polyurethane foam for appliance insulation, and absorption and centrifugal chiller plants for space cooling in large buildings.

- TEWI combines the global warming effects of direct emissions of greenhouse gases and the indirect effects of CO_2 emissions from energy use by using the IPCC GWPs of the gases. TEWI for an application computed using GWPs corresponding to a time horizon that allows for substantial atmospheric cleansing of even the long-lived CO_2 accounts for most of the effect of the greenhouse gases (i.e. refrigerants, insulation blowing agents, and CO_2) on the atmosphere and provides a balance between energy efficiency and fluorocarbon emissions. Using GWPs for short time horizons understate the importance of energy efficiency and the effects of CO_2 or other long-lived trace gases on potential global warming.

- the direct effect of refrigerant and blowing agent emissions in non-hermetic refrigeration and insulation applications is exacerbated by using short-term (100 year) GWP values in calculating the TEWI (the 100 year GWP values are approximately three times the 500 year values used in the 1991 TEWI report). The choice of time horizon is beyond the scope of this report. The aggressive industry response to reduce fluorocarbon losses will result in significantly lower TEWI for applications such as supermarket refrigeration and automobile air conditioning.

- in most applications, not-in-kind technologies will need to compete with the improving energy efficiencies of conventional refrigeration, air-conditioning, and insulation technologies; since the mid-1970's energy use for refrigerator/freezers in the U.S. has been reduced by over 50%, the efficiency of unitary heat pumps and air conditioners have improved an average of 2½% per year for the last 25 years, and the specific powers of centrifugal chillers have improved from 0.21 kW_{input}/kW_{output} to 0.15 kW_{input}/kW_{output} (0.75 kW/ton to 0.52 kW/ton, a 30% reduction). Some calculations in this report are based on efficiencies achieved with fluorocarbons and hydrocarbons by early 1994 and in the future efficiencies will be higher as equipment and foam formulations are optimized for replacement refrigerants and blowing agents.

- TEWI are calculated for centrifugal, double- and triple-effect absorption, and engine driven chillers to examine the sensitivity to fuel source emissions across a range of plant emissions representative of the CO_2 discharge rates from older coal-fired plants to modern combined cycle gas-fired plants. Under most conditions, electric driven centrifugal chillers using HCFC-123, HFC-134a, ammonia, or HCFC-22 have lower TEWI than the gas-fired alternatives. Absorption chillers are often used in "hybrid" chiller plants working in tandem with centrifugals to reduce peak electrical demand and are unlikely to displace electric centrifugal/fluorocarbon technologies in the U.S. to any significant degree in the foreseeable future.

APPLICATION SPECIFIC CONCLUSIONS

Automobile Air Conditioning

- the direct effect of refrigerant emissions for HFC based automobile air conditioners is a significant part of the total at present and an aggressive industry response to reduce charge size and leakage could significantly reduce the impact of fluorocarbon loss from this application. Opportunities exist for alternative technologies such as CO_2 compression and adsorption air conditioners to reduce TEWI for this application, but the long term efficiency, cooling capacity and equipment lifetimes of those technologies must be proven through extensive testing and none are expected to be viable in consumer products before 2003 to 2010.

Refrigerator/Freezers

- based on the latest available published information, there are no significant differences between the energy consumption for the European refrigerator using HFC-134a as both refrigerant and blowing agent and those using propane/isobutane and cyclopentane. The most efficient combination of refrigerants and blowing agent examined is the use of HFC-134a as the refrigerant and HCFC-141b blown foam insulation as developed for the North American refrigerators. It should be noted from existing data that the design of the refrigerator is a major factor in minimizing the TEWI of the system. At present HCFC-141b blown foam has the highest insulation value of the foam blowing alternatives investigated, and results in reduced energy consumption for equivalent fridge design and insulation thickness, when compared to other foam blowing alternatives. Vacuum panel technology can further improve insulation value.

- the direct impacts of the fluorocarbon refrigerants and blowing agents range from 15% to 29% of the total equivalent warming impact for new refrigerator/freezers in North America. The refrigerators using HFC-134a as the refrigerant and HCFC-141b blown foam have 15% of the TEWI due to fluorocarbon emissions and those using HFC-134a as refrigerant and blowing agent have 18% of the TEWI from the direct effect. In each case, almost all of the direct effect is due to the foam blowing agent; assumptions on mandatory recovery of the refrigerant would result in marginal reductions in the TEWI. The same trends exist for the European refrigerator/freezers, although the percentage of direct effects is somewhat higher, primarily because the refrigerators are smaller and consequently have lower annual energy use. The absolute value is similar. The CO_2 emissions rate for the power generation mix in Europe (higher percentage of nuclear and hydroelectric power generation with virtually zero CO_2 emissions) is also a factor. In fact running a fridge in the best or worst European country in terms of CO_2 generation per kWh has a far greater effect on the TEWI than any of the factors discussed here.

- the use of hydrocarbons could result in TEWI reductions for refrigerator/freezers, but efficiency, safety, and product liability considerations can be of equal or greater importance than the lower direct effects of the hydrocarbon system. Hydrocarbons have been accepted as both refrigerants and foam blowing agents for refrigerator/freezers in Europe and could be used in the developing countries. Appliance energy efficiency requirements and both technical and non-technical issues related to safety and manufacturer liability will restrict or prohibit application of hydrocarbon in home refrigeration in other markets.

