Energy and Global Warming Impacts of HFC Refrigerants and Emerging Technologies

Prepared by

Oak Ridge National Laboratory
Oak Ridge, Tennessee

AFEAS
Alternative Fluorocarbons Environmental Acceptability Study

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1. EXECUTIVE SUMMARY

This report summarizes the results from the third phase of a project sponsored by the U.S. Department of Energy (DOE) and the Alternative Fluorocarbons Environmental Acceptability Study (AFEAS).[1] The study is a comparative analysis of the global warming impacts of alternative technologies for refrigeration, air conditioning, and appliance insulation that could be commercialized during the phase-out of hydrochlorofluorocarbons (HCFCs) mandated under the Montreal Protocol. The analysis utilizes a *systems approach* to determine the overall contribution to global warming from an individual air conditioner, heat pump, or refrigeration system during its operating lifetime- a concept known as the "total equivalent warming impact (TEWI)."

Air conditioners, heat pumps, and refrigeration systems can affect global warming through the release of refrigerants and insulation blowing agents directly into the atmosphere and also through the release of carbon dioxide from the generation of electricity to power the system throughout its lifetime. The results of the TEWI analysis for refrigeration and air conditioning applications are presented in this report.

The TEWI concept provides a useful tool in the assessment of various competing technologies. However, it is only one of many criteria that must be considered as illustrated in Fig. 1. Safety, health, and other environmental concerns, system initial and operating costs, regional energy considerations, and ease of maintenance are among other important factors that must be evaluated in the selection of the "best technology" for any given application. Finally, TEWI is dependent on the electricity fuel mix in a given area; as such it should not be considered absolutely, but rather comparatively.

1.1 INTRODUCTION

The initial TEWI study compared the global warming impacts of alternatives to fully halogenated chlorofluorocarbons (CFCs) in refrigeration, air conditioning, insulation, and solvent cleaning applications that could conceivably 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, polymer 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. Direct contribution to 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 (CO_2) as a reference gas (GWPs of CO_2 = 1.0 regardless of time horizon). TEWI provides a measure of the environmental impact of greenhouse gases from operation, service and end-of-life disposal of equipment.

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

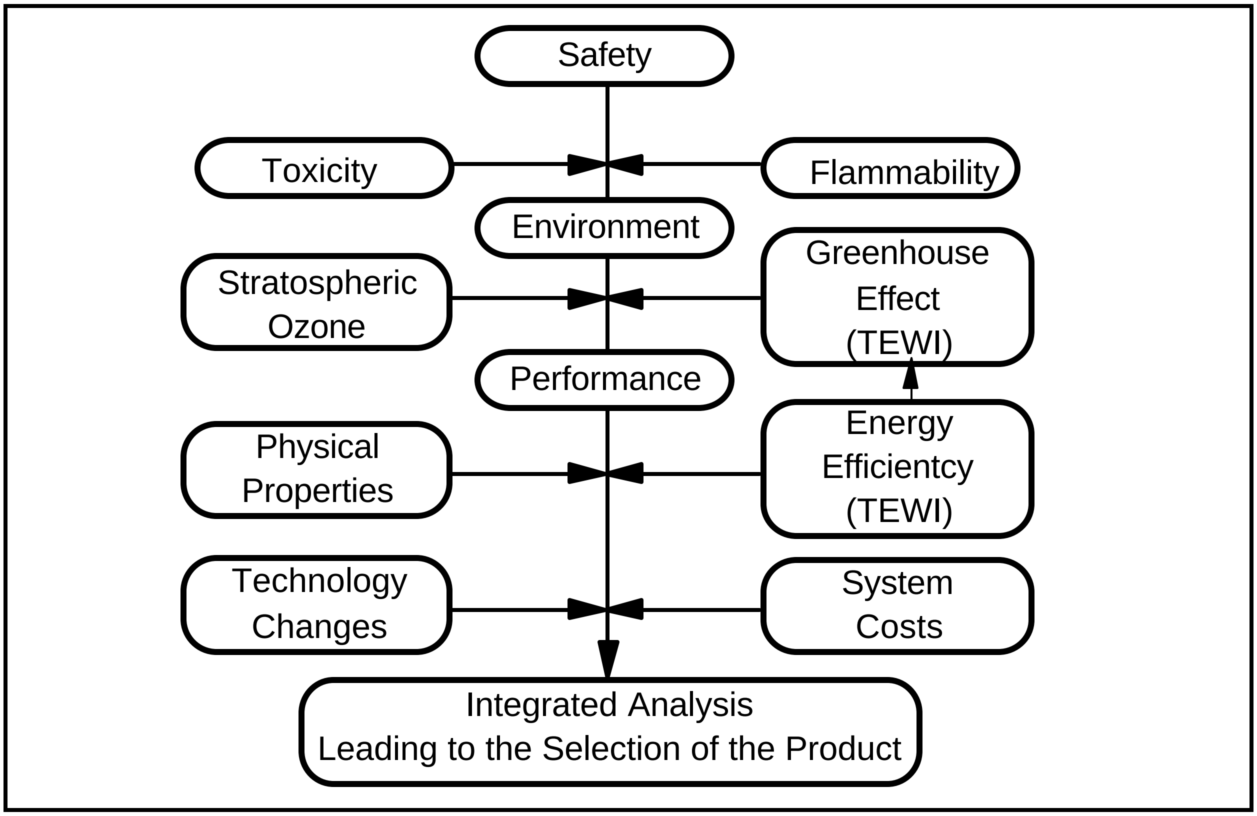

Figure 1. Factors to consider in an *Integrated Approach* to select an optimum solution for a given heating, cooling, or refrigeration requirement.

The second AFEAS/DOE study was initiated to evaluate the energy and global warming impacts of newly developed fluorocarbon replacements for chlorine-containing refrigerants and blowing agents as well as not-in-kind (NIK) non-fluorocarbon technologies that could conceivably be developed or improved in the future to displace vapor compression refrigeration and air conditioning and foam insulations (AFEAS has continued work separately on solvent cleaning applications, but these activities are not part of the second and third phases of work sponsored by AFEAS and DOE). Next generation refrigeration and insulation technologies were classified into three categories:

- zero ODP HFC refrigerants and blowing agents;
- existing NIK commercial technologies (i.e., ammonia (NH_3) and hydrocarbon (HC) compression, desiccant drying, evaporative cooling and absorption chillers); and
- NIK technologies not currently at a commercial level (i.e., acoustic compression, adsorption, Stirling, transcritical vapor compression, etc.).

The major change in assumptions between the first and second studies was the use of GWPs based on a 100 year time horizon in the second study. GWPs of different gases depend on the infrared absorption properties of the gas and the elapsed time before it is purged from the atmosphere, and are relative to the natural cycle of carbon dioxide. The IPCC published GWPs of many gases relative to CO_2 for short, medium, and long time periods, or integration time horizons. Scientifically, there are arguments for using GWPs based on 500 year time horizons and these were used as the basis for the 1991 AFEAS/DOE project. Since 1991, however, the prevailing argument has been that the continued high release rates of greenhouse gases could affect the rate of climate change in the next several decades and that policy decisions should be based on shorter time horizons.. Policy makers have therefore chosen a 100 year integrated time horizon for global warming studies. The second study presented the principle comparison data on a 100 year time horizon basis.

AFEAS and DOE undertook this current study to assess the significant developments that have occurred in HFC blends (i.e. R-404, R-407C, R-410A, and R-507) and the application of non-fluorocarbons like hydrocarbons (HCs), carbon dioxide (R-744 or CO_2), and ammonia (NH_3 or R-717) as refrigerants or foam insulation blowing agents. New data with regard to thermal performance of these compounds made it possible to perform an objective evaluation of the energy and global warming impacts of these "third generation" refrigerants and blowing agents. Refrigerant and equipment manufacturers have also made significant advances in the use of high pressure blends of HFCs as alternatives to HCFC-22 in both refrigeration and air conditioning applications. Analytical and experimental results became available to perform quantitative comparisons between HFC blends and the application of hydrocarbons, ammonia, and carbon dioxide as refrigerants. Additionally, new technologies for gas-fired air conditioning systems are being commercialized and operating data are now available.

As with the previous studies, analyses for end use applications in North America, Europe, and Japan include the effects of cultural and technical differences in each region. The differences include 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, and climate differences for building heating and cooling loads. The results in this report expand the work in the initial studies and indicate important regional differences in TEWI for some applications.

1.2 TECHNICAL EVALUATIONS

1.2.1 Alternative and Next Generation Technologies

Several alternative technologies for the conventional vapor compression (reverse Rankine) cycle employing halocarbons were identified in the Phase II (TEWI-2) report that had been laboratory tested or had been developed to a point where they were considered as potential alternative technologies for those currently used. For example, employing CO_2 in a transcritical vapor compression cycle or using a high efficiency absorption cycle heat pump showed sufficient promise that they were revisited as potential alternatives in this study. Additional development work has occurred on triple-effect, direct-fired absorption chillers and on absorption heat pumps for residential or light-commercial applications so that estimates of TEWI for this equipment are included in the appropriate sections of this report. Alternative technologies which showed little near commercial promise in the TEWI-2 report, such as thermoelectric cooling, magnetic heat pumps, thermoelastic heat pumps, etc., are not considered here.

1.2.1.1 Engine Driven Systems

Engine driven chillers and heat pumps are considered which employ the same reverse Rankine cooling cycle as conventional electric powered systems, but the electric motor is replaced by an internal combustion engine. This change has very little or no effect on the *direct* contribution to TEWI resulting from inadvertent releases of the working fluid to the environment, but it can change the *indirect* contribution associated with CO_2 emissions. Factors affecting the *indirect* contribution result from the primary energy source used to drive the cycle and any inherent differences in cycle efficiencies such as changes in part load efficiencies or waste heat recovery, resulting from this substitution. Engine driven air conditioning and refrigeration equipment allows consumers to select natural gas as the primary system fuel source in situations where there are significant energy price advantages, utility rebates, or other incentives. These opportunities have prompted considerable support for research and development (R&D) at HVAC manufacturers and other organizations and have resulted in the commercial availability of the newest generation of high-efficiency packaged natural gas, engine-driven chillers and residential heat pumps. TEWI calculations are presented for gas engine-driven chillers and heat pumps.

1.2.1.2 Absorption Chillers

Absorption chillers are commercially available and represent a major share of the commercial air conditioning market in Japan and a portion of the market in North America. Absorption equipment is often used in “hybrid” plants working together with electric centrifugal chillers to reduce electric peak demands and utility demand charges. Absorption equipment can be an effective component for managing end users’ total energy costs. Single effect absorption chillers are also used in applications powered by waste heat, in which case the lower efficiency may not be as important. Direct-fired, double-effect chillers can simultaneously provide chilled water for air conditioning and hot water for space heating. Triple-effect absorption chillers are under development which are expected to be 20-45% more efficient than current double-effect chillers.

1.2.1.3 Absorption Heat Pumps

Absorption heat pumps are under development for heating and cooling in residential and light commercial applications. Generator absorber heat exchange (GAX) ammonia-water absorption heat pumps are under development in Europe and the U.S.; several field test units have been built in Japan.

One or two manufacturers have prototype units undergoing laboratory and field testing in 1997, with a goal to commercialize this technology by around 2000. Seasonal performance data are not available for these prototype units. These systems have potential to reduce TEWI in areas where heating load dominates, but may have higher TEWI in areas where cooling dominates.

1.2.1.4 Desiccant Dehumidification

Two types of desiccant systems are available for meeting the latent cooling load in building air-conditioning applications. One of these is based on solid desiccants in rotating wheels and the other on liquid desiccants pumped between various components in the circuit. Current designs are used primarily in special applications such as supermarkets, hospital operating rooms, and other niche markets requiring low humidity. Further improvements are necessary in the efficiency, cost, size, reliability, and life-expectancy of desiccant components and systems in order for them to penetrate the broader commercial air conditioning market. Integrated systems which combine desiccant-based components with more conventional air conditioning equipment are available. System efficiency is greatly enhanced if the desiccants are at least partially regenerated with waste heat from other system components. Since these systems tend towards individually-engineered or custom-designed installations, no calculation of TEWI was attempted with the available data.

1.2.1.5 Advanced Vapor Compression

Conventional vapor compression technologies continue to be improved and efficiencies of refrigeration and air-conditioning equipment will be higher in the future for both electric and engine driven systems. Developments leading to these improvements include the use of higher efficiency motors and compressors, more effective heat exchangers, and adaptive controls. Refrigerant losses from applications such as automobile air conditioning and supermarket refrigeration have been, and will continue to be, reduced. Indeed, supermarket equipment now under development and entering the market, such as electric or engine driven systems that circulate a chilled secondary fluid or distributed electrically driven compressor racks, show promise of dramatically reduced refrigerant charge and emissions. Regulations and refrigerant costs provide an incentive to reduce emissions of refrigerant by eliminating intentional venting during servicing, improving maintenance practices and procedures, mandating charge recovery and recycling, and minimizing leaks.

TEWI calculations for fluorocarbon compression systems in this report are based on demonstrated production efficiencies, modeled efficiencies, or proven performance from R&D research laboratory tests (which would probably show different efficiencies than optimized production designs). Refrigerant loss rates used in the previous studies were from a time when it was standard practice to simply vent the refrigerant charge during servicing. These 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 refrigerant make-up rates based on information from industry are used for cases presented in this report.

TEWIs of viable, commercially available compression systems are compared in some sections of the report to estimated TEWIs for emerging technologies which have not yet reached the stage of commercial production. In these instances the best available laboratory or computer modeled performance data are used for new 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 differences in TEWI because of the more uncertain nature of the data used for these non-commercialized, emerging technologies.

1.2.1.6 Evacuated Panel Insulation

Evacuated panels which can be used to improve appliance insulations have very low thermal conductivities. Thin, flat panels are constructed using a filler material such as aerogel, diatomaceous earth, or open cell filler foam enclosed by one or more plastic or metal membranes under a vacuum. "Total panel" thermal conductivities are usually significantly higher than the "center-of-panel" measurements usually cited, because heat transfer through the plastic or metal membranes along the surfaces and near the edges of the panel is enhanced. Evacuated panels, at last report, are being used by a Japanese refrigerator-freezer manufacturer for a commercial product. While evacuated panel insulation could be an effective means to reduce TEWI for refrigerator-freezers, these panels are quite expensive compared to the foamed polyurethane insulations usually used. There continues to be doubt that panels used for cabinet insulation will retain a vacuum and maintain high thermal resistances over the 15 to 20 year lifetime of an appliance. Such panels could be used in conjunction with blown foam insulation to improve the thermal properties of appliance cabinets or to achieve comparable performance using thinner walls, thereby permitting more usable internal volume.

1.2.2 Update of the 1991 and 1994 Analyses

Literature searches, personal contacts, and review meetings were used to obtain up-to-date information on alternatives to CFCs, HCFCs, and HFCs for the applications evaluated in the previous studies. This information was then used with similar procedures and methodologies established in the first two studies to recalculate the TEWI for those applications where significant developments and changes have occurred since 1991 and/or 1994. Underlying assumptions on equipment lifetimes, CO_2 emission rates from power generation, equipment dimensions, are consistent with those used in the 1991 and 1994 studies. Some of the key findings are summarized below:

1.2.2.1 Refrigerator-Freezers

The latest available published information indicates no significant difference between the measured energy efficiency of refrigeration circuits utilizing HFC-134a or Iso-butane (HC-600a) as the refrigerant.

Insulating foams blown with cyclopentane or pentane isomers consistently show higher thermal conductivities than HCFC-141b blown foams and refrigerators produced with these HC foams would have 8 to 10% higher energy consumption assuming the same foam thickness. Most of the R&D work for this application has centered around finding an alternative for HCFC-141b, which is scheduled for a 2003 phaseout date. Current data shows HCFC-141b blown foam has the lowest thermal conductivity and highest insulating value of the foam blowing agents investigated, which results in the lowest energy consumption for the refrigerator design when equivalent wall thicknesses and internal volumes are assumed. Optimized HFC-245fa or HFC-365mfc blown foam is expected to show similar conductivity and insulating values. Vacuum panel technology can further improve cabinet thermal performance but with significantly increased costs. It should be noted that the design of the refrigerator is a major factor in minimizing the TEWI of the system. Designs consistent with the "average" models prevalent in each region were postulated based on data from manufacturers and industry associations and consistent assumptions on components, wall thicknesses, and internal volumes were applied.

The direct impacts of HFC-134a and the various halocarbon blowing agents range from 8% to 15% of the total equivalent warming impact for refrigerator-freezers in North America. One-tenth of the TEWI for refrigerators using HFC-134a as the refrigerant and HCFC-141b as the blowing agent is due to fluorocarbon emissions. Almost all of the direct effect is due to the foam blowing agent. Mandatory refrigerant recovery would result in a 2 to 3% decrease in total lifetime TEWI in North America.

The direct contribution due to fluorocarbons in European refrigerator-freezers is about 19%, primarily because the refrigerators are smaller and have lower annual energy use. The lower CO_2 emissions rate for electric power generation in Europe, which has a higher percentage of nuclear and hydroelectric power generation than North America, is also a factor. The difference between operating a refrigerator in a country with a very low emission rate from power generation (e.g., 0.02 kg CO_2/kWh) and in one with a high rate (e.g.., 1.08 kg CO_2/kWh) is much greater than the difference in direct effects between using fluorocarbon or hydrocarbon refrigerants and blowing agents.

In 1996, hydrocarbon refrigerators were available in both manual and automatic defrost models in parts of Europe, particularly Germany. Previously, the use of HC refrigerants had been limited to manual defrost models because of concerns that electrically switched devices such as fans, thermostats, defrost heaters, etc. might provide an ignition source for refrigerant that has leaked inside the cabinet. Isobutane (HC-600a) frost-free refrigerator designs which incorporate a foamed-in evaporator and explosion proof electrical devices or switches located outside of the food compartments are now being built and sold. Additional safety precautions and system components have resulted in higher manufacturing, purchasing, and servicing costs for refrigerators using HCs. Flammable refrigerants are not used in United States or Japanese refrigerators due primarily to the higher costs required to mitigate the associated safety risks.

1.2.2.2 Unitary Air Conditioning Equipment

Unitary air conditioners and heat pumps generally fall into four distinct categories, based on primary use and capacity: room air conditioners with capacities between 2.0 and 10.5 kW (0.6 and 3.0

tons[2]); ductless packaged and split systems ranging in capacity from 2.0-20.0 kW (0.6-5.7 tons); ducted systems with capacities from 5.0 to 17.5 kW (1.4 to 5.0 tons); and single packaged or split systems for commercial use whose capacity can be from 20 to 420 kW (5.7 to 120 tons). HFC mixtures have been proposed and tested as substitutes for HCFC-22 in unitary equipment. Hydrocarbons have been studied for this application as well. Building codes and safety concerns in most developed countries limit the use of hydrocarbons in applications where a refrigerant leak could result in explosive mixtures at atmospheric conditions. These restrictions apply to air-to-air heat pumps and air conditioners in North America and Japan (90-95% of the approximately 214 million unitary heat pumps and air conditioners in the world). Hydrocarbon refrigerants might be able to satisfy safety requirements for the air-to-water or water-to-water unitary systems used in Europe where the entire refrigerant charge remains outdoors.

Comparisons were made for HCFC-22 and two non-flammable HFC mixtures identified as likely HCFC-22 replacements in the Air-Conditioning and Refrigeration Institute (ARI) Alternative Refrigerants Evaluation Program. The direct TEWI effects for both HCFC-22 and HFC alternative mixtures are a small fraction of the total in each case. Energy efficiency is very important for this application and contributions to global warming from energy usage with HFC blends are expected to be about the same as or slightly lower than those of current technology air conditioners using HCFC-22; engineering optimization is expected to reduce energy use and resultant CO_2 emissions with the mixtures in future systems. Propane's performance as a refrigerant for air-to-air equipment was reduced by assuming an intermediate heat transfer loop was needed to keep this flammable refrigerant out of the occupied space. Engine driven heat pumps and the Generator Absorber Heat Exchange (GAX) absorption heat pumps are evaluated in a residential gas heating/cooling options chapter and compared with a gas furnace/electric air conditioner combination.

Refrigerant make-up rates and end-of-life losses assumed in this study for 1996 vintage equipment, which are critical in evaluating the direct effect of emissions on TEWI, were suggested by industry experts in each region and are the same as those used for the 1994 study; a 4% annual make-up rate and 15% loss of charge upon equipment decommissioning.

1.2.2.3 Supermarket Refrigeration Systems

These 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 use of ammonia chillers with indirect heat transfer loops is seeing some use in Europe. Alternative refrigerants and technologies are considered as replacements for R-502 in low temperature refrigeration (e.g. freezers and ice cream display cases) and HCFC-22 in medium temperature refrigeration (e.g. meat, fish, and dairy cases). The alternatives

[2] 3.52 kW = 1 ton of refrigeration

include HFC mixtures in direct expansion systems using remote and distributed compressor racks and HFC mixtures or ammonia in secondary loop refrigeration systems.

Secondary loop systems are a means of reducing refrigerant charge and controlling leakage and emissions, albeit with first cost and efficiency penalties. This approach to commercial refrigeration avoids long, field erected refrigerant lines which run to individual cases in the store and confines the refrigerant charge to a smaller, more leak-tight refrigeration circuit in the store's equipment room. The secondary loop approach generally must operate over a larger temperature lift to accommodate the intermediate level of heat exchange (though the magnitude of this extra temperature lift can be minimized by appropriated design optimization) and has an additional parasitic load associated with a fluid circulating pump. 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 densely populated areas near the stores. When ammonia is used, secondary loops are mandatory so that the refrigerant lines do not enter the retail sales areas of the building .

Another improved commercial refrigeration design, usually referred to as the *distributed system* approach, moves the compressor with its associated high pressure liquid and suction gas lines as close as practical to the case evaporator loads and utilizes a closed-loop, water circuit to reject the heat of condensation. The distributed system with compressors located near or in the refrigeration cases requires a larger number of smaller compressors located throughout the store. It, too, has a parasitic load associated with the heat rejection water loop and pump albeit smaller than for the secondary loop system. Both the distributed system and secondary loop approaches drastically reduce the refrigerant charge (by as much as 75 to 90%) and make it more practical to minimize refrigerant leaks and maintain system efficiency.

1.2.2.4 Chillers

The air conditioning loads of larger commercial buildings are generally met with water-cooled chillers which employ cooling towers for heat rejection and which distribute chilled water or a water/antifreeze mixture to building air handlers and fan coil units. Centrifugal or screw compressors are used for larger, 350 to 35,000 kW (100 to 10,000 ton), chillers because of the high volumetric flow rates of refrigerant required. Replacement of CFC refrigerants in chillers with HCFC and HFC alternatives has had the most significant impact on the direct contribution for this equipment. An increased awareness of the environmental impact of refrigerants, recently enacted legislation which requires extensive record keeping, increasing refrigerant prices, and improved equipment designs have all served to dramatically reduce refrigerant loss from chillers. Typical annual loss rates of low pressure refrigerant from new centrifugal chillers has been reduced more than 50-fold in seven years. New systems are equipped with electronic alarms alerting operators to the first indications of leaks or unusual purge pump operation. Refrigerant leak and annual make-up rates have been improved to the point where the GWP of chiller refrigerants has very little effect on the total TEWI.

The TEWI of chillers has been reduced through simultaneous and substantial improvements in chiller efficiencies as well. Rating point COPs for new electric chillers have increased by more than 40% (from 5.0 to 7.0) over the last ten years which has resulted in nearly a 30% decrease in the indirect contribution from CO_2 emissions. Market competition and a greater emphasis on lower life cycle operating costs, as opposed to governmental legislation, are responsible for these dramatically improved performance efficiencies. Even with these improved operating efficiencies, lifetime energy consumption is the predominant factor influencing TEWI for this equipment.

Efficiency of natural gas engine-driven chillers and of absorption chillers has also been significantly improved as noted on pages 4 and 14. Gas-powered chillers are sometimes used together with electric chillers to decrease peak electrical demand and lower building operating costs.

TEWI calculations were made for both vapor compression and absorption chillers of two discrete capacities - 1,200 and 3,500 kW (350 and 1,000 tons) - in North America. These calculations were based on the use of Integrated Part Load Values (IPLVs) and annual operating hours for an Atlanta, Georgia office building. Direct contributions to TEWI were computed for centrifugal and screw chillers using HCFC-123, HFC-134a, HCFC-22, and NH_3 as refrigerants for a range of annual make-up rates up to 4%. 1993 vintage CFC-11 and CFC-12 machines were included for comparison purposes. Similar computations were made for 1,055 kW (300 ton) chiller options in Japan. In this case, rated full-load performance data and associated annual operating hours for a Tokyo location were used. Vapor compression chillers using HCFC-123 and HFC-134a were considered.

For the Atlanta location examined in the report, the direct contribution to TEWI for fluorocarbon-based technologies is at most about 7% of the total (about 0.5% for the HCFC-123 machines) even when the maximum annual leak/make-up rate is assumed. Of course, both the direct and indirect impacts are dependent upon the total operating hours assumed for the analysis. For example, a more northerly location with half the operating hours of Atlanta would yield indirect TEWI contributions one-half as large and the percentage contribution due to refrigerant emissions would approximately double.