Unitary Air Conditioning Equipment

- the direct effects of the TEWI for all four vapor compression systems presented are important, but manageable, fractions of the total in each case. Rigorous service procedures for refrigerant recovery and accounting for refrigerant usage should be effective at reducing the direct effect. Contributions to global warming from energy usage are either better or not significantly worse than those of current technology air conditioners; engineering optimization is likely to reduce CO_2 emissions with the mixtures even further.

- there are opportunities for some NIK technologies with lower TEWI than electric heat pumps and air conditioners for residential and light commercial space conditioning in certain climates. Evaporative cooling can reduce TEWI substantially in dry climates relative to vapor compression cooling. Evaporative cooling can be integrated with conventional vapor compression air conditioning or desiccant dehumidification with efficiency and TEWI improvements but with a higher equipment and maintenance cost. Gas-fired adsorption and absorption heat pumps can reduce TEWI in climates dominated by heating requirements but could result in significantly larger TEWI values in climates with high cooling requirements. In any gas to electrical driven comparison it will be important to compare actual CO_2 emission factors from specific power plants instead of the North American regional average used in this analysis. The long term performance of these systems needs to be demonstrated.

Supermarket Refrigeration Systems

- supermarket refrigeration systems have historically used large refrigerant charges and experienced high leakage rates. The current high costs of refrigerants and environmental regulations are resulting in better efforts at refrigerant containment and lower loss rates. The most likely substitutes for CFCs and HCFCs in supermarket refrigeration are mixtures of HFCs, although ammonia compression with indirect heat transfer loops is also being used in Europe. The HFC mixtures developed and demonstrated reduce the TEWI for conventional supermarket refrigeration systems without increasing the component from energy consumption. The direct contribution to global warming from refrigerant emissions continues to be relatively high with each of the mixtures considered. For existing equipment designs significant reductions in

TEWI can be achieved through better containment practices; government regulations and policies, such as the U.S. requirement to meet a 15% leakage rate in systems containing greater than 22 kg (50 lb) of refrigerant. The industry is targeting a 5% leakage rate for new systems. Refrigerant prices are likely to encourage end-users to reduce refrigerant emissions from their historically high rates with consequential reductions in TEWI.

- Building codes in the developed countries make it expensive to use, or in some cases prohibit the use of, ammonia in most supermarkets because of the public safety risks in high population densities near stores. In those areas where ammonia could be used, secondary loops are required so that the ammonia lines do not enter the retail sales area of the building; this imposes both first cost and efficiency penalties. Secondary loops are one means of reducing refrigerant charges and controlling leakage and emissions, whether ammonia or HFCs are used as the refrigerant, and reducing TEWI for this application, depending upon energy consumption and the refrigerant emissions balance. Conscientious preventive maintenance programs for conventional systems with rigorous efforts to locate and fix leaks will also reduce TEWI.

Chillers for Air-Conditioning

- it is difficult for alternative technologies to compete on a TEWI basis with centrifugal water chillers for air conditioning of large buildings because chillers are highly efficient and advances in low-loss purge units for systems with low pressure refrigerants have reduced refrigerant emissions significantly. The direct effect for a chiller using HCFC-123 is less than 1% of the TEWI using refrigerant make-up rates for a state-of-the-art purge system. High pressure HFC refrigerant chillers are projected to have similar leakage rates in the future. Evaporative coolers for office buildings can have a TEWI much lower than that for chillers under arid climatic conditions. TEWI were only calculated for centrifugal chillers and evaporative coolers providing 1050 kW (300 tons) of equivalent cooling in a commercial building. The TEWI of an evaporative cooler is only 40% that of the chiller in southwest U.S. (Tucson, Arizona) but 55% higher in the midwest (Chicago, Illinois), where the evaporative cooler also could not meet all of the cooling requirements. Such data demonstrate the difficulties in using evaporative cooling where humidity conditions are high or vary greatly.

- gas-fired chiller technologies have lower TEWI than electric centrifugal chillers when simultaneous heating and cooling are required and heat recovery is used. Electric driven centrifugals have the lowest TEWI when their installed COPs are above 4.69 (specific powers below 0.75 kW/ton). A triple-effect absorption chiller, based on computer projections of efficiency, and an engine-driven HCFC-22 chiller could have lower TEWI than an electric centrifugal with an installed COP of 5.41 (specific power of 0.65 kW/ton) if the power plant emissions are above 0.8 kg CO_2/kWh. These conclusions are relatively insensitive to the assumed refrigerant make-up rate.

Building Insulation

- plastic foam insulations are used either as part or the major source of insulation in residences. The use of HCFCs in place of CFCs in polyurethane, polyisocyanurate, and extruded polystyrene foams can result in significant reductions in TEWI for this application. Hydrocarbon blown foams could result in lower TEWI than HCFC or HFC blown foams, but at increases in energy use as great as 5%. Further optimization of HFC and hydrocarbon foam formulations are likely to improve foam thermal conductivities and reduce energy use and TEWI more than that projected in this report.

- HFCs are also being developed as foam blowing agents for building insulation, although much of this work is at a very early stage. Little data are available for fresh and aged foam thermal conductivities. Significant work needs to be done in optimizing foam formulations with these zero ODP, non-flammable blowing agents before meaningful conclusions can be drawn about their potential impacts on global warming.

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