1.2.2.5 Automobile Air Conditioners

Automobile air conditioning was identified in the two previous studies as one of the few applications in which the direct contribution of fluorocarbon refrigerant emissions was a significant fraction of total TEWI. While the conclusion has not been contested, the approach taken in those studies has been criticized because of reliance on efficiency data at a single design point coupled with estimates of equivalent full-load operating hours. These two assumptions greatly simplified the analysis but cannot account for varying performance over a range of operating conditions or the effects of different climates. The present analysis addresses these concerns by incorporating efficiency differences across a wide range of operating conditions, regional variations in ambient temperature, and changes in air-conditioner on-time with ambient temperature.

Three fundamentally different cooling systems were considered; a conventional HFC-134a-based system, a propane (HC-290) based system, and a transcritical vapor compression system using CO_2 as the refrigerant. The hydrocarbon system in this study includes the use of a secondary heat transfer loop to isolate the flammable refrigerant outside the passenger compartment. This safety feature reduces cycle efficiency relative to direct expansion systems, adds parasitic energy consumption due to the fluid pump, and increases overall system weight.

Though the analysis includes much more detailed information than the earlier studies, it relies on the same approach of evaluating energy use for operating and transporting the air conditioner and the direct contribution of refrigerant emissions. Results are computed for thirteen different countries. Depending on the location and assumptions about lifetime refrigerant emissions, the alternative systems show potential for lower TEWI than the HFC-134a system. Prototypes of both alternatives are being studied by manufacturers (the CO_2 system to a much greater extent than the HC system). More extensive prototype and field trial testing will be needed before fully developed, reliable commercial designs will be available.

1.2.3 Analysis Limitations

Any calculation of TEWI is based on a selection of assumptions about equipment performance, operating conditions, and CO_2 emissions. All of these assumptions contain a degree of uncertainty that is reflected in the results and this uncertainty must be considered when drawing conclusions about differences in TEWI for two or more alternatives for a given application. The indirect contributions to TEWIs in this report are calculated using data from the open literature for average annual power plant CO_2 emission rates for Europe, Japan, and North America. Calculations are sensitive to these CO_2 emission rates and different comparisons are observed and conclusions drawn when calculations are repeated using annual average CO_2 emission rates for specific countries (e.g. Norway with virtually all electricity from hydroelectric power, Denmark with a high percentage of coal-fired power plants) or emission rates for only peak and intermediate power generation from specific utilities.

It is difficult to calculate an *absolute* value for TEWI. Most of the benefits for TEWI come from using it as a *comparative* tool for assessing the relative global warming impacts of different technology options under a controlled set of assumptions. Uncertainties exist for all of the assumptions (many of which are estimates or averages) that enter into the TEWI calculations. Uncertainties in the direct effect include estimates of refrigerant emission rates and end-of-equipment life recovery rates as well as uncertainties in the GWPs themselves. The indirect contributions from CO_2 emissions can be determined with less uncertainty (especially for specific locales where the fuel source can be characterized accurately) but these values are still not precise. These various uncertainties minimize the importance of small differences in TEWI comparisons between similar, established technologies that use the same energy source. When making such comparisons, if the TEWI differences are small, the technology that shows lower energy use should be favored as long as safety and environmental considerations are adequately addressed and costs are reasonable. In some such cases, a decision between options based on lowest energy consumption could result in choosing a technology with a calculated TEWI that is slightly greater than the minimum. However, in these situations it can be argued that resource conservation and particularly energy resource conservation should take precedence over marginal TEWI differences, due to the uncertainties mentioned above.

It is also difficult to make accurate interfuel comparisons using TEWI *as computed in this report*. At the very least "local" CO_2 emission factors need to be used for electric power generation instead of the broad regional annual averages used in this study, and some technologies may in fact require time of year or time of day factors to account for differences in emissions due to peak power.

These differences are in part due to the types of generating equipment brought on line for peak demand, changes in the local fuel mix, and possibly lower transmission and distribution efficiencies during peak generation periods.

In some cases, sections of this report compare established technologies with known efficiencies to systems under development for which ultimate production efficiency levels and system configurations are less precisely known. When making these comparisons, if the emerging technology exhibits a TEWI 10% or more lower than that of the established alternative then further analysis and development may be justifiable. TEWI, however, would not be the sole determining factor in the ultimate choice between the alternatives.

1.3 CONCLUSIONS

Several broad conclusions can be drawn from the study.

! TEWI evaluations emphasize the combined environmental effect of the direct emission of greenhouse gases with the indirect effects of CO_2 emissions from energy use by equipment using these fluids as refrigerants or blowing agents. This is only one criterion in selecting between technology options. System costs, operating costs, regional energy costs, ease of maintenance, continuing technology improvements, etc., are equally important factors to consider in selecting *the most appropriate* technology for any specific application.

! Reductions in TEWI through the use of ammonia or hydrocarbons as refrigerants are insignificant for refrigeration systems with low emissions and may lead to an increase in energy use when applications of these fluids must meet the same safety design criteria currently defined as acceptable for fluorocarbon refrigerants.

- Ammonia and some hydrocarbon refrigerants have thermophysical properties comparable to (and for some applications superior to) those of HCFC or HFC refrigerants. They also have system irreversibilities and system design features necessary for safe products (e.g., secondary loops) which reduce their overall efficiency. Such changes often offset much of the TEWI benefit claimed for non-fluorocarbon refrigerants.
- Insignificant TEWI differences for most applications occur when design and service requirements, for low refrigerant emissions and safe operation of equipment using flammable or toxic refrigerants, are applied to systems engineered for non-flammable or non-toxic refrigerants.

! TEWI for systems with significant direct effects from refrigerant emissions (i.e. supermarket refrigeration, automobile air conditioning) can be reduced by improved refrigerant containment and maintenance practices or possibly by alternative technologies. Alternative technologies (e.g.

secondary loops, transcritical CO_2) with lower direct effects from refrigerant emissions have potential to reduce TEWI but at lower energy efficiency.

! Efficiencies of conventional technologies are likely to increase as electric and gas-driven equipment and insulating foam formulations are further optimized for replacement refrigerants and blowing agents.

! Innovative design and modifications of standard practice can lead to significant reductions in TEWI for refrigeration systems using ammonia, fluorocarbon, or hydrocarbon refrigerants. These include mandatory refrigerant recovery and recycling, distributed refrigeration systems, charge reduction, elimination of flared fittings and reduced numbers of brazed connections, highly efficient purge units, improved heat transfer surfaces, high-efficiency compressors, etc. Although not included in this study, both active and passive desiccant dehumidification and heat exchange technology are expected to have high potential for reducing TEWI. Use of heat pump technologies for water heating would significantly reduce energy consumption and indirect CO_2 emissions as well.

! Average annual CO_2 emissions from electricity generation vary widely for individual regions and countries - from 0.0 to over 1.0 kg CO_2/kWh compared to the 1993 World average of 0.58. Emission rates also vary with season and time of day depending on how the generation fuel mix changes. Overall TEWI values in any particular location will be peculiar to the local electrical power generating efficiency and seasonal and time of day generating characteristics. The direct contribution can range from all (or nearly all) of total TEWI in areas with low CO_2 emission rates [using mostly nuclear or hydro power] to a minor fraction of TEWI for areas with higher rates [using mostly coal].

1.3.1 Individual Applications

1.3.1.1 Household Refrigeration

The phase-out of HCFCs affects TEWI for household refrigerator/freezers through the choice of refrigerant and insulating material. High efficiency appliances rely on blown foam insulation to achieve a high internal volume for given outside dimensions and low heat leakage into the cabinet. No clear conclusions are possible at this time for foam blowing agents to replace HCFC-141b, although HFC blowing agents have been identified and are under active commercial development which produce insulation comparable to HCFC-141b blown foam. Hydrocarbon blown foams continue to have higher thermal conductivity than HCFC-141b and HFC foams and, consequently, exhibit higher energy use with increased impact on CO_2 emissions. The increased energy use must be balanced against any direct impact caused by the HFC blowing agent itself.

TEWI estimates from this analysis for household refrigerator-freezers using HC refrigerants and foams are about 4-5% lower, in North America and Japan, and about 13% lower in Europe, than those of HFC-134a refrigerant/HFC foam units assuming refrigerant recovery at end-of-life disposal. Energy use estimates for HC-based refrigerators are about 10% greater than that of the HFC units in all regions. Use of HCs in refrigerators raises safety concerns and has resulted in higher unit costs. Applying this cost differential to HFC designs (to incorporate vacuum panel insulation in the cabinet walls, for instance) could yield a product with potentially superior TEWI characteristics.

1.3.1.2 Automobile Air Conditioners

The direct effect of refrigerant emissions for HFC based automobile air conditioners is a significant part of the TEWI. The automobile manufacturers have responded with efforts to reduce charge size and emissions. Research and laboratory development of air-conditioning systems based on both transcritical CO_2 compression and hydrocarbon refrigerants show a potential to reduce TEWI for this application. Estimated TEWIs for CO_2 and hydrocarbon systems are lower than those for HFC-134a systems in regions with cool climates; TEWI are comparable to higher in climates with high cooling loads. Energy consumption estimates for HFC systems are consistently lower than those of CO_2 and hydrocarbon systems. The long term performance, lifetimes, viability, and TEWIs of both the alternative systems must be proven through extensive prototype and field trial testing. Energy consumption and TEWIs for HC-based systems are negatively affected in all regions due to use of an indirect loop with attendant efficiency penalties to keep the flammable refrigerant out of the passenger compartment.

1.3.1.3 Chillers

TEWI for this class of equipment has fallen significantly since the early 90's. New electric chillers have 25% to 30% lower TEWIs than models of 4-5 years ago due to replacement of CFC refrigerants with HCFC and HFC alternatives and to significant improvements in energy efficiency and reductions in refrigerant loss rates. The choice of refrigerant makes only a minor difference in direct TEWI in new equipment. Differences in chiller efficiencies for various refrigerant options can have a significant impact on the indirect contribution to TEWI, however, which is dominant in this application.

Significant advances have been made in gas-fired chiller technologies. Triple-effect absorption chillers now under development show potential for 25-30% reductions in TEWI compared to existing double-effect machines. Engine-driven chillers are now available with rated efficiencies more than 25% higher than the value used in the TEWI-II report. Estimated TEWIs for these machines are around 25% lower than those for the triple-effect absorption equipment.

1.3.1.4 Unitary Equipment

Transition away from HCFC-22 in this equipment appears to be achievable with either no change or a slight reduction in the estimated TEWIs. The HFC-400 blends, R-407C and R-410A, are the principal alternatives being considered as HCFC-22 substitutes. Laboratory and limited field testing indicates that R-407C has equivalent performance compared to R-22 while R-410A-based equipment has potential for slightly better efficiency, lowering the indirect contribution to TEWI. Geothermal heat pumps and premium grade air-to-air heat pumps can significantly reduce TEWI in this application, albeit with higher purchase costs.

Gas engine-driven and gas-fired absorption heat pumps for space heating and cooling show potential to reduce TEWI in climates dominated by heating requirements. Long term performance and reliability of the gas-driven technologies have not been demonstrated.

1.3.1.5 Commercial Refrigeration

Supermarket refrigeration systems have had high direct contributions because of historically high refrigerant charges and leakage rates. Equipment manufacturers have worked to reduce refrigerant leaks at the display cases and in the brazed and welded joints in refrigerant lines. The lower emission rates have reduced TEWI significantly from the values reported previously. New system design concepts (secondary loop and distributed compressor approaches) also dramatically reduce the direct effect of refrigerant emissions and result in lower overall TEWI estimates for this application. The differences in TEWI between the HFC mixtures that have been considered are due primarily to the GWPs of the refrigerants; the differences in energy use are not considered significant. Ammonia with secondary heat transfer loops has been shown to be a viable alternative for HFCs in this application, but there can be an energy penalty associated with necessary secondary heat exchangers. In many areas system designs will have to comply with regulation and permit requirements intended to ensure safe use in retail and commercial areas. Some European and developing countries have fewer regulations and are more open to using ammonia. Refrigerant containment measures necessary for ammonia and hydrocarbons could also be used with HFCs, resulting in essentially identical TEWI for these alternatives.

1.4 SUMMARY

The concept of Total Equivalent Warming Impact, or TEWI, was developed as a comparative index of the global warming impacts of an end use application by accounting for both the direct contributions from refrigerants and blowing agents to the atmosphere and the indirect contributions from energy consumption. It provides a useful tool in the assessment of various competing technologies that could be used to substitute for current technologies as chemical compounds are phased out of use under the Montreal Protocol. TEWI, however, is not the only criteria that must be considered. Safety, health, other environmental concerns, initial system and operating costs, regional energy considerations, ease of maintenance, and system reliability are some other factors that must be considered in evaluating the "best technology" for any given application. As with any other composite index of performance, estimated TEWI values are sensitive to an underlying set of key values and assumptions. Results from these studies show that it is essential to analyze both the *direct* and *indirect* contributions of all air conditioning and refrigeration alternatives to choose the most environmentally acceptable option. The studies have also shown that energy efficiency and reduced refrigerant emissions are the most effective means to mitigate future anthropogenic contributions to global climate change.

2. INTRODUCTION

The first study by AFEAS and DOE (Fischer, et al 1991) compared the global warming impacts of alternatives to CFCs in refrigeration, air conditioning, foam insulation, and solvent cleaning applications that could be commercialized before the year 2000. The refrigeration, air conditioning, and foam insulation portions of that study were conducted by Oak Ridge National Laboratory (ORNL) and the solvent cleaning portion by Arthur D. Little, Inc. That project developed the concept of total equivalent warming impact (TEWI) which combined the effects of the direct emissions of refrigerants, foam blowing agents, and solvents into the atmosphere with the indirect effects of energy consumption due to combustion of fossil fuels and generation of electricity.

AFEAS and DOE supported a second phase study (Fischer, et al 1994) to evaluate the energy and global warming impacts of not-in-kind (NIK) and next generation alternative technologies that could possibly be developed in the long term to replace current commercial practices for vapor compression refrigeration and air conditioning and foam insulations using CFCs and HCFCs. NIK alternatives evaluated included thermoelectric refrigeration, evacuated panel insulation, and adsorption and absorption technologies. Next generation technologies included chlorine-free fluorocarbon refrigerants and blowing agents (HFCs), non-fluorocarbon refrigerants like ammonia and hydrocarbons, and alternative compression technologies like acoustic compressors.

Both of the earlier studies examined end use applications in North America, Europe, and Japan and included effects of cultural and technical differences in each region. These differences included such things as the room and compartment temperatures for refrigerator/freezers, refrigerator size and insulation thickness, annual driving distances for automobiles, fuels used for electricity generation, and climate differences for building heating and cooling loads. In general, the principal results and conclusions in the prior studies were the same without regard for the geographic region.

AFEAS and DOE decided to sponsor this third study for a number of reasons. Significant developments have occurred in HFC and non-fluorocarbon refrigerants and blowing agents since 1993-94 which have not been considered in an independent evaluation of their global warming impacts. Data on thermal performance have become available that make it possible to do a meaningful evaluation of the TEWI of these "new generation" fluids. A significant amount of work has been done on HFC blends as alternatives to R-22 in both refrigeration and unitary space conditioning applications. Much effort has also been devoted recently on use of non-fluorocarbons (eg, ammonia, hydrocarbons, and CO_2) as refrigerants and foam insulation blowing agents in selected applications. Analytical and experimental results are available for quantitative comparisons between HFCs and non-fluorocarbons. In addition, new technologies for engine-driven compression systems and absorption systems for residential and commercial space conditioning are being commercialized and operating data are available. Thermal performance measurements have been made for refrigerator/freezer foam insulations blown with "new generation" fluids.

The scope of the analyses in this report is limited to:

a) NIK or alternative technologies identified in the second study which were concluded to have a reasonable potential for reducing TEWI and for which there are manufacturer development activities underway which could lead to commercial products by the early years of the next century; and

b) evolutionary improvements to existing technologies that have emerged since publication of the phase II report.

The NIK concepts primarily considered were absorption systems for heating and air conditioning residential and commercial buildings, the transcritical CO_2 cycle for mobile air-conditioning, and vacuum panel insulation for household refrigerators. Double-effect absorption equipment is already on the market and more advanced cycles are under development with market entry expected before 2000. A number of automakers and auto suppliers (principally in Europe and Japan) are seriously investigating the transcritical CO_2 concept and are testing prototype systems. Hydrocarbon (HC) vapor compression systems are also considered for household refrigerators, residential space conditioning systems, and automobile air-conditioners.

End use applications considered in the present study are:

* household refrigeration;

* commercial chillers;

* automobile air conditioning;

* supermarket refrigeration; and

* unitary space conditioning.

TEWI computations in both the earlier studies and in the present study are made using the global warming potentials (GWPs) for trace gases developed by the Intergovernmental Panel on Climate Change (IPCC) based on the use of carbon dioxide as the reference (GWP of CO_2 = 1). GWPs based on a 500 year integration time horizon (ITH) were used in the first study while 100 year values were used in the second study. The 100 year GWPs are about three times the 500 year values and, thus, yield TEWIs with a higher relative direct contribution due to the refrigerant or blowing agent. The long atmospheric lifetime of CO_2 results in an effect on climate change that accumulates on a time scale of centuries and, because the calculation of effects is truncated at the ITH, using GWPs at a short ITH can seriously distort the energy component of TEWI. This is particularly problematic at the 20 year ITH. More than 90% of the effect of CO_2 lies beyond this time horizon and so values calculated using it do not meet requirements of the Climate Change Convention for intergenerational equity. On the other hand, despite their better description of the long term effects of CO_2, the values calculated at a 500 year ITH are subject to the greatest uncertainty. Most of the international scientific community and national and international policy makers concerned with global climate change appear to have settled on the 100 year ITH as a reasonable compromise and so those GWP values are used here. GWPs calculated using a 500 year ITH are approximately one third those at a 100 year ITH, simply due to discounting the long term effect of CO_2.

The TEWI estimates in this report also depend on assumptions made about carbon dioxide emission from energy use, either from combustion of fossil fuels in automobile engines or furnaces or indirectly from use of coal, natural gas, or oil for electricity generation. Electric power CO_2 emission rates were obtained from a number of electric utility and independent sources. These data are summarized in Appendix A. Most of the TEWI estimates presented in the main body of the report were generated using regional annual average emissions of 0.473 kg CO_2/kWh delivered for Japan, 0.470 kg CO_2/kWh for Europe, and 0.650 kg CO_2/kWh for North America. Electricity transmission and distribution loss effects are included. The effects of time-of-day variations in electricity generation fuel mix and of fuel extraction and processing energy consumption were not considered in the main body of this report. Appendix A includes a discussion of these factors.

Carbon dioxide emissions from fuel use for furnaces or gas-fired chillers and heat pumps are assumed to be 53.0 g CO_2/MJ input heat value for natural gas. This includes an average natural gas transmission efficiency factor of 96.5%. This is an average for the years 1994-1996 obtained from natural gas consumption data for the U. S. from the Energy Information Administration (EIA, 1997). About 3.5% of the natural gas consumed (exclusive of extraction and processing consumption) was used, primarily by compressors, to distribute the gas to consumers through the pipeline network.

Emissions from gasoline used to drive automobile air conditioners is assumed to be 2.32 kg CO_2/liter of fuel. This assumption is consistent with the data used in the first two studies.

A number of simplifying assumptions are implicit in the TEWI estimates provided herein. It would be very difficult to compute absolute TEWI values for the full range of locations and alternatives included in this report. Such an analysis would have to include location-specific values for electric power plant carbon dioxide emissions (for some applications hourly emissions variations would have to be considered together with hourly energy use simulations). Conclusions from detailed analyses like this would be applicable only to the specific locations considered. This conflicts with a major element of the project's charter: to examine opportunities to reduce CO_2 emissions that are broadly applicable across regions. Thus, these detailed analyses would have to be repeated for numerous locations in each region to fairly illustrate how comparisons between given technology options for each end use application would vary throughout the regions. This is well beyond the scope of, and time available for, this study, therefore most of the comparisons and conclusions are based upon regional annual average values for CO_2 emissions.

In the previous studies, the energies required to manufacture the greenhouse gases themselves were not included nor have they been in this study. The climate change impacts of energy used for production of a number of common refrigerants - HFCs, non-methane hydrocarbons, and ammonia - have been evaluated directly by Campbell and McCulloch (1997). They show that the global warming impact of the production energy is insignificant when compared to the potential impact on global warming of releases of these gases.

Most of the value of the TEWI estimates included in this report come from using them as comparative measures of the global warming impacts of different technology options for meeting a given need in one of the end use applications. Uncertainties exist for all of the assumptions (many of which are estimates or averages) that enter into the TEWI calculations. Uncertainties in the direct effect include estimates of refrigerant emission rates and end-of-equipment-life refrigerant recover rates as well as uncertainties in the GWPs themselves. The indirect contribution from CO_2 emissions due to energy use can be determined with less uncertainty (especially for specific locales where the fuel source can be characterized accurately) but these values are still not precise. These uncertainties minimize the importance of small differences in TEWI between similar, established technologies using the same energy source (i.e. gas or electricity). When making such comparisons, if the TEWI differences are small, the technology that shows lower energy use should be favored as long as safety and environmental considerations are adequately addressed and costs are reasonable. In some cases, a decision between options based on lowest energy consumption could result in choosing a technology with a TEWI that is greater than the minimum possible. However, it can be argued that resource conservation and particularly energy resource conservation should take precedence over marginal global warming impact reductions when TEWI estimates are close.

In some cases, established technologies with known efficiencies are compared to systems under development for which ultimate production efficiency levels and system configurations are less precisely known. When making these comparisons, if the emerging technology exhibits a TEWI 10% or more lower than that of the established alternative then further analysis and development may be justifiable on this basis. TEWI, however, would not be the sole determining factor in the ultimate choice between the alternatives

As in the previous two phases, this study involved experts from industry, government, and academia around the world to characterize existing HCFC-based practices and to identify the operating characteristics of alternative technology options. Therefore, much of the initial scoping work prior to the start of the analyses may be viewed as a survey of the user industries concerning their candidate options to replace HCFCs in the above applications. Unlike the first two studies, more interaction was programmed through the course of the study with these experts. They were called on to provide review and comments at three distinct points: an initial review of the assumptions and methodology proposed for this round of analyses, a mid-term review of preliminary results and conclusions of the analyses, and review of this final report itself. The cooperation of these experts has been vital to the success of this study.

3. REFRIGERATOR-FREEZERS

3.1 INTRODUCTION

Residential refrigerator-freezers are one of the most familiar and inherently useful applications of HCFC and HFC technology which touches our daily lives. Market saturation for domestic refrigerators in the United States, Western Europe, and Japan is greater than 98% which is an indication of how useful and necessary this appliance has become for a modern urbanized society. It is anticipated that an explosive growth in the production and consumption of refrigerators will occur in developing countries like China and India as these societies become more affluent (Deloitte & Touche 1996). The global warming impact of a refrigerator-freezer includes the *indirect* emissions of CO_2 resulting from the combustion of fossil fuels to generate electricity to operate the refrigerator, and the *direct* global warming contributions from the refrigerant and insulation foam blowing agents eventually released to the atmosphere. Prior to 1993, CFC-12 was used as the refrigerant in this application because of its excellent efficiency, safety, and compatibility characteristics which had been fully developed over 50 years of use. The phaseout of CFC-12 in industrialized countries as a result of the Montreal Protocol has prompted the development of HFC-134a and hydrocarbons like iso-butane (HC-600a) and propane (HC-290) as chlorine-free, ozone-safe, alternative refrigerants for CFC-12.

Rigid polyurethane insulating foams blown with CFC-11 were the dominant insulation used in the cabinets of refrigerators prior to the ban on production and sale of CFCs. After the effective date for implementation of the Montreal Protocol, CFC-11 blown insulating foam was rapidly eliminated from refrigerator-freezer production throughout the world. Germany, northern Europe, Australia, and New Zealand are converting to cyclopentane as an appliance foam blowing agent, with some other European manufacturers using HFC-134a, HCFC-141b, and blends of HCFC-142b or HCFC-141b with HCFC-22. U.S. and Japanese manufacturers have predominantly converted to HCFC-141b as an insulation blowing agent for refrigerators because of its excellent insulating characteristics, non-flammability, and low volatile organic carbon (VOC) emission ranking.

3.2 ENERGY EFFICIENCY

Domestic refrigerator-freezers, used primarily for food preservation, are an important and major user of electricity. Approximately 62 million new refrigerators are manufactured world-wide each year primarily to serve as replacements in saturated markets for the hundreds of millions of units already in use (UNEP 1995). Energy efficient domestic refrigerators contribute substantially to world-wide efforts to reduce energy consumption and global warming. Efforts to decrease the energy used by refrigerators have focused primarily on improving the efficiency of the mechanical refrigeration process through more efficient compressor designs and improvements in refrigerant-to-air heat transfer. Improved cabinet insulation which results in a smaller refrigeration load has also significantly decreased the energy consumption of this appliance (Vineyard 1995). Government mandated minimum energy-efficiency standards like those established by the United States under the National Appliance Energy Consumption Act have been effective in motivating the adoption of energy saving technologies (NAECA 1987). A chart showing the actual and projected improvements in electrical energy use for U.S. refrigerators is shown in Fig. 2. The initial standards went into effect on January 1, 1990, and have had one revision in 1993, which resulted in a 25% reduction in energy consumption. The next revision is expected to require about an additional 30% reduction. This revision was originally scheduled for 1998 but has been rescheduled to 2001.

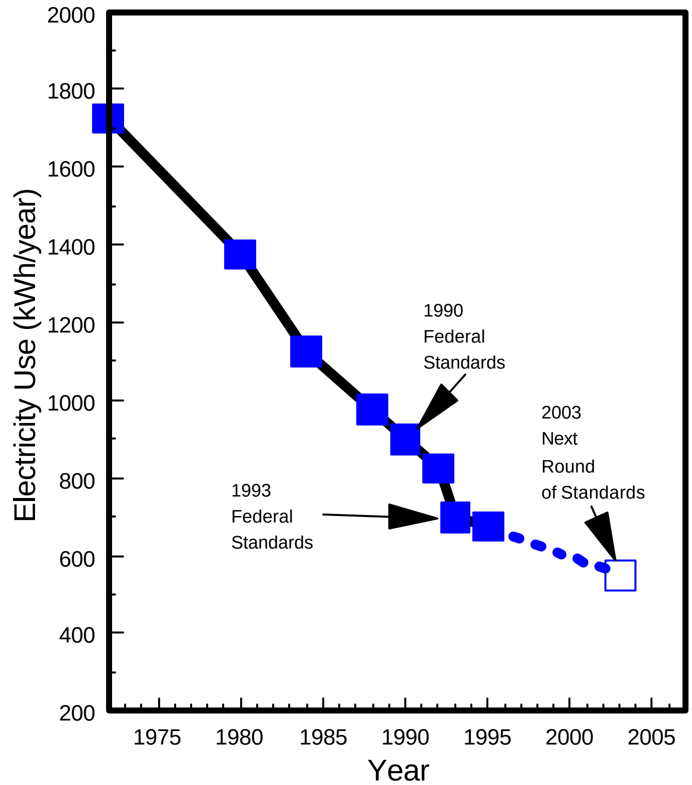

Figure 2. Historical and projected refrigerator-freezer energy improvements for the United States.

The efficient use of energy is important because of its limited availability and also because of international efforts to reduce the atmospheric concentrations of greenhouse warming gases. For refrigerators, which have very small refrigerant charges and perhaps the most leak-free hermetic refrigeration system, the indirect contribution to TEWI is about an order of magnitude greater than the direct contribution of the CFC- alternative refrigerants. So, selection of the most efficient refrigeration fluids for this application, regardless of their individual GWPs, is critical to minimizing its overall global warming impact.

3.3 REFRIGERANTS

3.3.1 HFC-134a

HFC-134a was initially the refrigerant of choice to replace CFC-12 in refrigerator-freezers because it is chlorine-free (zero ODP), non-flammable, non-toxic and has comparable cycle efficiency. Iso-butane, propane, and mixtures of propane and iso-butane were subsequently suggested as refrigerant alternatives, especially by the Northern Europeans, because they are comparable in efficiency to HFC-134a, miscible with conventional refrigeration oils and have much lower GWPs. The boiling point, vapor pressure characteristics, and refrigeration cycle efficiency of HFC-134a are quite similar to CFC-12, but its volumetric capacity is about 12% less, and it is insoluble in the mineral and alkylbenzene oils previously used with CFC-12. Despite a slightly smaller ideal thermodynamic COP for the domestic refrigeration cycle, extensive energy consumption testing in refrigerator-freezers has shown that HFC-134a can achieve approximately equal performance to CFC-12 in optimized units (Radermacher 1996). Synthetic polyolester (POE) oils are usually used in hermetic systems with HCF-134a. POE oils can hydrolyze in the presence of high levels of moisture to produce system damaging acids, therefore additional process controls are required to ensure the maintenance of low system moisture levels. This new refrigerant/lubricant combination has also forced re-evaluation of manufacturing and processing fluids used for production of compressors and heat exchanger tubing to avoid selective solubility and sludge formation that can result in capillary tube plugging (Kruse 1996).

3.3.2 Hydrocarbon Refrigerants

Hydrocarbon refrigerants are flammable which implies safety issues in the manufacture and home use of units charged with these refrigerants. Product liability concerns preclude the use of hydrocarbon refrigerants in the larger frost-free designs that are popular in the United States and Japan. The flammability of hydrocarbons causes increased investments in production factories for additional safety precautions and in the form of several design changes required for certain models to reposition electrical switches or change to explosion proof components needed to avoid potential accidents. German manufacturers have demonstrated that hydrocarbon refrigerants can be handled safely in the manufacturing process and that European customers are willing to accept products using hydrocarbons. Normalized energy consumption data presented in the phase II DOE/AFEAS report for 3-star refrigerators produced in Europe, Fig. 3, and subsequent testing indicate that roughly equivalent efficiencies can be obtained with HC-600a and HFC-134a refrigerants (Fischer 1994, Deloitte & Touche 1996, Wenning 1996). In Fig. 3, 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 cy-clopentane as the blowing agent and HFC-134a as the refrigerant. The broad scatter makes it impossible to draw any definitive

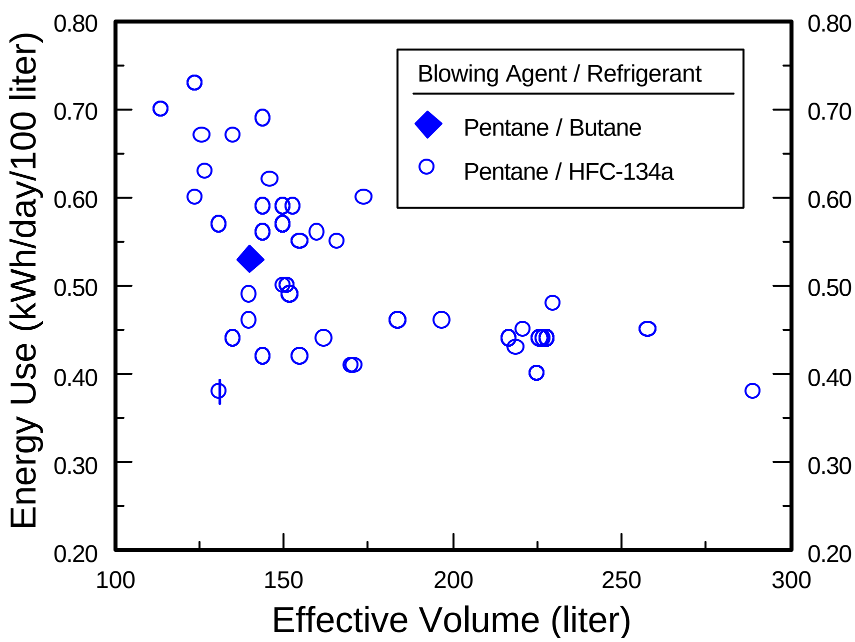

Figure 3.. Measured energy use for European 3-star refrigerator/freezers using CFC alternatives.

conclusions about any one combination being more efficient than the other and indicate that equivalent efficiencies can be obtained with either refrigerant.

In addition to the favorable GWP of hydrocarbon refrigerants, they are also compatible with the materials, lubricants, and manufacturing/processing fluids previously used with CFC-12 refrigerators. The latent heat of vaporization of hydrocarbons is large in comparison to halogenated alkanes and their low density results in smaller charge requirements. Initial hydrocarbon applications used HC-290 /HC-600a (propane/iso-butane, 50/50 mass%) blends to roughly simulate the volumetric capacity and allow the use of compressors designed for CFC-12 or HFC-134a. Use of this blended formulation has decreased in favor of using HC-600a alone which has a volumetric capacity . 80% lower than HFC-134a thereby requiring a larger displacement compressor to obtain the same refrigeration capacity. Compressors with larger displacement can be an advantage since compressor efficiency falls off in the smaller capacity ranges due to manufacturing tolerance limitations.

3.4 Blowing Agents

Heat leakage through the insulated cabinet of a refrigerator is the main source of the refrigeration load so improvements in the thermal conductivity of insulating foams used in the cabinet and the heat integrity of other cabinet components significantly improve the energy efficiency of the product (Sand 1994, Vineyard 1995, UNEP 1995). The rigid strength of the foam and its ability to adhere to cabinet walls and liners is also important because it constitutes an integral element of the cabinet structure and is essential to maintain dimensional stability for the appliance. The material compatibility of blowing agents with the polymer liner materials used for interior surfaces of the refrigerator is another factor which must be taken into consideration when an alternative blowing agent is being evaluated. Care must be taken to be sure that the insulating, structural, compatibility, and processing characteristics are not severely degraded by the choice of alternative blowing agents.

3.4.1 HCFC-141b and HFC Alternatives

HCFC-141b was chosen as a successor to CFC-11 for U.S. and Japanese refrigerators after several years of experimentation with various blowing agent options. Its use as an alternative blowing agent represents a greater than 9 to 1 reduction in the ODP and a 6 to 1 reduction in the GWP over CFC-11. HCFC-141b blown foams have excellent insulating values and equipment previously used for CFC-11 was easily converted for use with this alternative. Early material incompatibility problems of these foams with plastic door liners were solved by using co-extruded barrier on these liners.

In anticipation of the 2003 phaseout of HCFC-141b, appliance manufacturers through their international associations and members of the Polyisocyanurate Foam Manufacturers Association (PIMA), have generated and published a great deal of experimental information on the performance of insulating foams blown with chlorine-free alternative blowing agents (Haworth 1996) (Polyurethane 1995) (Polyurethane 1996). Alternative blowing agents evaluated in the United States and Japan for HCFC-141b include: HFC-134a, CO_2 (water- based), HFC-356mffm, HFC-365mfc, HFC-245fa, HFC-236ea, and cyclopentane. The Association of Home Appliance Manufacturers (AHAM) Insulation Technical Advisory Committee (ITAC) and member companies have measured the relative energy use of 22 ft^3, top-mount, refrigerators blown with HCFC-141b and all of these alternatives using the Department of Energy, 90E F, closed-door test, commonly known as the standard DOE test. Parallel cabinet energy consumption tests for the blowing agents listed above and for HFC-152a and HFC-245ca in Japanese refrigerators were reported by The Japanese Electrical Manufacturers Association (JEMA).

3.4.2 Cyclopentane and Other Hydrocarbons

Cyclopentane was introduced in 1993 as an alternative blowing agent for appliance foams in German refrigerators. Cyclopentane and other hydrocarbon blowing agents have zero ODPs and insignificant GWPs, but foams blown with these chemicals have higher insulation thermal conductivities compared to HCFC-141b which results in higher energy use for refrigerator-freezers. Foams blown with cyclopentane have no incompatibilities with other materials used in the refrigerator cabinet. These blowing agents would contribute to the volatile organic compound (VOC) emission inventory of industries that are controlled under the Clean Air Act.

Since hydrocarbons like cyclopentane are flammable, investment costs for changing a production line to HC use and implementing the manufacturing and production controls to ensure safe operation are higher than those associated with non-flammable technologies. After the manufacturing process, foams blown with HCFC-141b and cyclopentane have been reported to have similar flammability characteristics.

Conflicting data has been reported for the relative aging characteristics of HC versus HCFC or HFC blown insulating foams (Christian 1997, Heilig 1995).

3.4.3 Vacuum Panels

Vacuum insulated panels are the leading alternative technology to blown polyurethane foam for improving the cabinet insulation of refrigerators. These panels can be formed by using air and moisture tight metal or plastic barriers to contain a low conductivity filler material such as diatomaceous earth, aerogels, or open cell polyurethane foam. Selection of materials by appliance manufacturers will depend on material and panel manufacturing costs, weight, and proven performance. For this study, panels are assumed to be used in the door, each wall, and the top of the freezer cabinet, centered in each surface. Panels are assumed to extend to within 4 cm (1.57 in.) of each edge resulting in approximately 80% surface coverage for the freezer.

The annual energy use for a U.S. refrigerator with composite vacuum panels covering 80% of the exterior surface area of the freezer compartment was taken from the actual measured performance of a 20 ft^3, top-mount refrigerator tested at ORNL. The results indicate about an 8.5% performance improvement measured for a cabinet with evacuated panel enhancements to the freezer over a baseline cabinet prepared without panels (Vineyard 1997). TEWI calculations for U.S. refrigerators with composite wall construction are based on constant overall wall thickness; 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 approach was not considered in this study. The actual performance of evacuated panel insulation is very strongly affected by the dimensions of the panels because of the edge effects.

Evacuated panels may be a viable option to be considered in the 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 can be made cost-effective and have acceptable product lifetimes.

3.5 ALTERNATIVE TECHNOLOGIES

3.5.1 Absorption Refrigeration

More than a million thermally activated ammonia/water absorption refrigerators are manufactured and sold annually in world-wide markets. These products are commonly used in recreational vehicles because of their ability to use bottled gas as an energy source and in hotel rooms because of their quiet operation. These products are limited in size because of constraints imposed by the reliance on present day bubble pumps in their operating cycle and cannot be scaled up to displace conventional vapor-compression products. Recently reported research results indicate that better utilization of waste heat from the refrigerator rectifier and improvement in the design of the generator can improve the thermal COP of these refrigerators by as much as 50% (from a COP of 0.2 to 0.3) without degrading cooling capacity (Chen 1996). While the unique capabilities and operating characteristics will continue to allow absorption refrigerators to fill niche markets, low efficiencies and inherent design limitations make it very unlikely that they will be broadly adopted as replacements for vapor compression refrigerators.

3.5.2 Thermoelectric Refrigeration

Data for commercially available thermoelectric modules indicate they have a COP of approximately 0.32 at the same operating conditions as those used to rate compressors for refrigerator-freezers (Fischer 1994). Some commercial products like portable coolers and small drink cooler cabinets where portability and convenience of operation are important considerations use thermoelectric refrigeration systems. But this alternative technology is unlikely to replace vapor compression refrigeration for the broader domestic refrigerator market because of its extremely low efficiency.

No TEWI results were calculated for absorption refrigerators and thermoelectric refrigerators in this report.

3.6 ASSUMPTIONS

Refrigerants HFC-134a and iso-butane (HC-600a) or mixtures of hydrocarbons were assumed to give equivalent performance in the refrigeration circuit for all regions. Measurements of actual compressor performance with the two refrigerants need to be used to make a more definitive comparison.

3.6.1 North American Refrigerator-Freezer

TEWI calculations for a North American refrigerator-freezer design are based on a 510 liter (18 ft^3) frost-free (automatic defrost) model. TEWI calculations used refrigerator energy consumption data at the standard U.S. rating test; 32.2EC (90EF) room temperature, -15EC (5EF) freezer temperature, and 3.3EC (38EF) fresh food temperature as reported in the 1996 Directory of Certified Refrigerators and Freezers (AHAM 1996). 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 energy use for the baseline refrigerator/freezer using HCFC-141b blown foam and HFC-134a for the refrigerant is 646 kWh/year (1.77 kWh/day). These data are summarized in Table 1.

Table 1. Baseline North American refrigerator/freezer.

Parameter	Value
internal volume	510 liters
foam volume	0.267 m^3
refrigerant charge	0.155 kg HFC-134a
annual energy use	646 kWh/year (AHAM 1996)
blown foam insulation	1.08 kg HCFC-141b
energy use/unit internal volume	1.27 kWh/y/liter

Table 2. Baseline European refrigerator/freezer.

Parameter	Value
internal volume	230 liters
foam volume	0.1539 m^3
refrigerant charge HFC-134a HC-600a	 0.127 kg 0.045 kg
annual energy use HCFC-141b foam cyclopentane/pentane foam	 300 kWh/year 328 kWh/year
blown foam insulation HCFC-141b foam cyclopentane/pentane foam	 0.520 kg 0.279 kg
energy use/unit internal volume HCFC-141b foam cyclopentane/pentane foam	 1.30 kWh/y/liter 1.42 kWh/y/liter

3.6.2 European Refrigerator/Freezer

The typical European refrigerator chosen for this analysis is assumed to be a 230 liter (8 ft^3) model with a small freezer compartment. Reported energy use from

European manufacturers at rating conditions was used as a starting point for TEWI calculations. Typical temperatures for these tests are set at 25EC (77EF), -18EC (-0.4EF), and 5EC (41EF) for the room, freezer compartment, and fresh food compartment. Freezer door and wall thicknesses for this refrigerator would typically be 6.75 cm (2.66 in.), while the door and wall thicknesses for the fresh food section are about 3.50 cm (1.38 in.) and 4.50 cm (1.77 in.). TEWI calculations were performed for cyclopentane, HCFC-141b, HFC-245fa, and HFC-134a blown foams and HFC-134a or iso-butane (HC-600a) as refrigerants. The energy use for the baseline refrigerator-freezer using HFC-134a or HC-600a as the refrigerant and HCFC-141b blowing agent is taken as 300 kWh/year (0.82 kWh/day) or 1.31 kWh/y/l. The energy use for a baseline European refrigerator-freezer with either HFC-134a or HC-600a as the refrigerant and cyclopentane as the insulating foam blowing agent is assumed to be 328 kWh/year (0.90 kWh/day) or 1.42 kW/y/l. This performance difference between the HCFC-141b and cyclopentane cabinets is the result of the increased thermal conductivity of cyclopentane and assumed equivalent wall thickness of the appliance insulating foam. These data are summarized in Table 2.

3.6.3 Japanese Refrigerator-Freezer

The baseline Japanese refrigerator use for this TEWI evaluation is assumed to be a 355 liter (12.5 ft^3) automatic defrost model with an annual energy use of 522 kWh/year (Hara 1996a). The cabinet is divided into three compartments; one for frozen foods, one for cold storage, and the third for produce and vegetables (approximately 25%, 50%, and 25% of the refrigerated volume, respectively). The cabinet is foamed with 47 mm (1.85 in.) of HCFC-141b blown foam in the freezer walls and door. The walls and door of the fresh food compartment have 35 mm (1.38 in.) of insulation (same as used for the TEWI-1 report). The annual energy consumption given above is based on a refrigerator-freezer using HCFC-141b blown foam and 0.14 kg of HFC-134a as the refrigerant. Data for the Japanese refrigerator are summarized in Table 3.

Table 3. Baseline Japanese refrigerator/freezer.

Parameter	Value
internal volume	355 liters
foam volume	0.1796 m^3
refrigerant charge	0.140 kg HFC-134a
annual energy use	522 kWh/year
blown foam insulation	0.71 kg HCFC-141b
energy use/unit internal volume	1.47 kWh/y/liter

3.7 METHODOLOGY

Rather than relying on a refrigerator computer model to generate annual energy use, the foregoing baseline values for "typical" U.S., European, and Japanese refrigerators at operating conditions reflecting each countries' rating and certification standard conditions were chosen as the starting point for this analysis. These rating point energy consumption values were assumed to be representative of field operating conditions for each location. Alan Meier has shown that energy use values generated using the standard DOE test for U.S. refrigerator-freezers was a reasonably good predictor of field energy use (Meier 1993). An indication of how the same refrigerator would perform if tested according to the standard rating procedures for different countries can be estimated from results presented in a recently published comparison (Bansal 1995).

The relative energy consumption of refrigerator cabinets prepared using insulating foams blown with alternatives to *HCFC-141b* or *cyclopentane* was taken whenever possible from research results presented by the AHAM Appliance Research Consortium (ARC) Insulation Technical Advisory Committee (ITAC) at recent conferences and the parallel Japan Electrical Manufacturers Association (JEMA) Home Appliances Department program (Haworth 1996, Araki 1996). No internally consistent energy consumption data for the various combinations of refrigerants and blowing agents was found for the three- and four-star refrigerators marketed in Europe. “Baseline” energy consumption results were averaged from data provided by S. Brasch, M. Verhille, and H. Lotz (1996). Baseline energy consumption values for “typical” refrigerators in North America, Europe, and Japan were scaled in proportion to measured energy consumption increases or decreases published in the AHAM or JEMA reports for cabinets prepared with insulating foams blown with alternatives to HCFC-141b or cyclopentane. This adjusted annual energy use , the appliance lifetime , and the kg CO_2/kWh electric power plant emission rate were used to calculate the *indirect* TEWI for each refrigerant/blowing agent combination.

3.8 DISCUSSION

No attempt has been made to evaluate or balance all the advantages or disadvantages of the HFC- and hydrocarbon-based refrigerators. One has an impact from direct emissions of blowing agent and refrigerant, the other uses flammable materials and has a higher energy consumption but no direct impact on TEWI. Small differences in TEWI between the fluorocarbon- and hydrocarbon-based refrigerators must be weighed against the risks associated with the hydrocarbons. Japanese and American risk analyses have questioned the safety of large, automatic defrost refrigerators using hydrocarbons (Deloitte & Touche 1996).

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 low boiling liquid blowing agents (e.g. 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 gaseous blowing agents (e.g., HFC-134a, HCFC-22/HCFC-142b) over the range of temperatures relevant to a household refrigerator/freezer. This variation of k-factor with temperature has not been taken into account in any of the calculations in this study.

Where IPCC listed GWPs were not available for some of the newly developed blowing agents, values estimated by AFEAS member companies were used to calculate the direct TEWI. Considerable controversy exits on how accurate GWP values are calculated. IPCC in their 1995 report chose to use compound atmospheric lifetimes corresponding to tropospheric degradation mechanisms alone. When both tropospheric and stratospheric destruction mechanisms are considered, overall lifetimes and the resulting GWP value for the chemicals in question are considerably decreased. To be consistent, GWPs chosen for these calculations were taken from values listed in the 1995 IPCC report. Whenever values were not available in this listing, AFEAS estimates were used. Tabulated values for refrigerant and blowing agent GWP are given in Tables 23 and 24 in Appendix B.

3.9 RESULTS AND CONCLUSIONS

Detailed tabulations of the TEWI calculations for refrigerator-freezers are given in Appendix C for all regions. Key results are summarized below.

3.9.1 North American Refrigerator-Freezer

Energy use for North American refrigerator-freezers with various blowing agents are listed in Table 4. These results and data for foam density, weight percent blowing agent, and refrigerator dimensions

Table 5. Calculated energy use for refrigerator/freezers using alternative refrigerants and blowing agents (kWh/day).

	North America		Europe		Japan	
Blowing Agent / Refrigerant	HFC-134a	HC-600a	HFC-134a	HC-600a	HFC-134a	HC-600a
HCFC-141b	1.77*	1.77	0.82*	0.82	1.43*	NC
HCFC-22/HCFC-142b (60/40)	1.86	1.86	NC	NC	1.43	NC
HFC-134a	2.01	2.01	0.85	0.85	NC	NC
HFC-245fa	1.79	1.79	0.82	0.82	1.43	NC
HFC-365mfc	1.93	1.93	NC	NC	NC	NC
HFC-356mffm	1.84	1.84	NC	NC	1.48	NC
HFC-236ea	1.85	1.85	NC	NC	1.49	NC
Cyclopentane	1.95	1.95	0.90	0.90*	1.57	1.57
CO_2	1.98	1.98	NC	NC	NC	NC
HFC-152a	NC	NC	NC	NC	1.67	NC
HFC-245ca	NC	NC	NC	NC	1.50	NC
Vacuum Panels	1.62	1.62	NC	NC	NC	NC

* baseline values

were used to compute the TEWI for each combination of insulation and refrigerants shown in Fig. 4. TEWI are calculated using 100 year GWPs, a 15 year service life for the appliance and an average CO_2 emissions rate of 0.65 kg CO_2/kWh. The refrigerator using HCFC-141b for cabinet foam shows 12% of the TEWI due to fluorocarbon emissions. HFC blowing agent alternatives for HCFC-141b show from 11 to 15 % of TEWI due to chemical emissions. As an alternative to HCFC-141b, refrigerator foams blown with HFC-245fa give the best balance of energy efficiency and lowest TEWI. With the exception of cyclopentane base insulations, almost all of the direct effect from refrigerators is due to the foam blowing agent. Assumptions on mandatory recovery of the refrigerant would result in marginal, 2 to 3% reductions in the TEWI.

An 8 to 9% improvement in TEWI is indicated for a refrigerator with vacuum panels incorporated into the freezer walls. Essentially all of this improvement is due to a reduction in the indirect TEWI contribution resulting from decreased energy use. A cost/benefit analysis performed on this vacuum panel freezer enhancement indicated a 36 year consumer pay-back period and concluded that other readily available energy saving options were more promising (Vineyard 1997).

Essentially no interest remains in using HCFC blends like HCFC-22/HCFC-142b or HCFC-22/HCFC-141b as blowing agents because of the impending phaseout dates for chlorine containing refrigerants and blowing agents. Despite low GWP values, the use of CO_2 alone as a blowing agent for appliance foams is not considered viable because the high initial thermal conductivity and rapid aging of foams blown with this gas (Deloitte & Touche 1996).

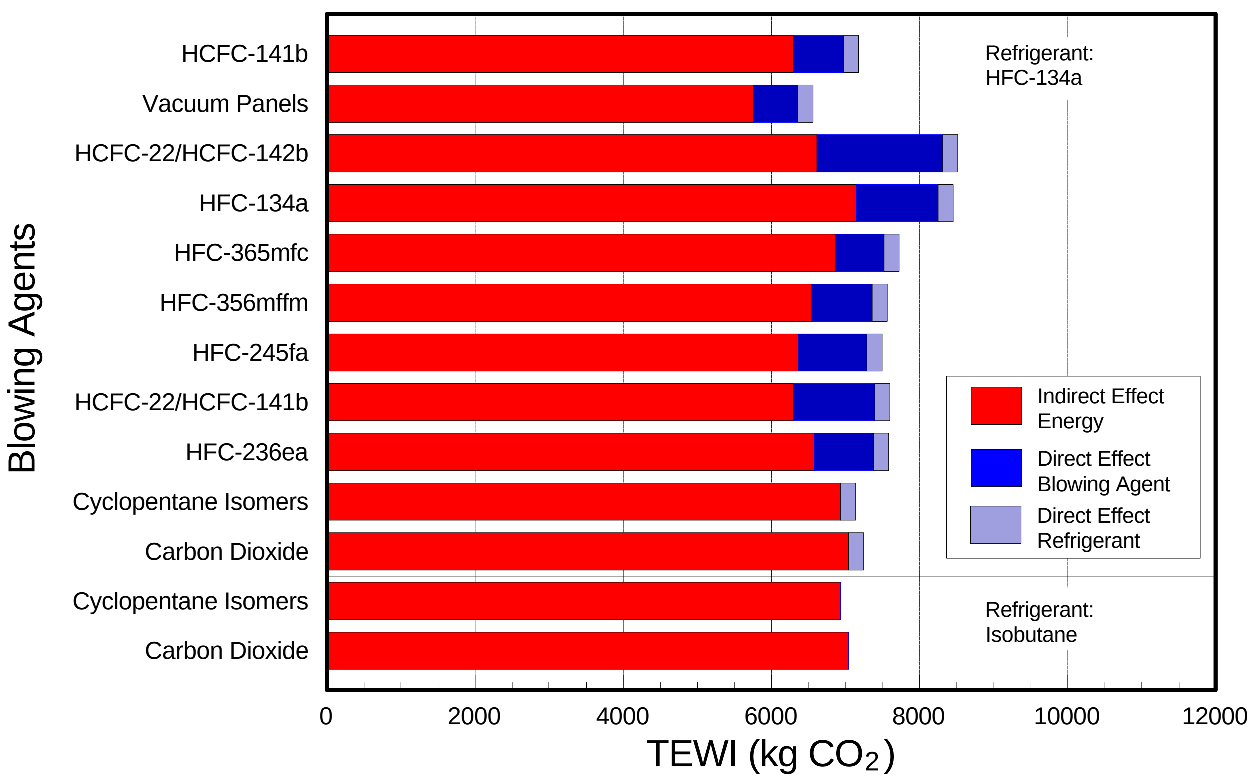

Figure 4. TEWI results for a 510 liter (18 ft^3) North American, top-mount refrigerator/freezer insulating foam options.

3.9.2 European Refrigerator-Freezer

Baseline and projected daily energy use for the European style refrigerator/freezer are also contained in Table 4. All refrigerators with cabinet insulation blown with cyclopentane or pentane isomers are estimated to use 9 to 10% more energy than similar models using HCFC-141b as the foam blowing agent (Haworth 1996, Deloitte & Touche 1996). Use of HFC-134a as a blowing agent in Europe is decreasing because of poor thermal conductivity results and the availability of other blowing agents (Deloitte & Touche 1996).

TEWI results shown in Fig. 5 for European refrigerators are calculated using a 15 year service life for the appliance and CO_2 emission rate for electricity generation of 0.470 kg CO_2/kWh. These results show the same trends as those in Fig. 4 for the North American refrigerator/freezer, although the percentage of direct effects due to fluorocarbon emissions is higher (# 30%), primarily because of the lower CO_2 emissions for the power generation mix in Europe (higher percentage of nuclear and hydroelectric power generation with virtually zero CO_2 emissions). The lower rate of CO_2 emissions for power generation results in a relatively smaller indirect effect, a smaller overall TEWI, and hence a larger percentage direct effect than the North American refrigerators. Refrigerant recovery at end of life would reduce TEWI's for the HFC-134a options by 5-6%, making them equal to those of the HC-600a refrigerators.

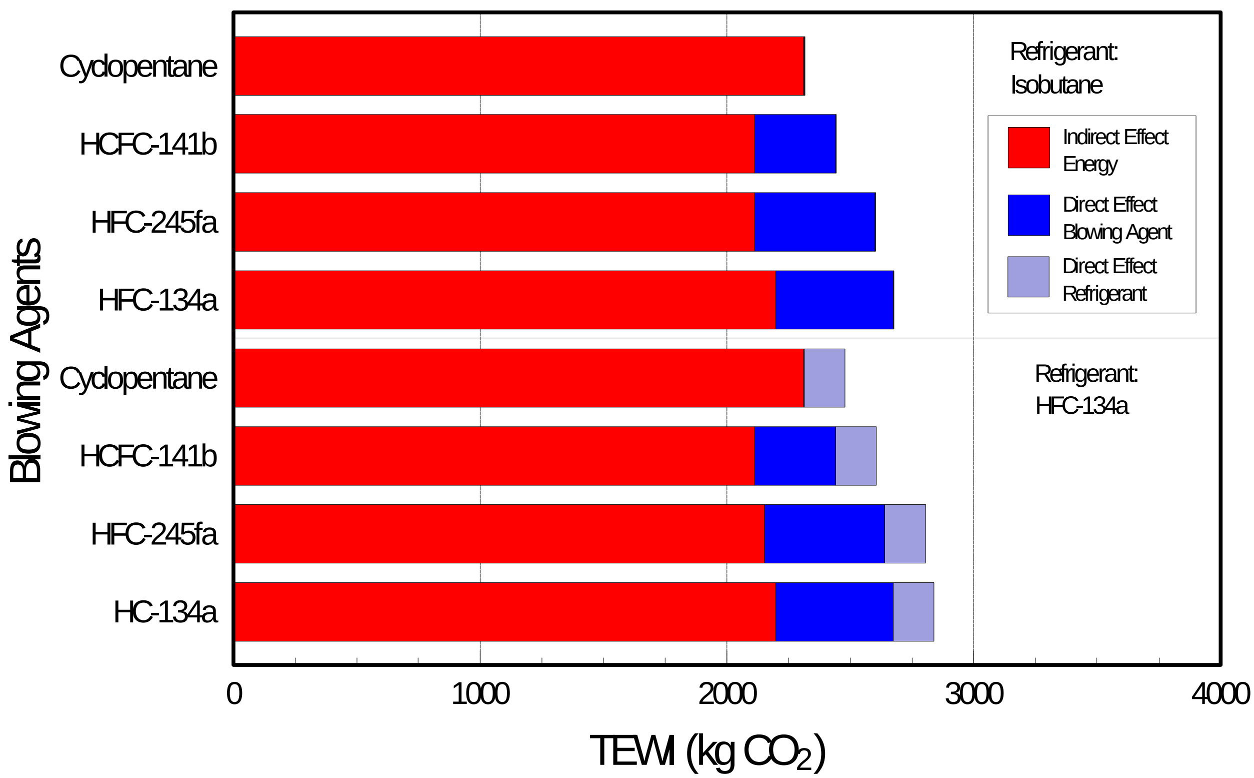

Figure 5. TEWI results for a 230 liter European refrigerator/freezer versus cabinet insulating foam options.

3.9.3 Japanese Refrigerator-Freezer

Projected daily energy use for a 355 liter (12.5 ft^3) Japanese refrigerator-freezer is shown with the North American and European refrigerators in Table 4. TEWI results presented in Fig. 6 are calculated using a 15 year service life for the appliance and a 0.473 kilogram of CO_2 per kilowatt-hour power plant emission rate for electricity in Japan. No recovery of the blowing agent or refrigerant is assumed. The relative performance of cyclopentane blown appliance foams for these refrigerators was extrapolated from the AHAM ITAC energy consumption results for North American refrigerators and confirmed with JEMA representatives, with the exception of the HCFC-22/HCFC-142b blown foam (Hara 1996b). The direct TEWI contribution from fluorocarbon blowing agents and refrigerant are about 5 to 15% of the total TEWI (Fig. 6).

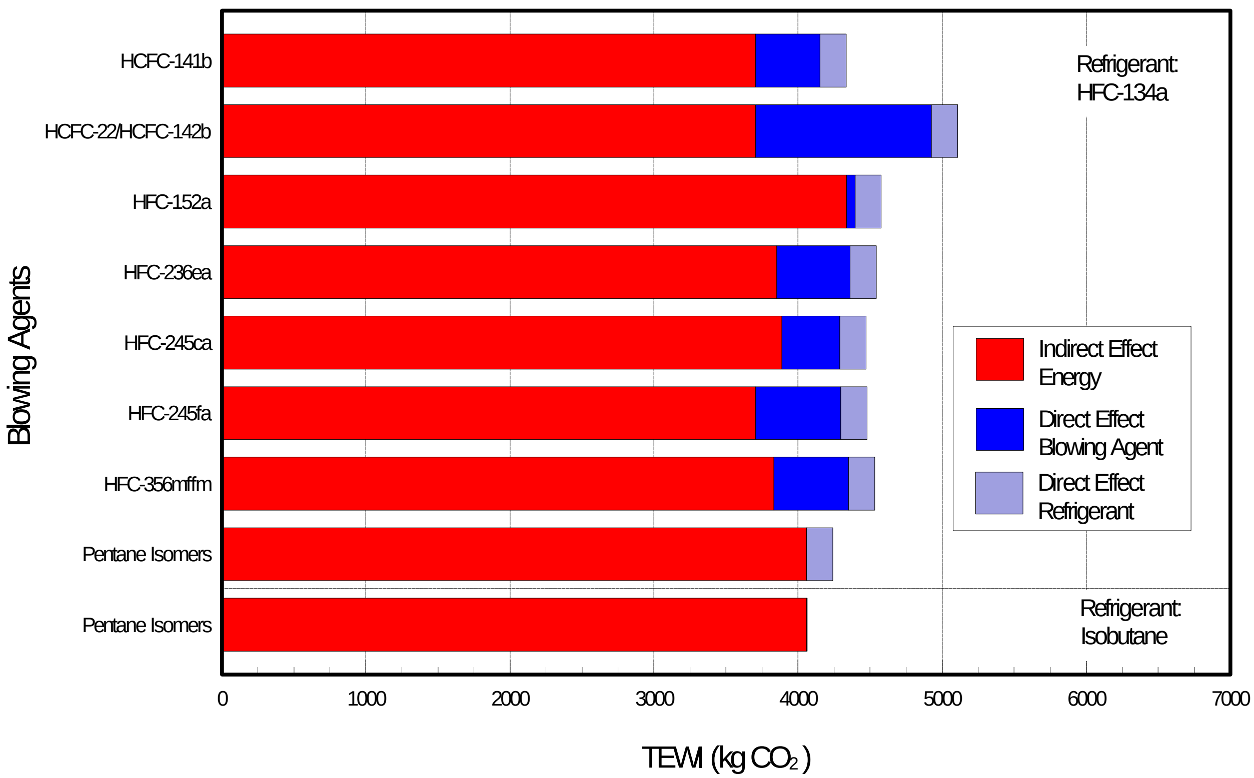

Figure 6. TEWI results for a 355 liter Japanese refrigerator/freezer vs. cabinet insulating foam options.

Despite the equivalent insulating characteristics reported by JEMA for HCFC-141b foam and foams blown with a 60/40 mass% HCFC-22/HCFC-142b mixture, the larger GWP of the HCFC blend results in an 18% greater overall TEWI. The low GWP for HFC-152a is not enough to overcome the poor performance of foams blown with this fluorocarbon, so no TEWI advantage is seen over refrigerators using foams blown with HCFC-141b, HFC-236ea, HFC-245ca, HFC-245fa, or HFC-356mffm which have larger GWPs.

No clear TEWI advantage is shown for refrigerator-freezers using HC-600a as the refrigerant and cyclopentane or hydrocarbon blown insulations because of the diminished thermo-insulating

No clear TEWI advantage is shown for refrigerator-freezers using HC-600a as the refrigerant and cyclopentane or hydrocarbon blown insulations because of the diminished thermo-insulating performance seen with these foams and the corresponding increase in indirect TEWI.

Assumptions on recovery of the refrigerant would result in a 3-4% reduction in the TEWI, which would essentially equate the calculated TEWIs of the HFC and HC refrigerators.

Basic research on recovery of the blowing agent is being conducted in Japan. If the technology is established and foam blowing agent is partially recovered, the direct TEWI of refrigerators with HCFC or HFC blown insulating foams could decrease by . 10%, but indirect TEWI would be increased in proportion to the amount and type of energy utilized in this recovery process.

3.10 CONCLUSIONS

Much of the R&D effort associated with improving the environmental acceptability of refrigerator-freezers has been directed at improving the efficiency of the vapor compression refrigeration cycle through more efficient circuit components (Vineyard 1997). No alternative cooling technology appears ready to displace vapor compression for domestic refrigeration. Vacuum panels can be used to improve cabinet insulation and decrease TEWI but current costs make this approach unattractive compared to other options. The use of HC refrigerants and foams in refrigerators and freezers as compared to HFC-134a refrigerant with HFC foams results in estimated TEWI reductions of about 6% in North America, 9% in Japan, and 18% in Europe under the baseline assumptions used in this analysis. If the HFC-134a is assumed to be fully recovered at disposal of the refrigerator, TEWI reductions for the HC option become about 4% in North America, 5% in Japan, and 13% in Europe. Use of HCs in these products also raises additional safety concerns and has resulted in higher unit costs (Deloitte and Touche, 1996).

Consistent improvements in refrigerator efficiencies lead to lower TEWIs but accentuate the *direct* contribution. Energy consumption estimates for the HFC-based refrigerator alternatives average about 10% lower than those of the HC alternatives in all regions.

A concerted research effort has been directed to finding an alternative blowing agent for HCFC-141b which contains chlorine and will be phased out of use in 2003. Some general conclusions relating to refrigerator-freezer applications that can be drawn from this effort are listed below:

! Several of the alternative HFC blowing agents being considered give insulating foams comparable to HCFC-141b, but have higher GWPs than HCFC-141b which result in larger direct TEWIs relative to the HCFC-141b baseline. The alternative HFC blown foams have increased energy efficiency compared to hydrocarbon blown foams which gives a lower indirect contribution to TEWI and offsets some of the differences in direct effects.

! HCFC-141b and all the alternative HFC blowing agents with the exception of HFC-134a have produced appliance insulating foams with better k-values than HC blown foams.

- In this comparison, insulating foams resulting in refrigerators with better energy efficiencies may be favored because decreased electrical energy use has other long range environmental implications.

Uncertainties, as discussed in the executive summary, are embedded in the parameters that factor into these calculations. Further, as stated in the opening paragraphs of the executive summary, TEWI should not be considered the sole determining factor in deciding between technology options for this or any other application. Not withstanding it must be noted that the 13-18% estimated TEWI reduction from HC use in refrigerators for Europe should be considered significant. It must also be noted that the lesser HC TEWI advantage estimated for the North American and Japanese cases is not considered to be significant. Substantial added costs would be needed for production safety and environmental concerns at the factories and for reducing/eliminating spark sources in the products themselves to produce a frost-free refrigerator (typical of the major type used in the latter two regions) using hydrocarbons. These higher costs could be applied to HFC designs (to incorporate vacuum insulation in the cabinet walls, for instance) to yield a product with potentially superior TEWI.

4. UNITARY AIR CONDITIONING

4.1 INTRODUCTION

Estimates of the installed unitary air conditioning capacity worldwide range from 763x10^3 kW (217x10^3 tons) (Fischer 1991) to 1,450x10^3 kW (410x10^3 tons) (UNEP 1995) exceed the worldwide installed chiller capacity by at least a factor of four. Over 90% of this unitary equipment is in North America and Japan, although air conditioning is becoming more common in Europe and other countries. By its broadest definition, the unitary term encompasses all equipment that cools or heats building air via direct heat exchange with a refrigerant coil or indirect heat exchange through a hydronic loop. It applies to air conditioners which only cool and heat pumps which can only heat or heat and cool. The refrigeration capacity of unitary systems can range from 2 to 420 kW (0.6 to 120 refrigeration tons). As indicated in Fig. 7, the efficiency of U.S. central air conditioning products has steadily increased over the last twenty years with slight spurts in performance brought on by governmental regulation and market competition (ARI 1996). Improvements in performance and reliability of this product over the years have not been matched by an increase in constant dollar cost to the consumer because the market for unitary equipment is highly competitive. Alternative refrigerants for this market segment or alternative technologies must deliver equivalent or better performance than current systems with similar or lower costs in order to compete effectively with existing technologies.

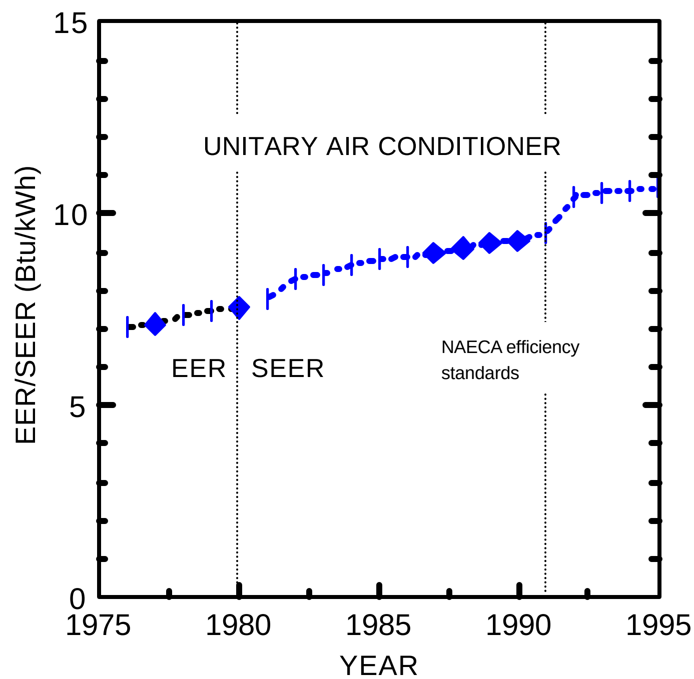

Figure 7. Efficiency ratings of unitary air conditioners in the United States (sales weighted average).

4.2 TECHNOLOGY DESCRIPTION

Ducted air-to-air residential systems using HCFC-22 constitute most of the North American unitary market. A compressor and heat exchanger located outside of the building supply refrigerant to a single indoor coil. Building air is cooled or heated by the indoor coil and distributed throughout the building via ducts. UNEP estimates that 59 million ducted, air conditioning systems are installed in North America and world wide. Due to differences in building practices, ductless split systems are more popular in Japan and many other countries. In ductless systems, a compressor and heat exchanger are installed outside the conditioned space and refrigerant is distributed to one or more fan coils inside the building. There may be one refrigerant-to-air fan coil unit for each room. This ductless approach requires a larger refrigerant charge per unit of capacity, usually 0.32 to 0.34 kg of HCFC-22 per kW as compared to a centralized ducted system average of about 0.26kg per kW. An estimated 80 million ductless units are installed world wide.

Because of the wide diversity of systems covered by this category, TEWI calculations in this chapter are limited to 10.5 kW (3.0 ton) ducted residential systems and 26.4 kW (7.5 ton) ducted roof top units in North America; a 2.8 kW (0.80 ton) room air conditioner/heat pump and 14.0 kW (4.0 ton) ductless, package air conditioner/heat pump in Japan; and a residential heating or cooling-only system and commercial heating/cooling unit in Europe defined only by seasonal COPs, relative refrigerant efficiencies, and annual loads (IEA 1994).

4.2.1 Refrigerants

HCFC-22 is the refrigerant used in virtually all unitary equipment because of its inherent efficiency and high refrigeration capacity. Provisions of the Clean Air Act (CAA) in the United States call for HCFC-22 to be phased out of “new equipment” by 2010, and allow production of smaller amounts of the refrigerant until 2020 for servicing installed equipment. This date has prompted a flurry of research and development work at universities, research institutions, and by refrigerant manufacturers and HVAC companies to find suitable HCFC-22 replacements.

No single-component refrigerant or blend has been identified which can successfully replace HCFC-22 in all of these unitary applications. The Air-Conditioning and Refrigeration Institute (ARI), through its Alternative Refrigerants Evaluation Program (AREP), led an international effort to measure, evaluate, and disseminate data on potential HCFC-22 and CFC-502 alternatives. The primary goal of the program was to measure the performance of alternatives under conditions indicative of the refrigeration and air conditioning application currently served by these refrigerants. Issues such as flammability, toxicity, fractionation, etc. were not addressed. Based on testing performed under AREP, several alternative refrigerants were identified which gave similar or slightly improved performance when compared to HCFC-22 (Godwin 1994). The program also clearly indicated that no single alternative studied was superior to HCFC-22 in <u>all</u> of its current applications, and that several refrigerants or blends of refrigerants may be needed to fill the void left when HCFC-22 is phased out of production.

TEWI results from previous reports have indicated that the direct GWP of the refrigerant used for unitary equipment contributes less than 10% of the total TEWI for these products depending on what assumptions are used for the analysis, and that the direct GWP of the refrigerant is less important than the overall efficiency of the unitary system (Fischer 1991) (Fischer 1994). This indicates that any refrigerant or refrigerant blend proposed as an alternative for HCFC-22 must provide good cycle efficiency in addition to a low to moderate GWP.

4.2.1.1 HFC-134a

HFC-134a is one commercially available refrigerant initially considered by AREP as an alternative candidate for HCFC-22. It is a single component refrigerant that has been widely adopted by the domestic refrigeration, automotive air conditioning, and chiller air conditioning market segments, so the major refrigerant manufacturers have constructed manufacturing plants and have extensive product application literature. HFC-134a has a 40% lower refrigeration capacity than HCFC-22 under unitary operating conditions and has shown a 5% decrease in efficiency for typical unitary applications, so it cannot be considered as a “drop-in” replacement. Refrigerants with markedly lower cooling capacities than HCFC-22 are at a disadvantage because losses in refrigerant volumetric capacity necessitate larger compressors and heat exchangers to maintain system capacity. These translate into higher equipment costs to the consumer.

HFC-134a is not expected to be widely used in U.S. or Japanese unitary equipment. Some heating-only, HFC-134a based units are included in the TEWI results for European unitary equipment with the assumptions that it was used in units specifically designed for this refrigerant, and that it performed at an efficiency equal to equipment using HCFC-22 (Fischer 1994). With the current information on prospective HCFC-22 alternatives and the increasing emphasis on more efficient cooling performance, it is unlikely that HFC-134a will be extensively used in unitary air conditioning or heat pumps (UNEP 1995).

4.2.1.2 HFC Mixtures

The most likely replacements for HCFC-22 which came out of the AREP program are binary or ternary mixtures which are ozone-safe, non-flammable, non-toxic, efficient, and have performance levels close to that of HCFC-22.

R-407C

R-407C is a 23/25/52 mass % blend of HFC-32, HFC-125, and HFC-134a. It showed equivalent capacities to HCFC-22 but efficiencies averaged about 5% lower in soft-optimized equipment (Godwin 1994). R-407C is a zeotropic blend which will fractionate or change composition during evaporation and condensation in vapor compression refrigeration applications and will show about a 5° C (9° F) change in temperature (temperature glide) across the heat exchangers due to this composition change. This heat exchanger temperature glide and tendency to fractionate make zeotropes less attractive commercially. It is a departure from the isothermal phase change behavior of pure refrigerants to which the industry has become accustomed, and system leaks with zeotropes may result in composition changes making service and repair more difficult.

Testing of R-407C in laboratory and soft-optimized commercially produced equipment has established capacity and system efficiency levels relative to HCFC-22 which allow TEWI evaluations for unitary equipment (Hwang 1995, Murphey 1995, Judge 1995, Berglof 1996, Linton 1996).

R-407C has an ASHRAE safety classification “A1/A1” which designates it as a commercially available refrigerant with low toxicity and no flame spread.

R-410A

R-410A is a mixture of HFC-32 and HFC-125 with a 50/50 mass % composition. AREP found the blend’s capacity was essentially the same as HCFC-22 given compressors appropriately sized for the difference in volumetric capacities of the two refrigerants. The ARI results also indicated that system cooling efficiencies averaged from 1% to 6% higher than HCFC-22 (Godwin 1994). Extensive testing of R-410A has also occurred subsequent to the AREP reports (Hwang 1996, Murphey 1995, Feldman 1995, Linton 1996). The latest results from these tests indicate that R-410A can be used in redesigned unitary equipment with no decrease in system capacity and a 4 to 7% increase in system efficiency. Most of this system efficiency gain is attributed to improved thermophysical properties of the blend over HCFC-22. System efficiency results from this latest series of tests were used for the TEWI results presented here.

R-410A is considered a “near azeotrope” in that it does not fractionate during a phase change in refrigeration equipment. One drawback of the mixture is that it has a system operating pressure approximately 50% higher than HCFC-22 so it cannot be considered as a drop-in replacement or for retrofit into existing unitary systems. Design changes will be required to accommodate these higher operating pressures. Another more subtle drawback of this HFC mixture is a critical temperature significantly lower than that of HCFC-22 (73.3E C for the HFC blend versus 96.1E C for HCFC-22). This could diminish efficiency relative to HCFC-22 at higher condensing temperatures. R-410A has an “A1/A1” ASHRAE safety classification.

4.2.1.3 Propane

Propane (R-290) can be a good refrigerant and it is attracting attention as an alternative to HCFC-22. The major disadvantage with propane, naturally, is that it is flammable Due to their flammability, hydrocarbon refrigerants like propane, are seriously considered in low-charge systems such as refrigerators, freezers, and packaged coolers (Stene 1996). The most significant use of hydrocarbons as working fluids is found in the United Kingdom and Germany. There is a strong reluctance on the part of manufacturers in the United States and Japan to expose customers to the hazards of flammable refrigerants in residential and commercial applications, which will make it difficult for propane to gain wide acceptance in these markets.

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). Part of this work involved a cost estimate for a 12 kW (3½ ton) unitary air conditioner using propane which came out be 30% higher than a comparable system using HCFC-22. 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 cost items are not required when using a non-flammable refrigerant. In addition, there are costs to the manufacturer associated with safe storage, charging, and handling of a flammable refrigerant in the plant. More recent evaluations on the feasibility of substituting propane for HCFC-22 in ducted residential air conditioners have also focused on the relative costs versus environmental benefits, an approach suggested by Kuijpers (1995). Three engineering solutions proposed to mitigate the flammability risk of propane are to: 1) prevent leakage and remove all sources of ignition as suggested by Treadwell above; 2) add a flame suppressant like HFC-227ea or a fluoroiodocompound to propane in sufficient quantities to make the mixture non-flammable; and 3) use a secondary loop to prevent the propane from entering residences (Douglas 1996, Keller 1996). Conclusions from these studies indicate that TEWI could be reduced if propane is used as a refrigerant in leak-tight air conditioner/heat pumps designed according to specifications suggested in option number 1 above, but that applying the additional costs to improve the efficiency of systems using non-flammable refrigerants would be more cost effective. The market for unitary equipment is very price sensitive and any cost premium would put air conditioners at a great disadvantage in free, competitive markets.

Several recent research reports indicated that propane is being substituted for HCFC-22 with slight (. 5%) increases in system efficiency in hydronic, heating only heat pumps commonly used in Europe (Lystad 1996, Rodecker 1996).

In this report, TEWI calculations are performed for propane in a ducted air conditioning/heat pump system which employs an intermediate heat exchanger and indirect loop to prevent flammable refrigerant from entering the conditioned space. Since the steady-state COP of propane and HCFC-22 are essentially equivalent, the TEWI for a propane system that does not use an intermediate loop or a mixed flame suppressant can be reasonably estimated from the indirect contribution for HCFC-22 systems.

4.2.1.4 Ammonia

Ammonia (R-717) is a good refrigerant which is likely to experience broader application as CFCs and HCFCs are phased out, but it is not a choice well suited for unitary equipment (Fairchild 1995). The chemical properties of ammonia present material compatibility problems not experienced with the fluorocarbon and hydrocarbon alternatives.

Residential and light commercial air conditioning systems are mass produced using copper for refrigerant tubing, in the heat exchangers and connecting components, using hermetic compressors whose electric motors employ copper wire windings, and using direct heat exchange evaporators. Ammonia is incompatible with copper in the presence of water, and 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 material 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 or other high-occupancy applications, 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. Ammonia also has high discharge temperatures that need to be reduced to avoid damaging the system; this can be done economically on larger refrigeration systems that are used in centralized cold storage or food processing applications but the cost could be prohibitive on small systems.

In view of the above considerations, it is not considered likely that ammonia will be used commercially in unitary equipment as a replacement for HCFC-22**.** No TEWI calculations are performed for unitary systems using ammonia as a vapor compression refrigerant. Ammonia-water absorption equipment options are considered.

4.3 ALTERNATIVE TECHNOLOGIES

Electric resistance heat is evaluated in appropriate locations to provide TEWI comparisons to heat pump options. Residential gas heating and cooling options are evaluated in a subsequent section.

4.4 ASSUMPTIONS

National averages are used for the power plant emission rates in these calculations. The averaged electrical power plant emission rates are 0.650 kg CO_2/kWh for the United States, 0.470 kg CO_2/kWh for Europe, and 0.473 kg CO_2/kWh for Japan , Appendix A. These CO_2/kWh emission rates are compiled from open literature data rather than calculated from the fundamental heat content of fuels, fuel mix used in power production, plant efficiencies, and transportation and distribution losses.

Published measurements for steady-state COP data relative to HCFC-22 and fixed values for the seasonal energy efficiency ratio (SEER) or heating seasonal performance factor (HSPF) of HCFC-22 were used to calculate SEERs and HSPFs for propane and the R-407C and R-410A mixtures. The relative efficiency values used for these calculations are summarized in Table 5. These values are based on recent test results in “soft optimized” systems. Further development of air conditioners specifically designed to use these alternative refrigerants could lead to more favorable comparisons relative to HCFC-22.

Propane’s performance for a 10.6 kW (3 ton) ducted unitary systems was attenuated by assuming an intermediate heat transfer loop to keep this flammable refrigerant out of the occupied space. A 2.8E C (5E F) or 5.6E C (10E F) lower evaporator or higher condenser temperature were assumed to drive this intermediate loop, resulting in lower cycle efficiencies. A parasitic pump load of one-fourth horse power was assumed for the 10.6 kW (3.0 ton) residential systems. Since hydronic heating systems with intermediate, secondary heat exchange loops are more common in Europe, the additional energy used and CO_2 generation associated with this intermediate loop was not applied to results for European residential heating.

Table 5. Relative efficiencies for alternative refrigerants for unitary equipment (relative to HCFC-22).

		Efficiency Relative to HCFC-22			
		1996		2005	
Refrigerant	Components (mass % composition)	Cooling	Heating	Cooling	Heating
HCFC-22	HCFC-22 (100%)	100%	100%	100%	100%
HFC-134a	HFC-134a (100%)	100%	99%	100%	99%
HC-290					
secondary hx at 2.8ECÄT	HC-290	85%	88%	85%	88%
secondary hx at 5.6ECÄT	HC-290	72%	77%	72%	77%
R-407C	HFC-32/HFC-125/HFC-134a (23/25/52)	NA*	NA*	100%	100%
R-410A	HFC-32/HFC-125 (50/50)	NA*	NA*	105%	105%

* not available

Fifteen year lifetimes are assumed for U.S. and European unitary equipment. Based on information provided by Hara from JICOP and Hisajima of JRAIA, the lifetime of the Japanese 2.8 kW (0.8 ton) split, room air conditioner/heat pump was taken as 12 years and the 14.0 kW (4.0 ton) packaged heat pump was set at 10 years (Hara 1996b).

Based on information assembled from ARI member companies, the maximum residential heat pump and air conditioner annual leak rates of 4% of the charge for 1996 equipment and 2% per year for equipment available in 2005 were used for the direct TEWI calculations (Hourahan 1996a). This same group recommended a 0.25 to 1.5% of charge annual loss rate for 1995 roof top equipment (1.5% was used) and a 0.25 to 1.0% rate for 2005 equipment (1.0% was used). Since roof top units require no field refrigerant line connections, there is less opportunity for leakage. An end-of-life (E.O.L.) charge loss rate of 15% was rationalized for both residential and roof top units on the basis of recovering 90% of the charge from 95% of the field units, but allowing for a 100% charge loss from about 5% of field stock (Hourahan 1996a). The 1996 annual leak or make up rate and 15% E.O.L. loss were also used for the European equipment calculations.

Hara and Hisajima recommended that an annual charge loss rate of 0.5% be used for the Japanese 2.8 kW (0.8 ton) split, room air conditioner/heat pump and 0.2% for the 14.0 kW (4.0 ton) packaged heat pump (Hara 1996a). End-of-life charge loss rates were taken as 15% of the total charge for both of these heat pumps (see Table 6).

4.5 METHODOLOGY

Table 6.. Performance data for unitary Japanese systems used for the TEWI calculations.

	Room Air Conditioner / Heat Pump System	Packaged Air Conditioner / Heat Pump System
Base System	HCFC-22; 2.8 kW split	HCFC-22; 5 PS split
COP Cooling Heating	 2.67 3.20	 2.50 3.00
Annual Equivalent Full Load Operating Hours (Tokyo) Cooling Heating	 350 450	 600 900
Charge Weight HCFC-22 (kg)	1.0	4.45
Annual Refrigerant Make Up Rate (rates suggested by Hara 1996a)	0.50%	0.20%
Charge Loss at End-of-Life	15%	15%
Equipment Lifetime	12 years	10 years
COP Relative to HCFC-22 R-407C R-410A	 --- 1.0	 0.95 0.95
Charge Weight Relative to HCFC-22 R-407C R-410A	 --- 1.0	 1.0 1.0

Total equivalent warming impacts were calculated for baseline 10.5 kW (36,000 Btu/h) heat pumps with SEERs of 10, 12, and 14; and corresponding HSPFs of 7, 8, and 9 with a refrigerant charge of 2.8 kg (6.2 lbs) of HCFC-22 for three locations in the U.S. In calculations where air conditioning is combined with some other heating technology, a central air conditioner with SEERs of 10 and 12 was used. SEERs and HSPFs for a geothermal heat pump were chosen from information provided by major manufacturers and results of standard rating/certification tests (ARI #330-93 1993). Seasonal energy use is computed based on a "typical" 1,800 sq. foot residence with a $74.7x10^6$ Btu/yr ($78.8x10^6$ kJ/yr) heating load and $16.1x10^6$ Btu/yr ($17.0x10^6$ kJ/yr) cooling load in Pittsburgh; a $34.8x10^6$ Btu/yr ($36.7x10^6$ kJ/yr) heating load and $33.8x10^6$ Btu/yr ($35.7x10^6$ kJ/r) cooling load in Atlanta; and a 0 Btu/yr (0 kJ/yr) heating load and $82.2x10^6$ Btu/yr ($86.7x10^6$ kJ/yr) cooling load in Miami (Ballou 1981).

For ducted residential equipment available in 2005, the baseline heat pump was assumed to have an SEER of 12 and a HSPF of 8 and the SEER of a central air conditioning unit was taken to be 12.

Cooling-only TEWI calculations for 26.4 kW (7.5 ton) roof top units with full load rating EERs of 10 for 1996 and 11 for 2005 with a HCFC-22 charge of 6.9 kg (15.2 lbs) were calculated for Pittsburgh with 600 equivalent full load cooling hours, Atlanta with 1,400 equivalent full-load cooling hours, and Miami with 2,700 equivalent full-load cooling hours. Annual energy use for these units was calculated based on their capacity (90,000 Btu/hr), the EER, and equivalent full load hours.

Residential heating and cooling-only TEWI calculations were performed for Europe using averaged loads and seasonal performance factors (SPFs), Appendix D. An average heating load for all European countries was used for heating TEWI calculations but only southern Europe (Greece) was used for the residential air conditioning calculations. Heating and cooling TEWIs for a composite European commercial building were calculated for heat pumps using averaged loads and heat pumps with various refrigerants and an HCFC-22 air conditioner used with electric resistance heat. The European average of 0.47 kg CO_2/kWh was used to convert electric use to CO_2 emissions with the exception of the residential air conditioning calculation which used Greece's emission rate of 0.976 kg CO_2/kWh.

TEWI results for Japanese unitary equipment were calculated using data from Table 6. This data was provide by Hara and Hisajima (Hara 1996b). The Japanese average of 0.473 kg CO_2/kWh was used to convert electric use to CO_2 emissions for these calculations.

Table 7. Current technology (1996) efficiency data.

System	Efficiencies Cooling / Heating
Air-to-Air Heat Pumps	
HCFC-22:	
minimum efficiencies	SEER-10 / HSPF-7
high efficiencies	SEER-12 / HSPF-8
Premium Technologies	
Air-to-Air Electric Heat Pump	SEER-14 / HSPF-9
Geothermal Heat Pump	SEER-15.8 / HSPF-12

Table 8. Future system efficiencies (2005).

System	Efficiency Cooling / Heating
Air-to-Air Electric Heat Pumps	
HCFC-22	SEER-12 / HSPF-8
R-407C	SEER-12 / HSPF-8
R-410A	SEER-12.6 / HSPF-8.4
HC-290	(SEER-12 / HSPF-8)
2.8EC ÄT in secondary loop	*SEER-10.9/HSPF-7.2*
5.6EC ÄT in secondary loop	*SEER-10.2/HSPF-6.2*
Premium Technologies	
Air-to-Air Electric Heat Pumps	SEER-14 / HSPF-9
Geothermal Heat Pumps	SEER-17.2 / HSPF-12.8

4.6 RESULTS AND DISCUSSION

Total equivalent warming impacts for various residential heating/cooling options were calculated for Pittsburgh, Atlanta, and Miami in the United States, and the results are shown in Figs. 8 to 13. These results are computed using the efficiency data in Table 7 and 8. Each figure has two sections, the upper portion shows "standard technologies," or heating/cooling options that represent baseline cost for a residential system

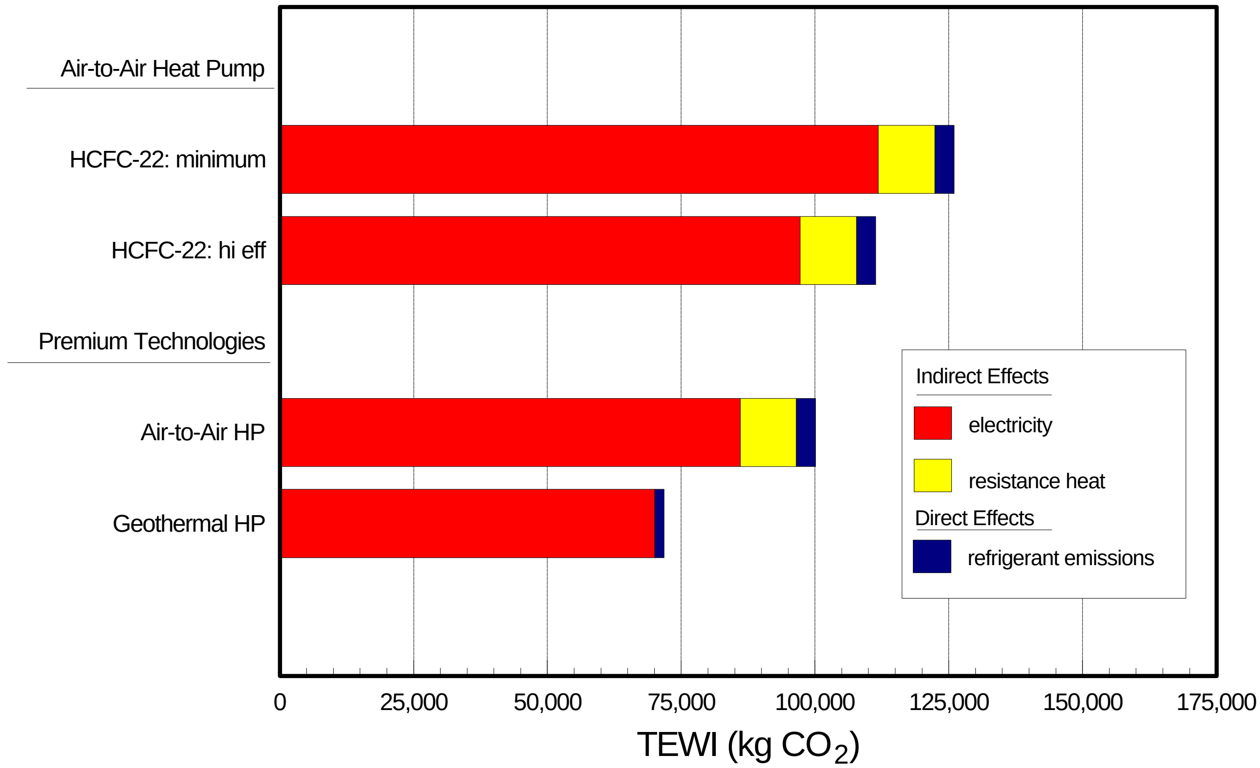

Figure 8. TEWI for residential heating and cooling options in Pittsburgh, Pennsylvania USA (1996 efficiency levels).

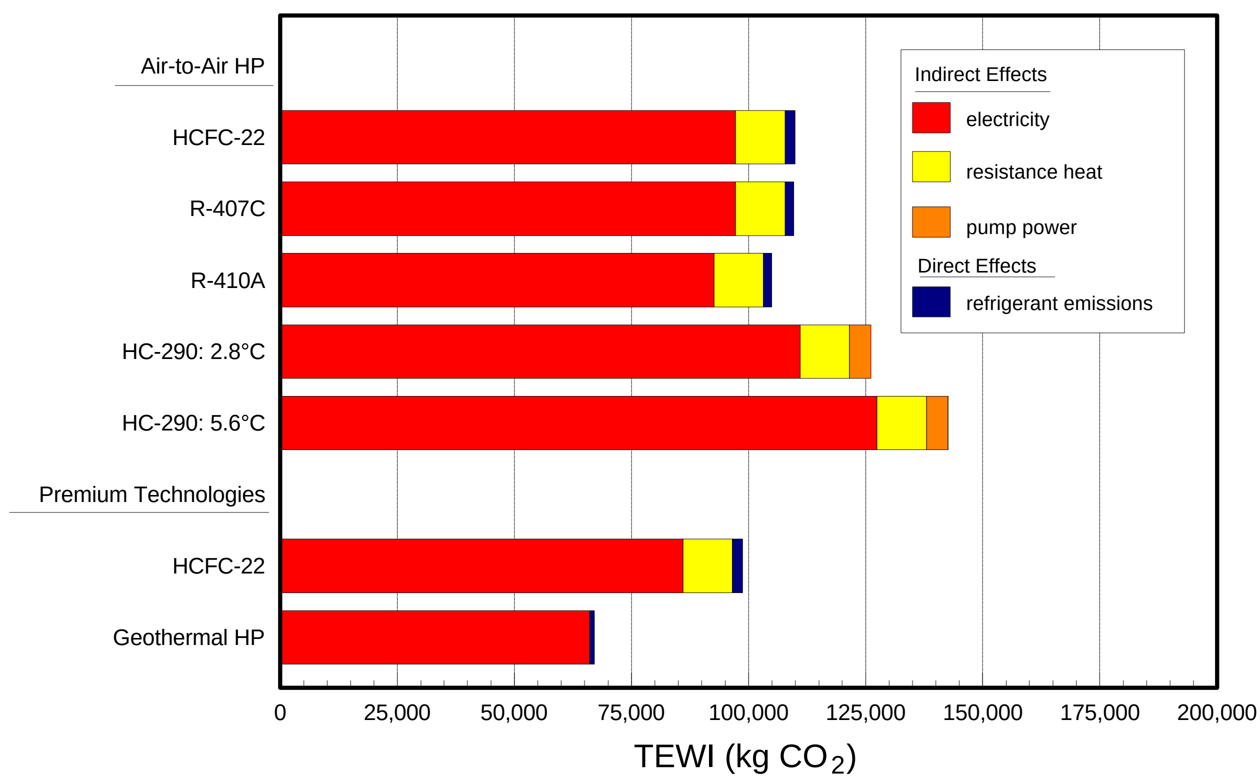

Figure 9. TEWI for residential heating and cooling options in Pittsburgh, Pennsylvania USA (2005 efficiency levels).

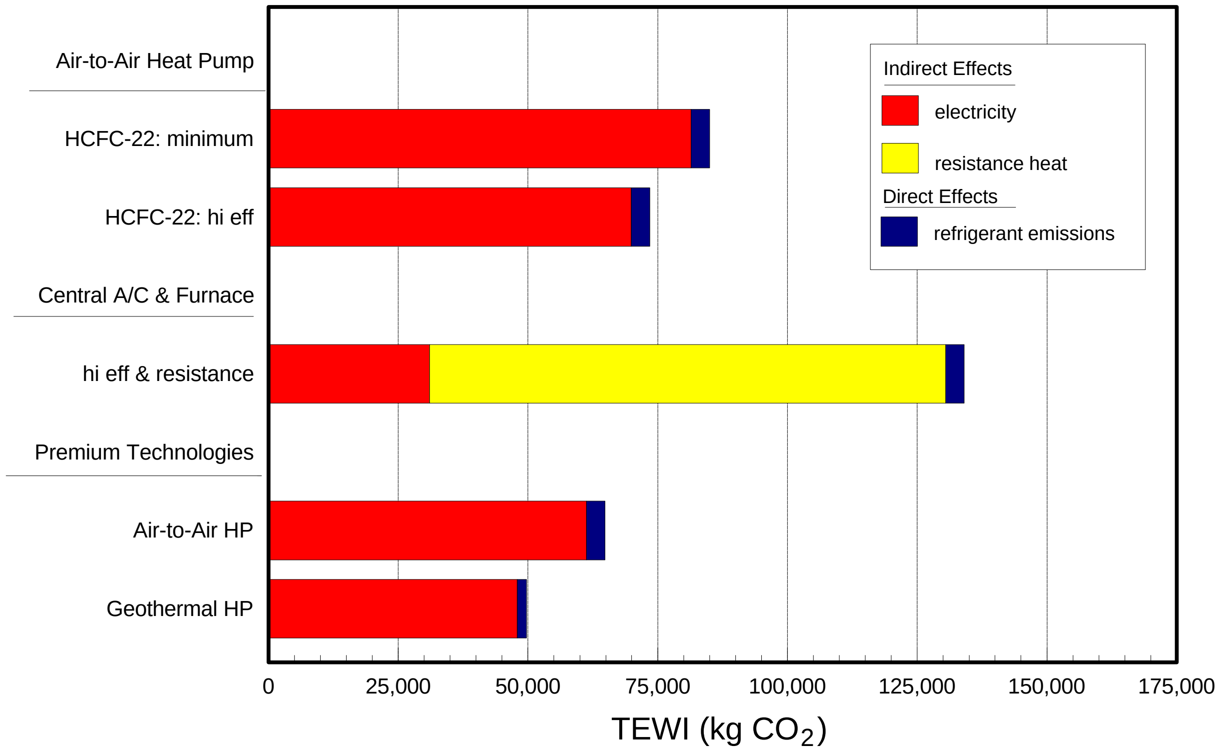

Figure 10. TEWI for residential heating and cooling options in Atlanta, Georgia USA (1996 efficiency levels).

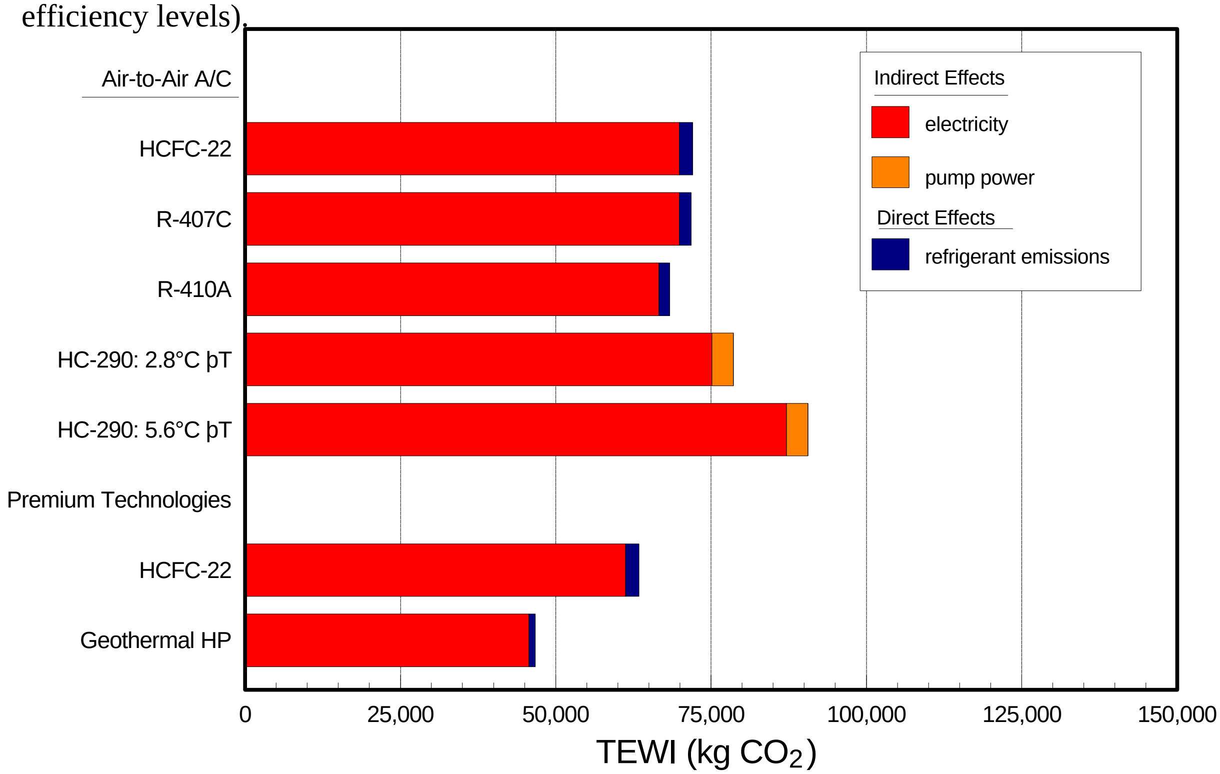

Figure 11. TEWI for residential heating and cooling options in Atlanta, Georgia USA (2005 efficiency levels).

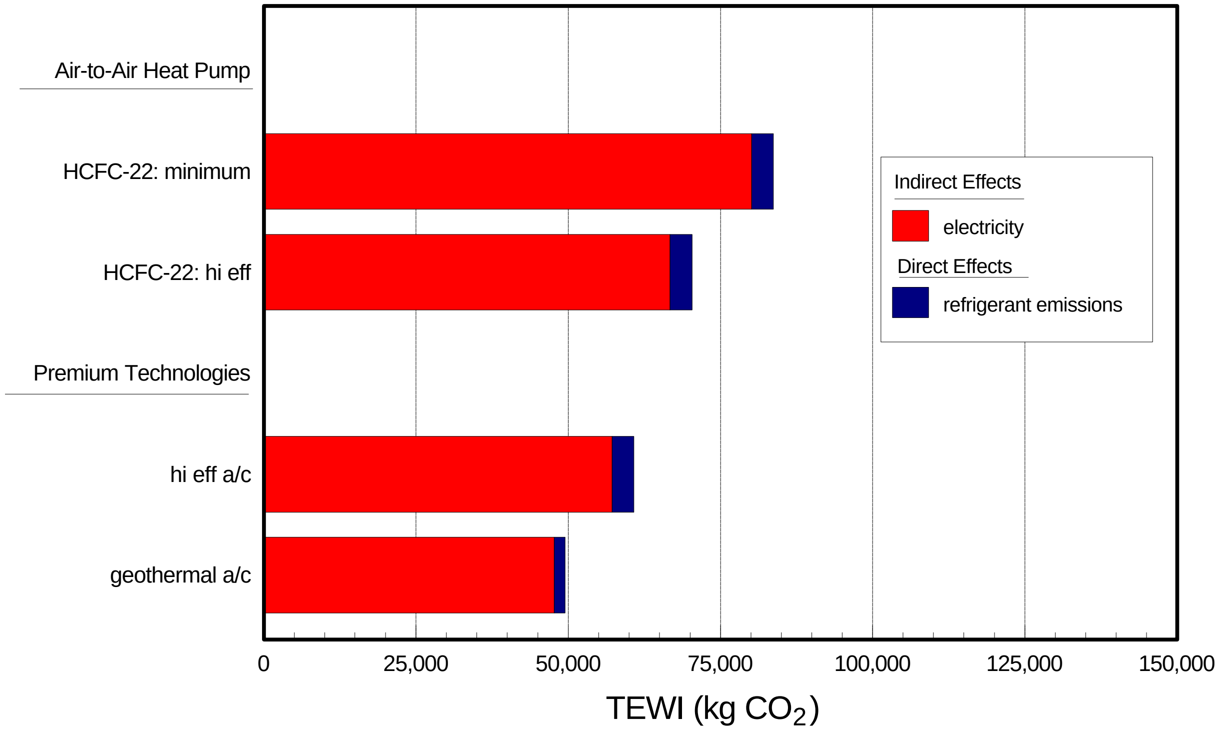

Figure 12. TEWI for residential heating and cooling options in Miami, Florida USA (1996 efficiency levels).

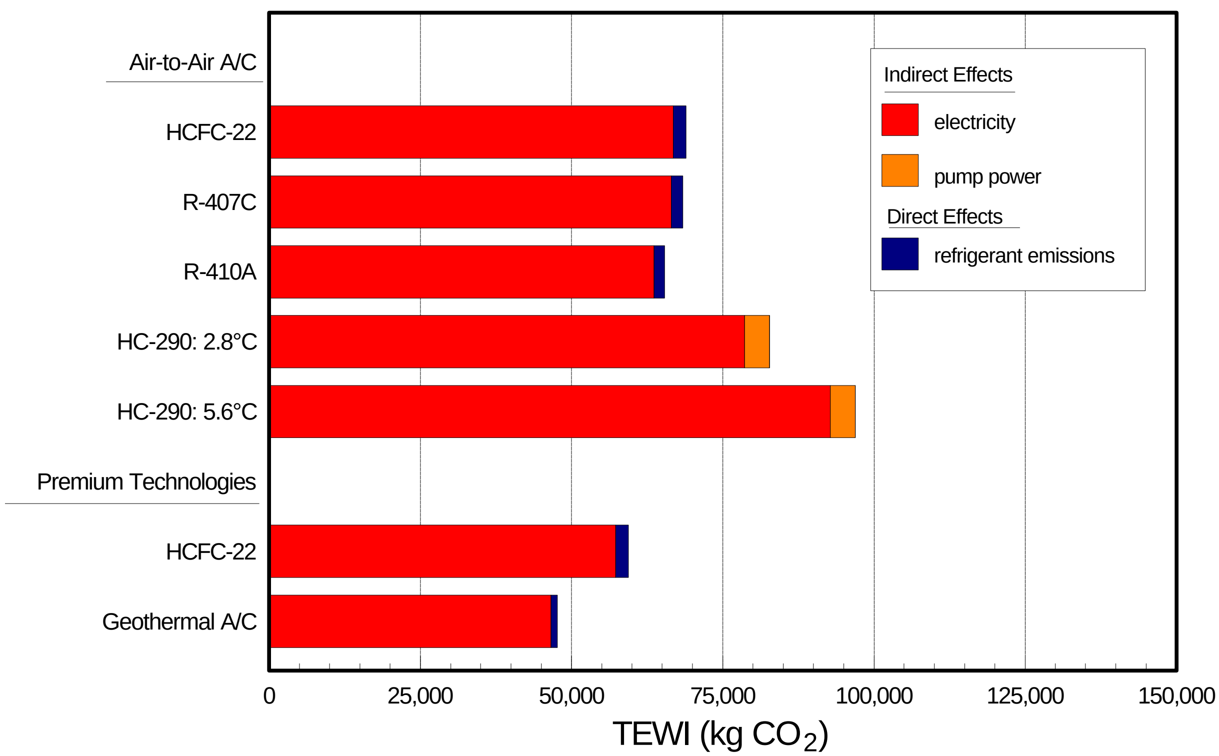

Figure 13. TEWI for residential heating and cooling options in Miami, Florida USA (2005 efficiency levels).

in each of these cities, while the lower indicates a “premium technologies” section which is a heating/cooling option that is significantly more expensive than the baseline technology. Figures 8, 10, and 12 show unitary systems currently available, whereas Figures 9, 11, and 13 suggests what TEWI is possible from technologies anticipated by the year 2005. Attaching specific prices to each option is difficult and often-times misleading because HVAC manufacturers, dealers, installers, and local utilities can all influence the final price paid by the consumer. While specific prices are problematic, it is clear that newly developed and more efficient options shown will cost more than conventional systems, and dividing these technology options into standard and premium categories gives some indication of the added investment required to obtain a TEWI benefit (Kuijpers 1995).

Each segment of the bar graphs plotted in these figures indicates TEWI contributions from different sources. The initial, darker gray section of most bar graphs is the indirect TEWI contribution from electric power used for the vapor compression heating and/or cooling process. In the Pittsburgh results, Figs. 8 and 9, electric resistance heat to supplement heat pump operation is also shown. The lighter section on some bars shows the TEWI contribution from auxiliary electric technologies such as pumps or fans that are not included in the SEER or HSPF ratings of the equipment. Electricity used for pump power on the secondary loop for the HC-290 heat pump/air conditioner would be an example of contributions added here. The moderate gray section on the ends of most bar graphs is the direct TEWI contribution caused by refrigerant losses.

Using Figs. 8 and 9 as examples, the advantages of increasing unit efficiencies become quite obvious if the HCFC-22, SEER=10/HSPF=7, SEER=12/HSPF=8, and SEER14/HSPF=9 (in the premium technologies section) are compared. Total TEWI values for these three heat pump options in Pittsburgh are about 126,000; 111,000; and 100,000, respectively. A 10 to 12% improvement in TEWI is indicated for each step of efficiency improvement. Relative TEWI decreases with increased efficiency are greater in climates with a higher cooling/heating ratio.

Figure 9 also shows the benefits in TEWI and relative energy savings associated with the added expense of a geothermal or ground-source heat pump

In all three U.S. cities, use of propane as the refrigerant with a secondary loop incurred a TEWI penalty. Direct propane systems with all the added safety precautions and increased costs needed to make them safe for the U.S./North American market are not shown, but are assumed to perform similar to the HCFC-22 heat pump/air conditioners with essentially no direct TEWI contribution from the refrigerant (Keller 1996).

Figure 10 shows a TEWI reduction resulting from residential heating in Atlanta via a heat pump rather than electric resistance heat. Similar reductions in energy use and indirect TEWI are possible if heat pump water heaters replace conventional electric water heaters (resistance heat) for domestic hot water applications in residential and commercial buildings (IEA 1993).

There are bars for the TEWI results for HFC mixtures R-407C and R-410A as well as for HCFC-22 in Figs. 9, 11, and 13. None of the systems represented were specifically designed for optimal performance with these mixtures and consequently the energy efficiencies reflect test results in “soft optimized” systems. In all the cases presented, the direct contribution of refrigerant to the TEWI was no larger than 6% of the total. The average direct contribution is more like 3 to 4%. Essentially no difference is seen in the TEWIs for HCFC-22 systems and those where R-407C or R-410A are used as substitutes, because unit efficiencies are very similar and the 100 year ITH refrigerant GWPs are 1700 for HCFC-22, 1530 for R-407C, and 1730 for R-410A. The smaller charge sizes per unit of capacity for R-410A and early indications of system efficiency improvements over HCFC-22 will help reduce TEWI even more for this option.

Nearly 80% 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 come into common usage and are followed, the direct effect will diminish in significance.

TEWI results calculated for 26.4 kW (7.5 ton) roof top air conditioning units operated with HCFC-22, R-407C, and R-410A in Pittsburgh, Atlanta, and Miami are shown in Fig. 14. In this graph, the indirect TEWI contribution from electricity consumed is indicated by the darker gray bar segment. Direct TEWI contribution from the refrigerant is indicated by the lighter gray portion of the bar. It is apparent from these results that the direct contribution to TEWI due to refrigerant emissions from this line of products is less than 8% in Pittsburgh which has the least cooling hours and correspondingly low *indirect* TEWI contribution. The relative contribution of refrigerant emissions to the total TEWI decreases rapidly in locations with more operating hours.

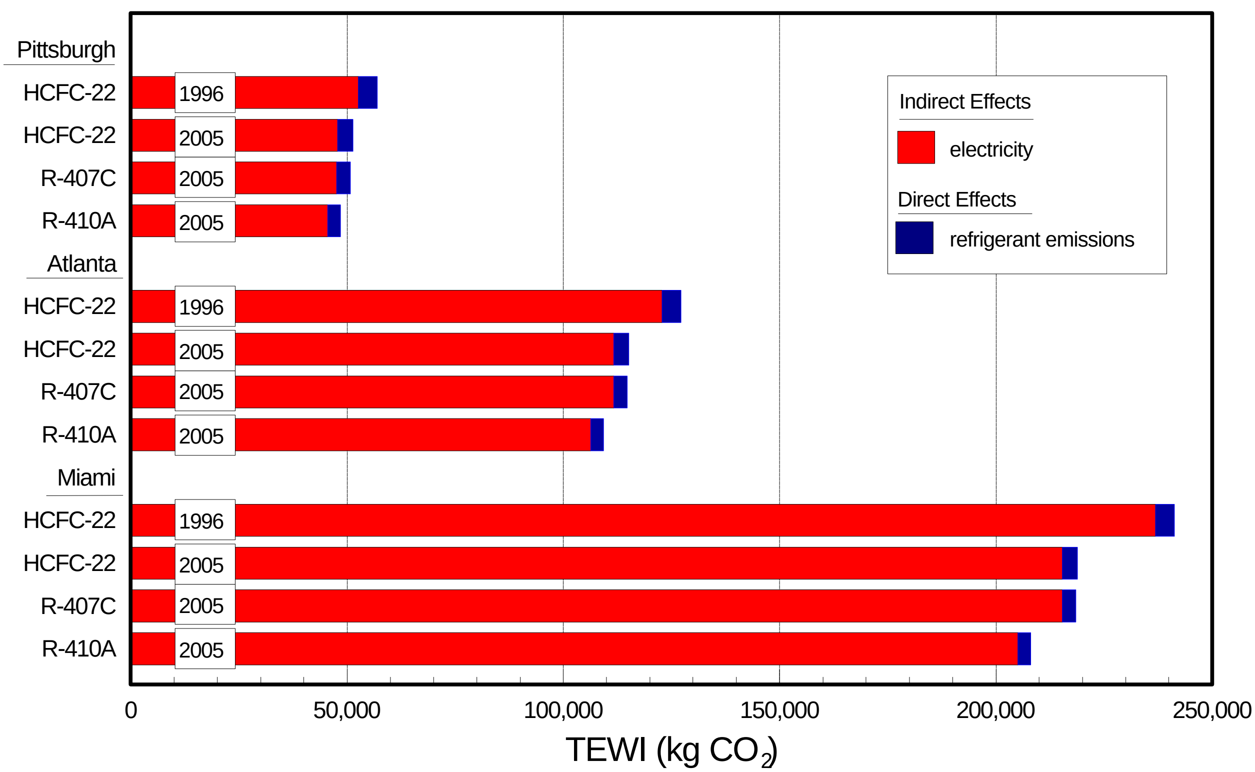

Figure 14. TEWI for 7.5 ton roof top air conditioning units in the USA.

Increasing the EER of this equipment from 10 to 11 results in a 9 to 10% decrease in indirect TEWI. Also indicated in Fig. 14, is the similarity of TEWI results for HCFC-22 and the HFC alternatives under the assumptions used for these calculations.

Figure 15 shows the relative magnitude of TEWI for several residential heating options in Europe. Direct TEWI from refrigerants for the vapor compression technologies average about 10% of the total if leak rates and end-of-life losses similar to North American equipment are assumed. The relative direct TEWI effect for this equipment in Europe is larger than for comparable North American systems because a smaller load is assumed and the average CO_2 emission rate from electric power plants, 0.47 kg CO_2/kWh, is less than the U.S. average. Even with this lower power plant CO_2 emission rate, the use of electric resistance heat results in a significantly larger TEWI when compared to the other technologies.

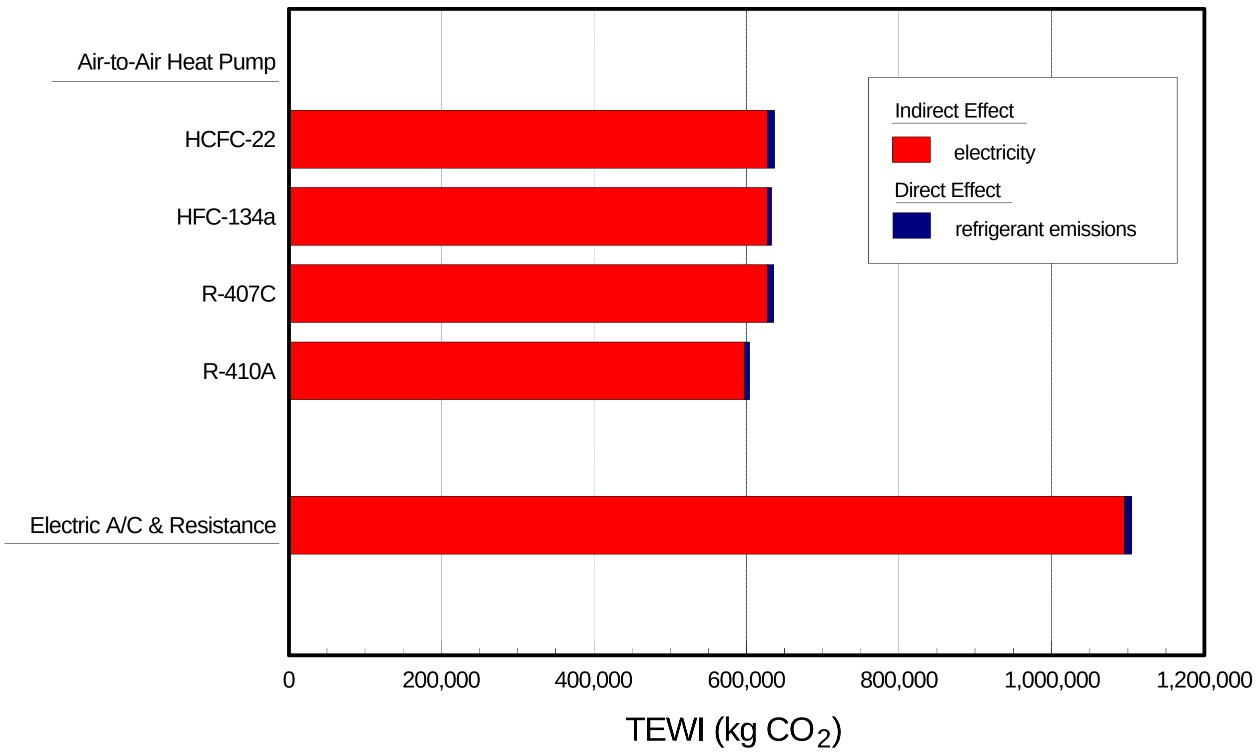

Figure 15. TEWI for residential heating options in Europe.

TEWI results for residential, cooling-only system options in Europe based on building loads and system performance values listed in Appendix D, are shown in Fig. 16. Since cooling loads given in the reference are specific to Greece, the power plant carbon dioxide emission rate for Greece, 0.976 kg CO_2/kWh, was also used for the calculations and results displayed. The direct TEWI portion of the various vapor compression technologies shown ranges from 10 to 15% of the total TEWI.

Figure 17 summarizes TEWI results calculated for a composite commercial building averaged from tabulated results presented for commercial building unitary applications in Europe (IEA 1994). With annual charge leak rates of 4% and a 15% loss of refrigerant on unit disposal, the direct TEWIs obtained in these results are less than 2% of the total. Clearly, the direct TEWI contribution for this application is not significant. No significant difference in TEWI is seen between systems using HCFC-22 and those using alternatives. Electric resistance heat shows increased TEWI.

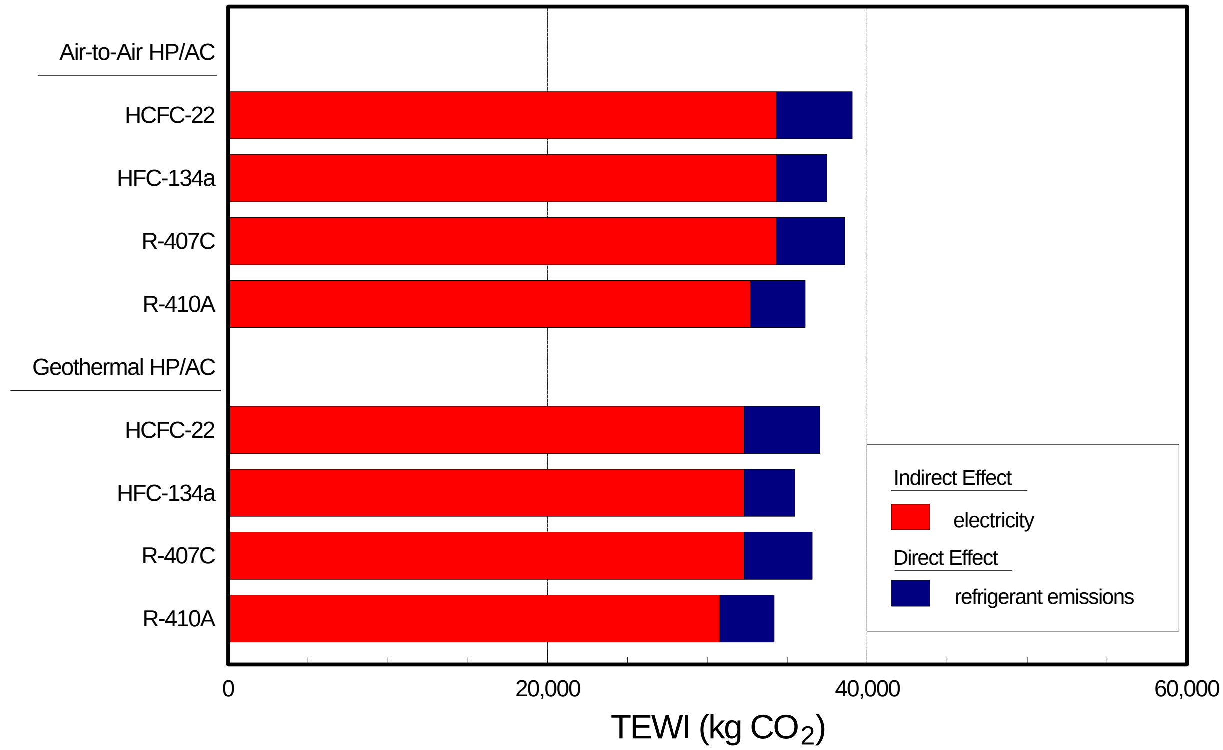

Figure 16. TEWI for residential cooling-only options in southern Europe (Greece).

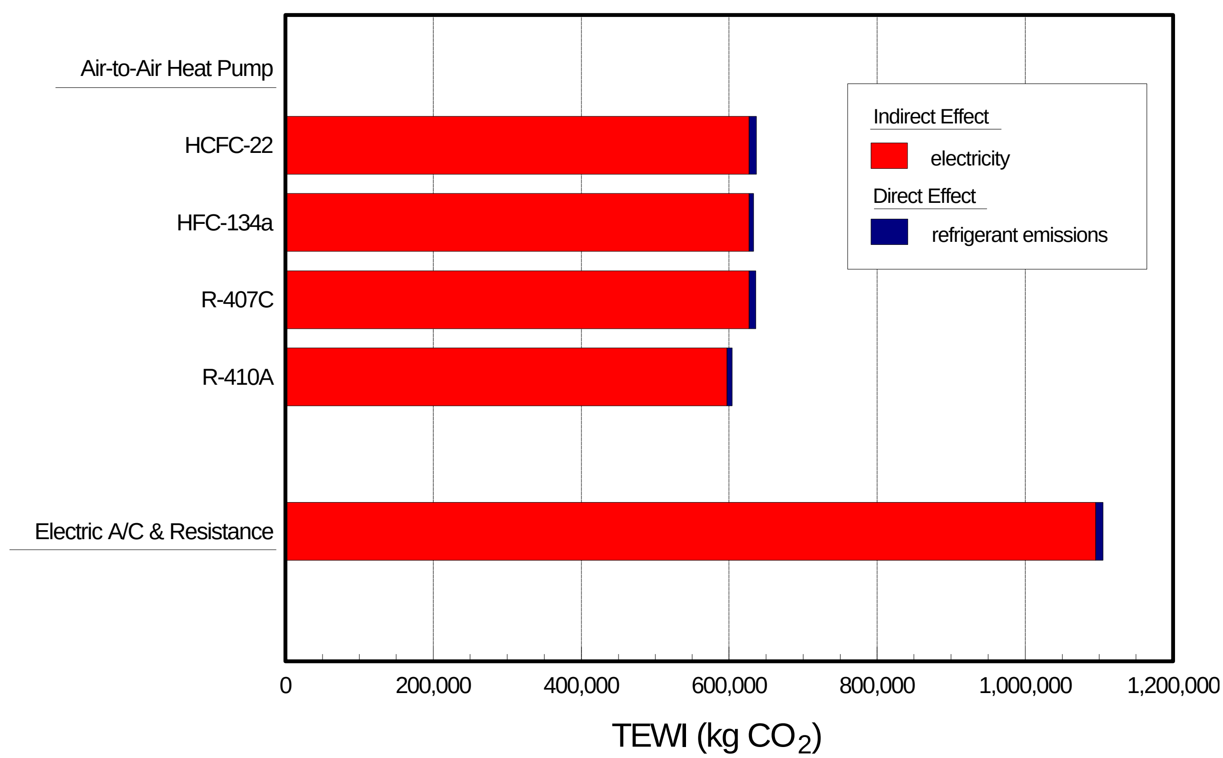

Figure 17. TEWI for commercial heating and cooling options in Europe.

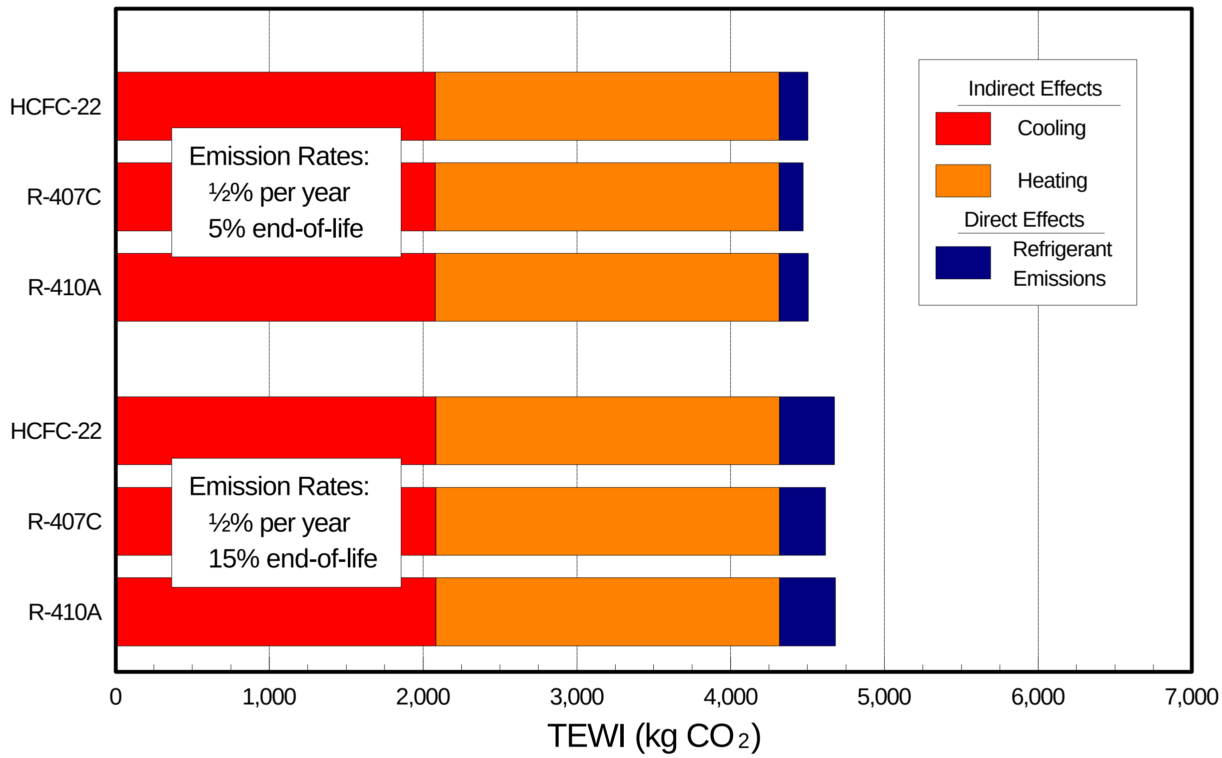

Figure 18. TEWI for 2.8 kW room air conditioner/heat pump in Japan.

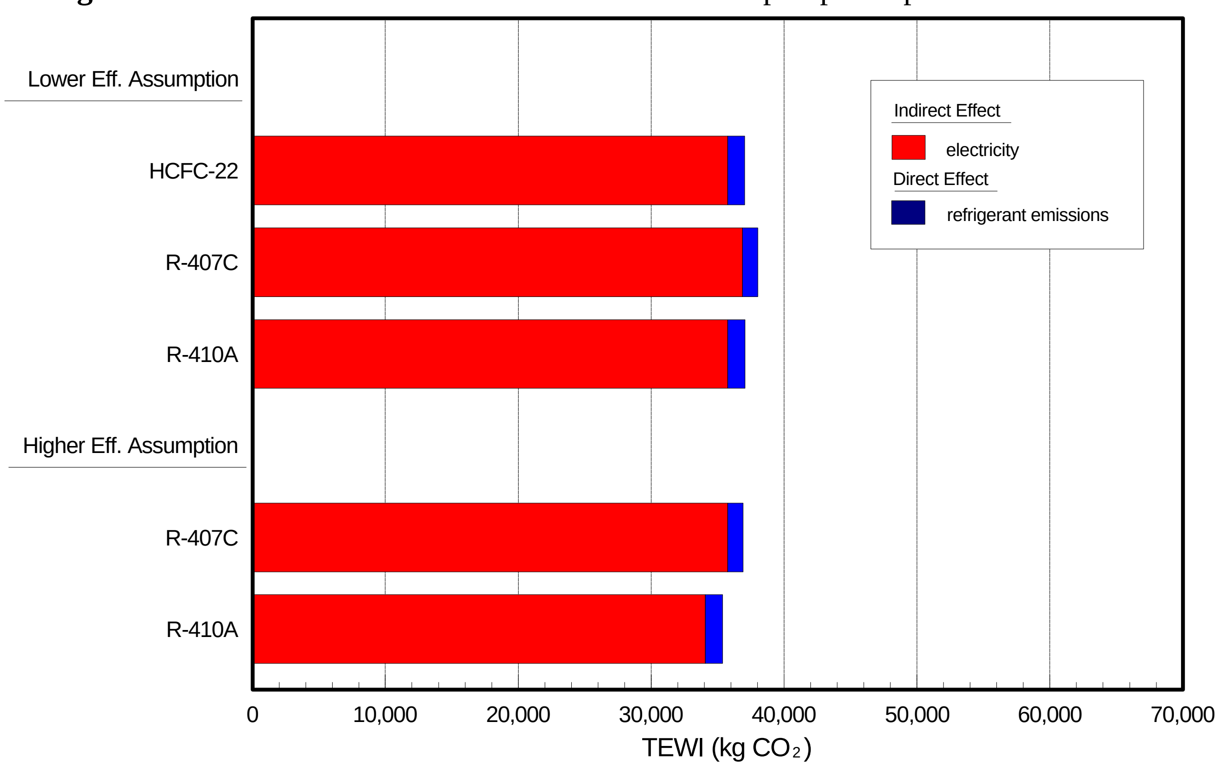

Figure 19. TEWI for a 14.0 kW packaged air conditioner/heat pump in Japan.

Results for TEWI calculations performed on a 2.8 kW room air conditioner/heat pump and a 14.0 kW packaged air conditioner/heat pump typical of those used in Japan, under operating conditions indicative of Tokyo are shown in Figs. 18 and 19, respectively. Direct TEWI accounts for less than 8% of the total under the most pessimistic leak and refrigerant recovery scenarios for the room heat pump units. Decreasing leak rates and improved refrigerant recovery estimates provided for this equipment by JRAIA (Hara 1996b) lowers the direct effect to less than 5% for the room heat pump system and to approximately 1% for the packaged system.

4.7 CONCLUSIONS

TEWIs for HFC mixtures proposed as HCFC-22 alternatives are not significantly different from those calculated for HCFC-22, and with optimization of equipment design efficiency should continue to improve. Refrigerant leakage, and the corresponding global warming impact of the refrigerant, from hermetic unitary equipment is very small and future service losses will be low because maintenance and replacement practices mandating refrigerant recovery and recycling are in place or under consideration in many countries.

The direct contributions to TEWI for all vapor compression systems presented are small fractions of the total in each case. These contributions should not be ignored, however. Procedures for handling refrigerants and accounting for refrigerant usage currently being adopted should be effective in reducing the direct effect from that shown in results presented in this chapter. TEWIs of fluorocarbons system are less than those of a propane vapor compression cycle with a secondary heat exchange loop. Arguments for not using propane in direct systems with adequate safety precautions to prevent fires and explosions, center on the relative effectiveness of additional investments and costs.

5. CHILLERS

5.1 INTRODUCTION

Comfort air conditioning in larger single-story and multistory commercial buildings is usually and most efficiently provided by chillers that cool water or a water/antifreeze mixture which is then circulated to fan coil units or air handlers in individual rooms to cool and dehumidify the air. Chillers are manufactured in cooling capacity ranges of 7.0 kW to 35,000 kW (2 to 10,000 refrigeration tons), but the larger capacity, centralized units which use screw or centrifugal compressors for the refrigeration circuit are of most interest in this chapter. Screw compressors are used in chillers with cooling capacities ranging from 250 kW (70 tons) up to a maximum of about 2300 kW (650 tons) because they are more compact, efficient, and reliable than comparable reciprocating compressors which dominate the field in lower capacity units. In chiller sizes greater than 700 kW (200 tons) centrifugal chillers are favored because of the larger volumetric flow rate of refrigerant that has to be circulated to achieve these levels of refrigeration (Stoecker 1982). Out of an estimated 211×10^6 kW of installed chiller capacity worldwide in 1991, roughly 134×10^6 kW or 64% was in North America, 39×10^6 kW (18%) in operation in Japan, and 23×10^6 kW (11%) in Europe (Fischer 1991).

Chillers using centrifugal compressors comprise approximately 70% of this installed capacity, with about 30% using a combination of reciprocating, screw, or scroll compressors. Historically, nearly two-thirds of the centrifugal chillers have been low pressure machines using CFC-11 or HCFC-123, about 12% are high pressure systems using CFC-12 or HFC-134a, with the balance using primarily HCFC-22. The discussion in this section focuses on capacity ranges served by electric-driven chillers of 1,200 and 3,500 kW (350 and 1000 tons) and alternative cooling technologies with equivalent capacities for large commercial and institutional buildings.

Commercially proven gas powered absorption chillers are currently available with cooling capacities up to 6000 kW (1700 tons). Absorption machines account for about 8% of large tonnage chiller sales in the United States and about 80 to 90% in Japan.

5.2 REFRIGERANTS

5.2.1 HCFC-123

HCFC-123 has been developed as a replacement for CFC-11 in low pressure chiller applications. Prior to the CFC phaseout, the vast majority of centrifugal chillers used CFC-11 because of its inherent efficiency and because the maximum working pressure in these machines was low enough to exempt systems from stringent pressure vessel code regulations in most countries. HCFC-123 has been the refrigerant of choice for retrofitting installed CFC-11 machines as well as for new low-pressure chillers because its thermodynamic properties are similar to those of CFC-11. In retrofit situations, HCFC-123 is a more aggressive solvent than CFC-11, so, as with other alternatives, hermetic motor insulation and other material compatibility concerns have to be resolved. HCFC-123 has a very short atmospheric lifetime (. 1.4 years) and consequently has extremely low ozone depleting and global warming potentials. The most efficient commercially available centrifugal chillers use HCFC-123 (UNEP 1995).

Ideal cycle, steady-state efficiencies were computed using thermodynamic properties at an evaporator temperature of 4.4EC (40EF) and condensing temperature of 40.6EC (105EF) to compare performance with other refrigerants (Smith 1993). These "ideal" COPs are 6.74 (0.52 kW/ton) for 0EC (0EF) subcooling and superheat, 6.89 (0.51kW/ton) for 2.8EC (5EF) subcooling and superheat, and 7.03 (0.50 kW/ton) for 5.6EC (10EF) subcooling and superheat.

Chiller efficiencies have improved significantly since the Montreal Protocol was first adopted, and chillers using alternative refrigerants are now more efficient than the products using CFCs that they replaced (Glamm 1996). Chillers using HCFC-123 are commercially available with COPs as high as 7.82 (0.45 kW/ton) at the ARI rating conditions. Significant progress has also been made in reducing refrigerant losses from purge units for low pressure chillers. Chillers with refrigerant emissions of less than 0.5% of the charge per year are being marketed (Smithart 1993); this is significantly less than projected "future loss rates" 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 resulting from losses in purge units, leakage, maintenance, and recovery upon equipment retirement for a well-maintained, normally operating machine would average 0.5% of the charge per year over the lifetime of the chiller (Hourahan 1996a).

5.2.2 HFC-134a

HFC-134a is the refrigerant of choice for the new generation of chillers that evolved from those designed to use CFC-12 and for retrofitting chillers originally designed to use CFC-12. HFC-134a has a moderate GWP, and since it contains no chlorine, has no ozone depleting potential. Very efficient HFC-134a chillers based on screw and centrifugal compressors are commercially available from a number of manufacturers (E Source 1995).

Steady-state efficiencies were computed using thermodynamic properties at an evaporator temperature of 4.4EC (40EF) and condensing temperature of 40.6EC (105EF) (Smith 1993). These "ideal" COPs are 6.29 for 0EC (0EF) subcooling and superheat, 6.48 for 2.8EC (5EF) subcooling and superheat, and 6.66 for 5.6EC (10EF) subcooling and superheat (0.56 kW/ton, 0.54 kW/ton, and 0.53 kW/ton, respectively).

Efforts to remain competitive with 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. For example, some machines use cycle modifications such as expansion turbines to recover some of the refrigerant throttling losses and apply them to supplement energy furnished by the chiller motor. Additionally, the heat exchangers are designed for very close approach temperatures to boost efficiencies. The reported COP of 6.76 is not directly comparable to the “ideal” efficiency calculations given above which do not include such cycle modifications. Ideal COPs are given to provide a rough sense of relative refrigerant efficiencies.

5.2.3 HCFC-22

HCFC-22 is a high-capacity refrigerant used in lower tonnage chillers with scroll, reciprocating and screw compressors and in the largest, 5,000 to 35,000 kW, capacity chillers with centrifugal compressors. Its low volumetric flow rate and favorable transport properties make it useful in compressors with small displacements and heat transfer surfaces. In the largest chiller size ranges, lower vapor pressure refrigerants like HCFC-123 require large compressor passages and interconnecting pipe diameters so a high-capacity, higher pressure refrigerant like HCFC-22 is favored. Under revisions of the Clean Air Act which implements the Montreal Protocol in the United States, the phaseout date for HCFC-22 in new equipment is scheduled for 2010 and a total production phaseout is scheduled for 2020. Each country is approaching this HCFC phaseout differently.

Azeotropic and zeotropic blends of HFCs, ammonia, and certain hydrocarbons have been considered as alternatives for HCFC-22. ARI, under AREP (Alternate Refrigerant Evaluation Program), has conducted a focused research effort to find and characterize suitable substitutes for HCFC-22. Most of this work was directed at finding alternatives for smaller, unitary equipment with direct expansion (DX) evaporators. A zeotropic blend of HFC-32/HFC-125 (R-410A) that is seriously being considered as a HCFC-22 alternative, operates at a significantly higher pressure which would result in new design implications for large chillers. One manufacturer has announced a line of 230-300 ton, flooded evaporator, screw chillers that will use R-410A (ACH&RN Oct. 1996). No zeotropic blends investigated in the AREP program are considered suitable for use in flooded evaporators where they would tend to fractionate. From information available at the time this report is being written, no suitable short-term replacement for HCFC-22 has been identified for use in larger capacity chillers with flooded evaporators (UNEP 1995).

As with HCFC-123 and HFC-134a, the steady state ideal efficiencies were calculated for HCFC-22 using its thermodynamic properties at an evaporator temperature of 4.4EC (40EF) and condensing temperature of 40.6EC (105EF). "Ideal" COPs under these evaporating and condensing conditions are 6.33 for 0EC (0EF) subcooling and superheat, 6.46 for 2.8EC (5EF) subcooling and superheat, and 6.58 for 5.6EC (10EF) subcooling and superheat (0.56 kW/ton, 0.54 kW/ton, and 0.53 kW/ton, respectively).

Market pressures force HCFC-22 chillers to have efficiencies similar to the HCFC-123 and HFC-134a based equipment against which they compete. Many of the same efficiency-enhancing design improvements applied to machines using other CFC alternative refrigerants are equally applicable to HCFC-22. A survey of currently available commercial equipment in the 350—3,500 kW (100—1000 ton) capacity range indicates that HCFC-22 screw compressors with full load COPs of 6.39 (0.55 kW/ton) are readily available. HCFC-22 centrifugals are certified with COPs of 6.51 (0.54 kW/ton) at the ARI rating point conditions (Hourahan 1996b).

5.2.4 Other Refrigerants

Other refrigerants suggested for large chiller applications are ammonia (R-717, NH_3), propane (HC-290), and HFC-245ca. Ammonia is an excellent refrigerant that is routinely used for large refrigerated warehouses. Since ammonia is moderately flammable and moderately toxic, its use has been confined to systems that are easily isolated from the general public. The technical expertise to develop and adapt ammonia to comfort cooling applications in chillers exits and is being applied (Fairchild 1995). Capital investment in redesign, retooling, training, maintenance, service, and marketing needed to introduce this product as a major competitor for an established non-flammable, non-toxic technology will be considerable (Stene 1996). TEWIs for ammonia chillers are calculated in this report to show what effect a zero-GWP refrigerant has on this application. An assessment of some of the performance and commercial potential considerations for ammonia chillers are given in the following section.

Theoretical efficiency estimates for hydrocarbons like propane (HC-290) in chiller applications have been made (Hayes 1989). Commercial chiller systems using hydrocarbon refrigerants in a capacity range from 10 to 90 kW (3 to 26 tons) are available in the United Kingdom (Morris & Young 1996), but hydrocarbon refrigerants are undesirable for bigger chillers because of the large refrigerant charge required in these machines. No TEWI calculations are performed for hydrocarbon refrigerants because they are an unlikely alternative technology for the capacity ranges considered for this study (UNEP 1995).

HFC-245ca has been proposed as a chlorine-free, zero-ODP alternative for HCFC-123 (Sand 1991, Smith 1993). Laboratory comparisons of HFC-245ca performance relative to CFC-11 and HCFC-123 have been performed in a 700 kW (200 ton) three-stage centrifugal chiller (Keuper 1996), and it was concluded that this HFC can obtain performance comparable to HCFC-123 in a redesigned centrifugal compressor. There are, however, some flammability concerns with HFC-245ca. This refrigerant has been found to be marginally flammable, and its flammability is dependent on the moisture content of the refrigerant/air mixture and the ignition source. In addition it has not been evaluated under the demanding toxicity tests required for a new commercial refrigerant and no chemical manufacturer has announced plans to build a production facility for HFC-245ca. Unless the situation changes, it will not be commercially available. Consequently, HFC-245ca was not evaluated as an alternative refrigerant for commercial chillers in this study.

5.2.4.1 Ammonia Chillers

Major chiller manufacturers have looked at using ammonia in chilled water systems, and while some machines are commercially available in Europe there is a great difference in attitudes between the U.S. and Europe on the use of ammonia as a chiller refrigerant (Lindborg 1997).

5.2.4.2 Theoretical Performance

Steady-state efficiencies were computed using thermodynamic properties at an evaporator temperature of 4.4EC (40EF) and condensing temperature of 40.6EC (105EF) (Smith 1993). These "ideal" COPs are 6.60 for 0EC (0EF) subcooling and superheat, 6.64 for 2.8EC (5EF) subcooling and superheat, and 6.68 for 5.6EC (10EF) subcooling and superheat (0.53 kW/ton, 0.53kW/ton, and 0.52 kW/ton, respectively).

No hard data were located comparing operating efficiencies of ammonia and other chiller systems. Ammonia chillers are commercially available in Europe and they are reported to be more efficient than HFC-134a chillers in this market (Mosemann 1993). Ammonia and HCFC-22 are expected to have comparable efficiencies in chillers using screw compressors (Calm 1994). Ammonia is incompatible with copper alloys, so chillers using ammonia would require an open drive and external electric motor (Mosemann 1993).

Screw chillers using ammonia are commercially available with good efficiencies. Some manufacturers are producing screw chillers for refrigeration equipment designed for ammonia which are initially using HCFC-22 (Fairchild 1995). Building and fire codes that have evolved restricting the use of ammonia out of concerns for public safety make it difficult to obtain the permits and licenses necessary to use ammonia chillers in urban areas of the U.S. and many other countries. These institutional factors limit the potential for ammonia chillers to replace fluorocarbon refrigeration in this application. 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 (Fairchild 1995).

5.3 ALTERNATIVE TECHNOLOGIES

5.3.1 Natural Gas Engine Driven Chillers

Engine driven chillers employ the same cooling process as conventional electric-powered systems except the electric motor is replaced by an internal combustion engine usually powered by natural gas. Since gas is substituted for electricity as the power source for this air conditioning equipment, the indirect TEWI is likely to be different for any given application. TEWI values are calculated for gas engine-driven chillers to provide a comparison with electrically driven technologies.

Some other differences in the application of electrically driven versus gas engine-driven chiller applications would be variations in maintenance and service costs, additional provisions for rejecting more heat from gas combustion technologies via the cooling tower, and the opportunity to effectively utilize the high quality waste heat from gas-engine and gas fired absorption applications for space or water heating and possibly desiccant-based air conditioning regeneration.

The theoretical cycle performance of gas-engine driven equipment is virtually the same as that of electric chillers using the same compressors and refrigerants. Full load gas-input-based cooling COPs for engine driven chillers ranges from 1.50 to 2.08 depending on the capacity and combination of compressor and refrigerant used (AGCC 1996). Full load COPs are generally smaller than the IPLV COPs used in TEWI calculations. These COPs are expressed in terms of the kWh of cooling output per kWh of high thermal input from the gas. They do not include auxiliary electric power required by electrical components on the chiller or (as with other chillers) any energy associated with cooling tower operation or chilled water circulation pumps.

Commercially proven, packaged, natural gas engine-driven water chillers are available today. Most natural gas engine-driven chillers in the 350 to 3,500 kW capacity range use HCFC-22 in screw compressors. Screw and/or centrifugal compressors are also available which use HFC-134a. Local market conditions and regional factors, such as the relative costs of gas and electricity, peak load charges and local rebates are the important criteria used to choose between electric and engine chillers.

5.3.2 Absorption Chillers

Gas-fired absorption chillers are a commercially available alternative to vapor compression centrifugal chillers. Kohler reported on the 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, 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. The incremental cost of electricity for the locality in question is a key factor in the life cycle costing [of electric and absorption chillers]"

One approach to a direct-fired, triple-effect absorption chiller seeks to improve efficiency by operating a direct-fired generator at a temperature high enough so that the condenser waste heat and the absorber waste heat from this high temperature stage are enough to operate two lower stages of conventional LiBr single-stage absorbers (Kujak 1996). A new water/absorption brine pair with new corrosion inhibitors and chemical performance additives are required for the upper stage of this unit.

As with natural gas engine-driven chillers, primary energy efficiencies (gas-input-based COPs) are used in characterizing the performance of absorption chillers and these 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 may or may not include combustion losses.

Theoretical efficiencies of single-effect absorption chillers are not given. Although cooling COPs 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; COPs of 2.0 have been reported, but these are based on 0.6EC ÄT's (1EF) in the heat exchangers instead of the 6EC (10EF) ÄT'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 COPs of single-effect, indirect fired chillers typically peak around 0.7 with COPs of 1.2 widely available for steam-fired double-effect chillers (Kohler 1993, AGCC 1996). COPs for double-effect direct fired absorption chillers are on the order of 1.0. The lower COP for the direct-fired chillers result from losses in the combustion process. The projected efficiency for a direct-fired, triple-effect absorption concept whose market introduction is scheduled for 1999, is 1.45 COP (Kujak 1996).

Absorption chillers are a viable technology, favored under certain economic circumstances. Three U.S. HVAC companies are developing direct-fired, triple-effect absorption concepts (DeVault 1997). The efficiencies which are ultimately achieved in commercially available equipment depend on technology and market conditions. 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. They are used to a lesser extent in the U.S., often in combination with centrifugal chillers and controlled in such a way as to minimize operating costs.

5.4 ASSUMPTIONS

5.4.1 Carbon Dioxide Emission Rates From Power Plants

National averages are used for the power plant emission rates in these calculations. The average electrical power plant emission rates are 0.650 kg CO_2/kWh for the United States, 0.470 kg CO_2/kWh for Europe, and 0.473 kg CO_2/kWh for Japan , Appendix A. These emission rates are compiled from open literature data rather than calculated from the fundamental heat content of fuels, fuel mix used in power production, plant efficiencies, and transportation and distribution losses.

The heat content and carbon dioxide emission rate used for natural gas were 38,200 kJ/m3 and 51.1 g/MJ, respectively. A 96.5% distribution efficiency was assumed for natural gas which raised the CO_2 emission rate to 53.0 g/MJ (55.9 g CO_2/1000Btu) at its point of use (EIA, 1997).

5.4.2 Chiller-Specific Assumptions

Analysis assumptions used for calculating the direct contribution to TEWI are summarized in Table 9.

5.4.2.1 Chiller Capacities

According to information reported to ARI, centrifugal chillers have capacities ranging from 260 kW to 28,000 kW (75 tons to 8,000 tons). The weighted average capacity of the roughly 10,000 chillers produced in the United States for domestic and foreign consumption in 1995 was 1,670 kW (475 tons). Chillers in the range from 350 kW to 2,100 kW accounted for 79% of the chillers produced in 1995(Hourahan 1996a).

Table 9. Refrigerant emissions for 1200 kW (350 refrigeration ton) and 3500 kW (1000 refrigeration ton) chillers.

	Refrigerant Charge		Annual Emission Rate (percent of charge / kg/y)		
Chiller	(kg/kW)	(kg)	0.5% (kg/y)	2% (kg/y)	4% (kg/y)
<u>1200 kW Screw or Centrifugal Chiller</u>					
CFC-11	0.28	336	1.68	6.72	13.4
CFC-12	0.35	420	2.10	8.40	16.8
HCFC-123	0.30	360	1.80	7.20	14.4
HFC-134a	0.23	276	1.38	5.52	11.4
HCFC-22	0.23	276	1.38	5.52	11.0
R-717	0.13	156	0.78	3.12	6.2
<u>3500 kW Screw or Centrifugal Chiller</u>					
CFC-11	0.28	980	4.90	19.6	39.2
CFC-12	0.35	1225	6.12	24.5	49.0
HCFC-123	0.30	1050	5.25	21.0	42.0
HFC-134a	0.23	805	4.03	16.1	32.2
HCFC-22	0.23	805	4.03	16.1	32.2
R-717	0.13	455	2.27	9.1	18.2

Note: additional data for computing direct effect includes equipment lifetime of 30 years, refrigerant GWPs, and assumed end-of-life loss (refrigerant lost when the equipment is retired).

5.4.2.2 Refrigerant Charge

Refrigerant charge varies by chiller size, refrigerant, machine vintage, compressor type, manufacturer, heat exchanger options, and other variables. The refrigerant charge used for these calculations was a composite kg/kW value for screw and centrifugal compressors integrated over the size range of interest. Individual sources for these values were 1) those published in the original TEWI report, 2) those estimated by an Ad-Hoc subcommittee formed by ARI member companies, and 3) those published in the 1995 UNEP Report (Fischer 1991, Hourahan 1996a, UNEP 1995).

5.4.2.3 Refrigerant Loss Rates – End of Life Charge Loss

Annual loss rates for new chillers were estimated by the ARI Ad-Hoc committee at 0.5% of the total charge. To allow for large, catastrophic charge losses and some loss as a result of routine service, TEWIs were also calculated for 1.0%, 2.0%, and 4.0% annual rates. In the United States, losses from refrigeration equipment have been dramatically reduced in response to the Clean Air Act (US 1993). One of the provisions of this act requires the “prompt” or “timely” repair of leaks showing an annual loss of more than 15% of the charge in systems containing over 50 pounds (23 kg) of refrigerant. The ARI committee estimated that approximately 5% of the original charge would be lost at the end of chiller life, reflecting current refrigerant prices and recovery, reclaim, and recycling laws.

5.4.2.4 Chiller Life

Chiller lifetime used in this report, 30 years, is the same as that used in the previous two studies and is consistent with estimates provided by ARI. The chiller life assumption affects estimates for *direct* TEWIs because refrigerant loss rates are expressed as a percent of charge lost per year. The *indirect* TEWI is also impacted because CO_2 emissions resulting for fossil energy consumption used for electricity or gas to operate the chiller are based on annual loads multiplied by the number of years of life.

5.4.3 Chiller Energy Consumption Calculations

5.4.3.1 U.S. Chiller Calculations

Chiller performance was expressed in terms of *Integrated Part Load Value* (IPLV) as defined by ARI Standards 550-92 for centrifugal and rotary equipment and 560-92 for absorption equipment. IPLVs were chosen for this work rather than Rated Full Load Performance and Integrated Full Load Hours because this equipment operates at part load most of the time, and the IPLVs purport to give a more accurate indication of chiller performance under part-load conditions (ARI #550-92 1992) (ARI #560-92 1992). Table 10 summarizes the IPLV used to calculate TEWIs for this report. These IPLVs, which are listed for the best new equipment currently available and for equipment projected to be on the market by the year 2005, were taken from estimates provided by ARI members, an equipment guide published by the American Gas Cooling Center, and a listing of currently available electric chiller models (Hourahan 1996b, AGCC 1996, E Source 1995).

Table 10. Chiller integrated part load values (IPLVs) for analysis of commercial chillers (kW/RT).

	1200 kW (350 RT)		3500 kw (1000 RT)	
Equipment	1996	2005	1996	2005
Screw Chillers				
HCFC-22	0.56	0.54	0.60	0.58
HFC-134a	0.60	0.54	NA	NA
R-717 (ammonia)	0.56	0.54	0.59	0.57
Centrifugal Chillers				
HCFC-22	0.59	0.53	0.54	0.48
HFC-134a	0.56	0.52	0.54	0.48
HCFC-123	0.52	0.47	0.47	0.45

5.4.3.2 Japanese Chiller Calculation

Some TEWI calculations were made on current electric chiller options available in Japan based on a unique set of parameters provided by K.Hara (1996a) which is summarized in Table 11. These calculations used full load rating performance efficiencies and annual full load hours for a Tokyo location.

5.5 METHODOLOGY

Table 11. Assumptions for Japanese chiller calculations.

Chiller Parameter	HCFC-123	HFC-134a
Refrigerant		
charge (kg/kW)	0.24	0.24
annual make-up rate (% of charge)	1%	0.5%
end-of-life recovery rate (% of charge)	95%	95%
Cooling Capacity		
kW	1055	1055
refrigeration tons (RT)	300	300
Efficiency		
COP	5.0	5.0
kW/RT	0.70	0.70
Annual Operating Rate (full-load hours)	700	700
Auxiliary Electricity Demand (cooling tower and cooling water pumps)	20%	20%
Equipment Lifetime (years)	25	25

Direct and indirect global warming contributions in kg of CO_2 are calculated and combined to estimate total TEWI. Direct TEWIs are calculated from estimates of refrigerant losses. The total refrigerant charge, measured in kg, is multiplied by the annual make-up rate and the lifetime of the unit in years. This is added to the number or kg of refrigerant lost when the chiller is scrapped or decommissioned and multiplied by the 100 year ITH GWP, whose units are in equivalent kg of CO_2 per kg of refrigerant, to get the direct TEWI.

Indirect TEWI results from the CO_2 released to the atmosphere as a result of fuel or electrical energy use by the chiller over its useful lifetime. An estimate of annual energy use for the chiller is based on its efficiency and the load. This is multiplied by the lifetime of the chiller and an appropriate factor for converting this energy into the kg of CO_2 released while providing that energy. The units of indirect TEWI are also kg of CO_2. Published equipment rating efficiencies in (kW/ton or gas COPs), IPLVs, an estimate of annual operating hours, an equipment lifetime, and power plant CO_2 emission rates per kilowatt of delivered electricity or the kg CO_2/1000 Btu value for gas are used to calculate indirect TEWIs.

IPLVs with the total number of annual cooling hours estimated for an office building in Atlanta (2125 hr/yr) are used for the bulk of the TEWI calculations presented in this section.

An additional electrical consumption burden was added to all the chillers for energy used by cooling tower pumps, chilled water pumps, and fans (air cooled chillers were not considered). A unified method to quantify the electrical energy associated with heat rejection equipment was used which is based on the quantity of heat that must be rejected by each of the chiller technologies. The rationale and supporting assumptions for this method are given in Appendix F.

Auxiliary electric loads of 0.035 and 0.046 kW/ton, respectively, for controls and solution pumps were added to the direct-fired double- and triple-effect absorption chiller indirect TEWI calculations based on data from the AGCC Natural Gas Cooling Equipment Guide (1996) and preliminary specifications for the triple effect product (Fischer 1994). An *IPLV* of 1.50 COP was used for the triple-effect chiller based upon a 1.45 rating COP and estimates obtained from ARI member companies. No auxiliary electrical burden (other than that assigned for cooling tower operation) was given to natural gas engine-driven chillers.

No additional electrical burden was assigned for ventilation or air movement energies associated with fan coil units or central air handlers in buildings with centralized chillers. The capacity, electric demand, and energy use of these subsystems are comparable for electric and gas chillers of the same cooling capacity. In making energy or TEWI comparisons between chillers and large unitary air-conditioning equipment, additional energy use should be assigned to chiller options reflecting energy required for air circulation within the building. HVAC fan energy consumption ranges from 32 to 43 kWh/m^2 (3 to 4 kWh/ft^2) per year, which amounts to 10% to 15% of the total building electrical energy use and 20% to 40% of the energy used for heating and cooling (McLain 1988).

5.6 RESULTS

TEWI results for large electric driven chillers in the U. S. are summarized in Figs. 20 and 21. Two chiller capacity ranges were used for these estimates, 350 refrigeration tons (1,200 kW) and 1000 refrigeration tons (3,500 kW). Estimates of TEWI were made for “best” electric driven chiller equipment available in 1996 and improved chillers likely to be available by 2005, Figs. 20 and 21. Improvements in chiller performance over this time range were estimated by ARI member companies that manufacture this equipment (Hourahan 1996b).

Segmented bar lengths are shown for the indirect TEWI which was calculated from the chiller IPLV, its capacity, the annual operating hours for an Atlanta office building (2125 hr/yr), the average annual CO_2 emission rate for U.S. electricity production including transmission and distribution losses (0.65 kg CO_2/kWh), the kg of CO_2/1000 Btu of delivered natural gas (where appropriate), and a 30 year lifetime, as explained in the Methodology section. Shorter segments on the ends of these bars show varying leak rate scenarios for these chillers, 0%, 0.5%, 1.0%, 2.0% and 4.0% (annual refrigerant leak or make-up rates taken as a percentage of the total chiller charge). In addition we assume a 5% loss of charge when the chiller is eventually scrapped. Some CFC-11 and CFC-12 chiller data of 1993 vintage equipment efficiencies are shown on Figs. 20 and 21 for comparisons with the CFC-free technologies and because many chillers are still operating with these refrigerants. IPLVs of 0.58 for CFC-11 and 0.59 for CFC-12 and an annual leak rate of 4.0% were assumed for this equipment. Tabulated results used for these figures are given in Appendix G.

Figure 22 shows the sensitivity of TEWI values to the CO_2 emission rates from electric power

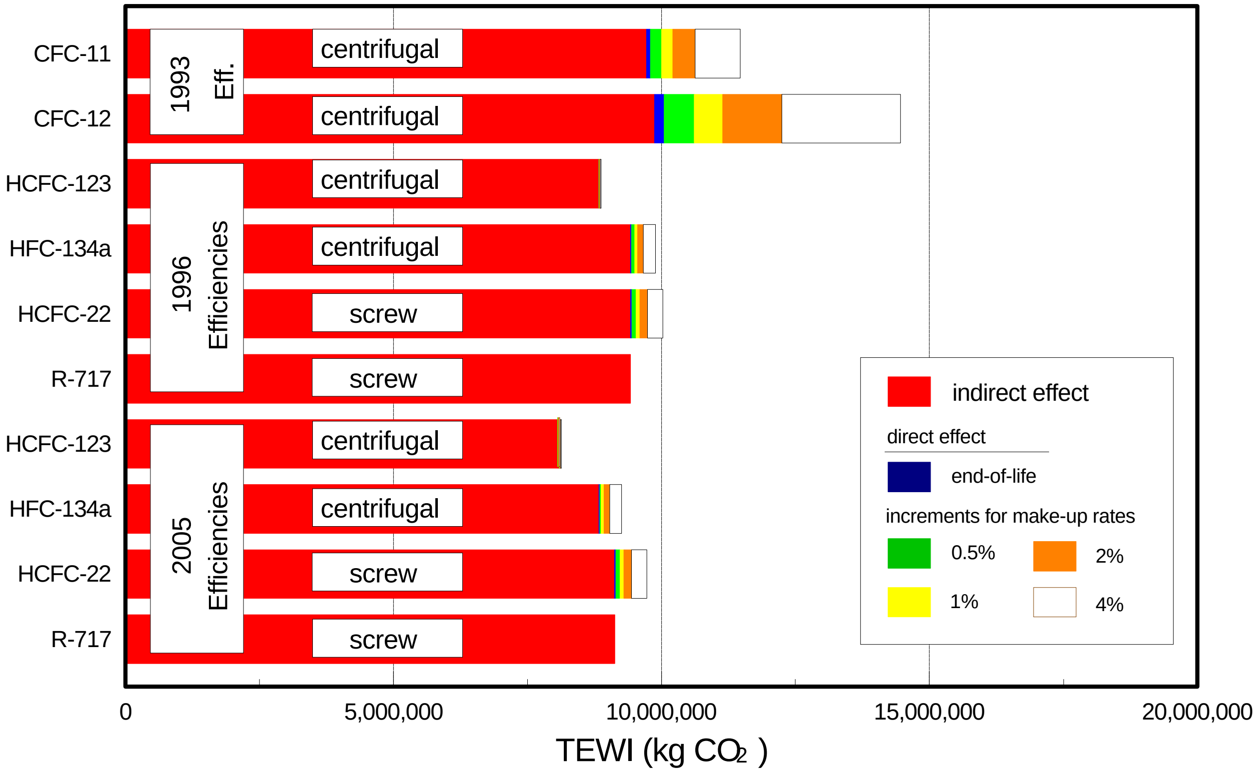

Figure 20. TEWI for 1200 kW (350 ton) electric driven chillers in an Atlanta.

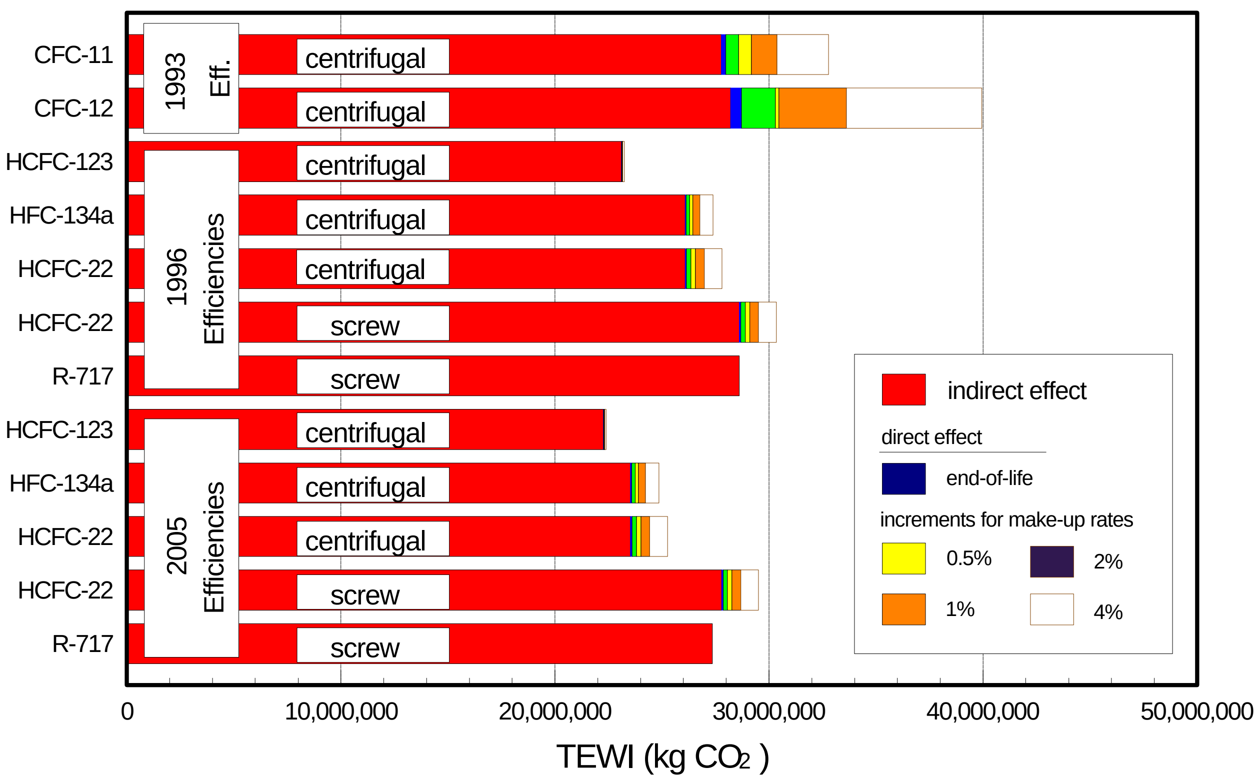

Figure 21. TEWI for 3500 kW (1000 ton) electric driven chillers in an Atlanta office application.

plants. Data for 3500 kW (1000 ton) equipment likely to be on the market in 2005 are presented.

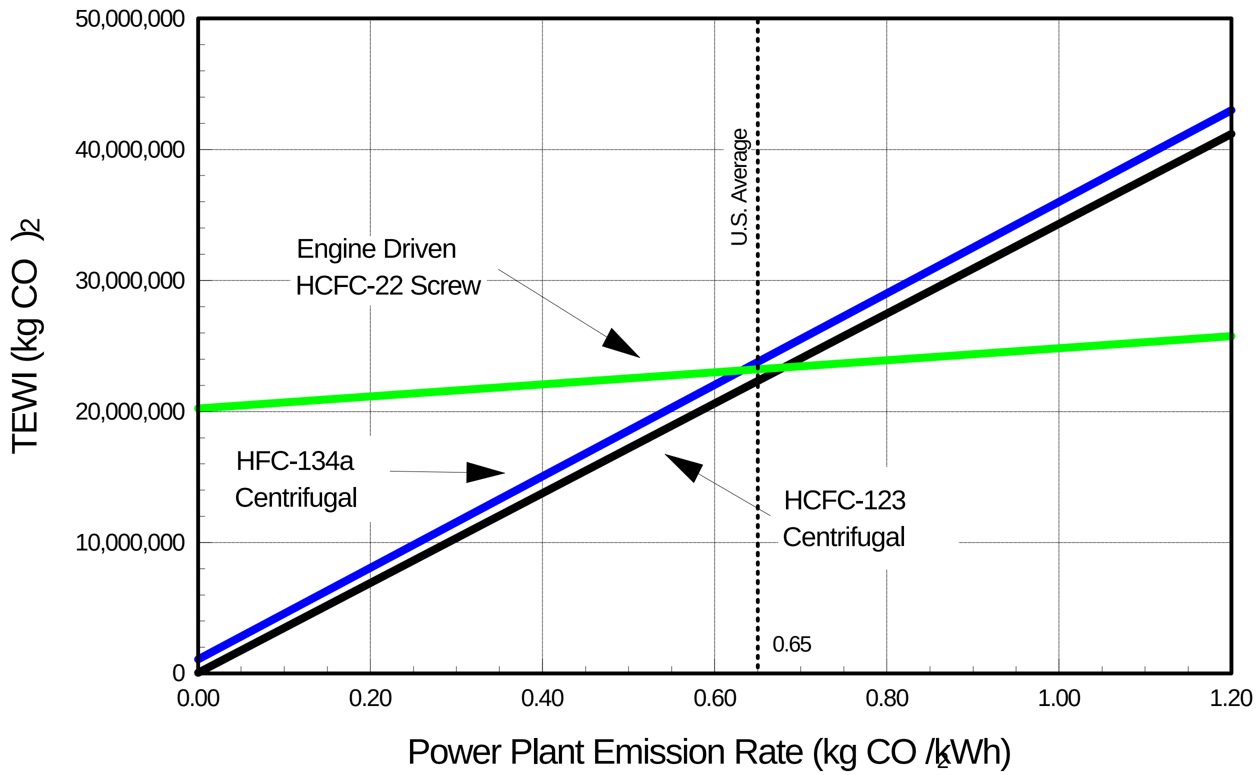

Figure 22. TEWI for 3500 kW (1000 ton) chillers in Atlanta across a range of CO_2 emission rates (2005 efficiencies).

Values chosen for the annual operating hours have a proportional effect on the indirect TEWI. Figure 23 shows a plot of TEWI for popular 3500 kW (1000 ton) chiller options as a function the annual operating hours. While indirect TEWI is directly related to the number of operating hours, the absolute value of direct TEWI remains fairly constant. Longer chiller operating hours minimize the direct TEWI contribution.

Figure 24 shows TEWIs for the electric chiller options suggested by Hara and JRAIA for the Japanese market. Power plant CO_2/kWh emission rates, equipment performance characteristics, and operating conditions suggested by Hara were used for the calculations (Hara 1996a). The large difference in total TEWI between Japanese and U.S. chillers is due to differences in hours of operation rather than any fundamental difference in technology.

No unique chiller calculations were carried out for Europe because large centralized chiller installations are not as popular in European countries, as indicated by the breakdown of installed chiller capacity presented in the introduction. To perform a calculation for a European application, chiller efficiencies, the chiller capacity, annual cooling hours, and the power plant CO_2 emission rate have to be substituted for those used for North American calculations. Using the same performance factors for North American and European chillers is reasonable because chiller operating conditions do not vary widely (Fischer 1991).

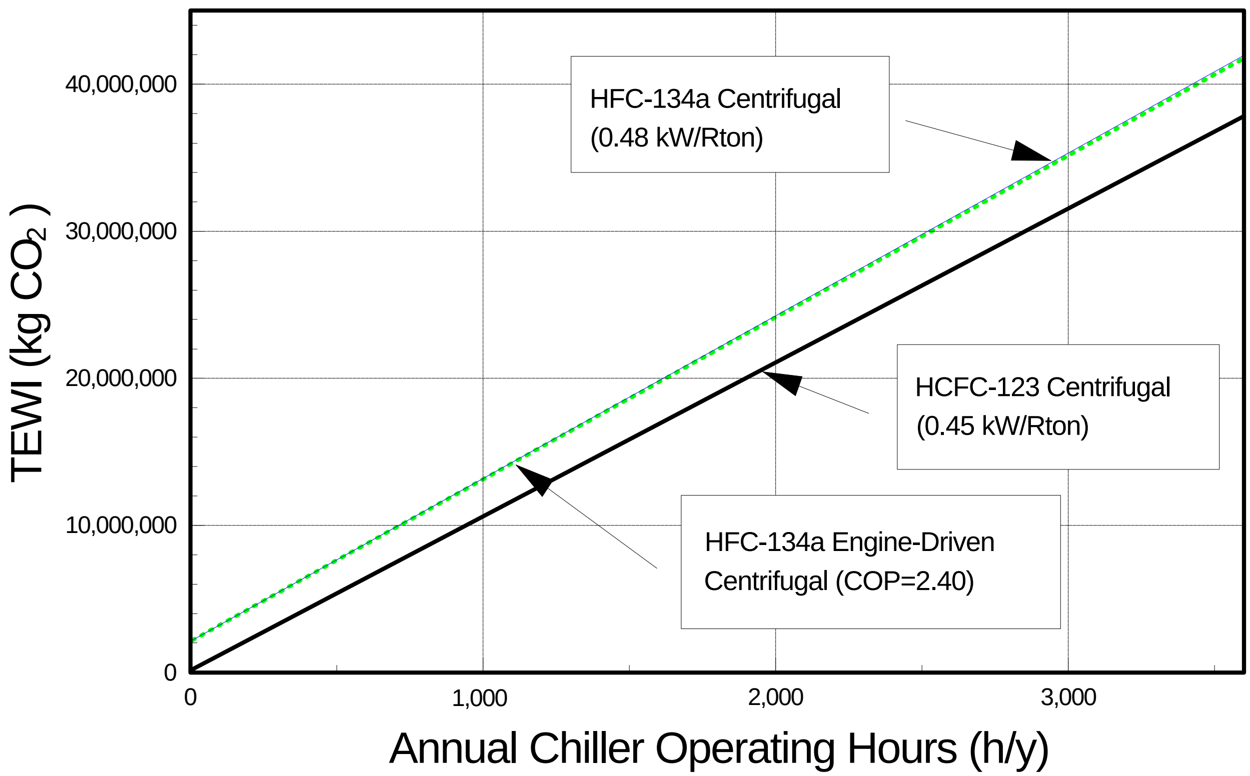

Figure 23. TEWI for 3500 kW (1000 ton) chillers in Atlanta across a range of operating hours (2005 efficiency levels)

The TEWI of 1200 kW (350 RT) and 3500 kw (1000 RT) engine driven and absorption chillers based on the efficiency data in Table 12 are presented in Figs. 25 and 26. These results show the low indirect effect from energy use for engine driven chillers as well as the significant reductions in indirect effects as the technology develops from 1996 double effect absorption chillers to triple effect chillers. The results for engine driven chillers using HFC-134a or HCFC-22 show a possible direct effect from refrigerant emissions at end-of-product life and annual losses, but it remains small even under the highest refrigerant loss rates

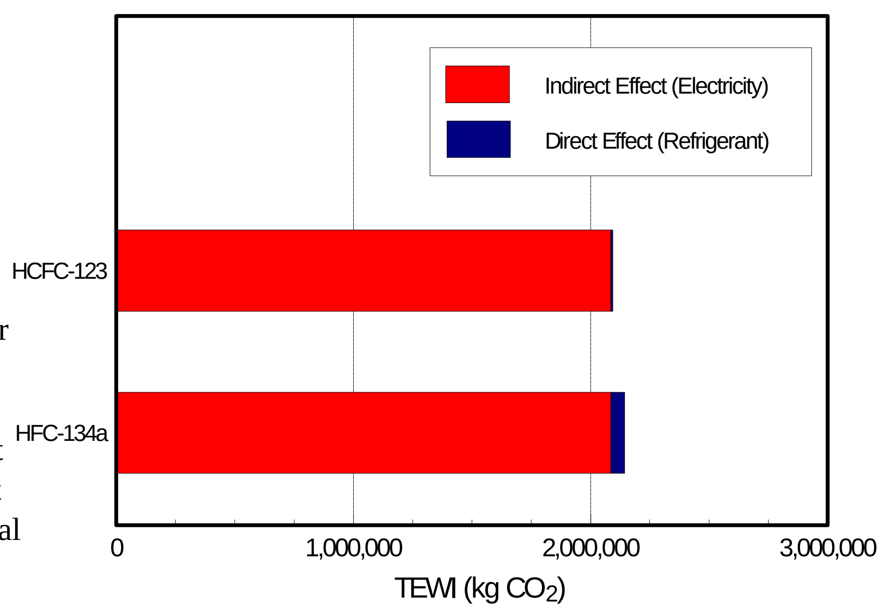

Figure 24. TEWI for 1055 kW (300 ton) chillers in Tokyo, Japan.

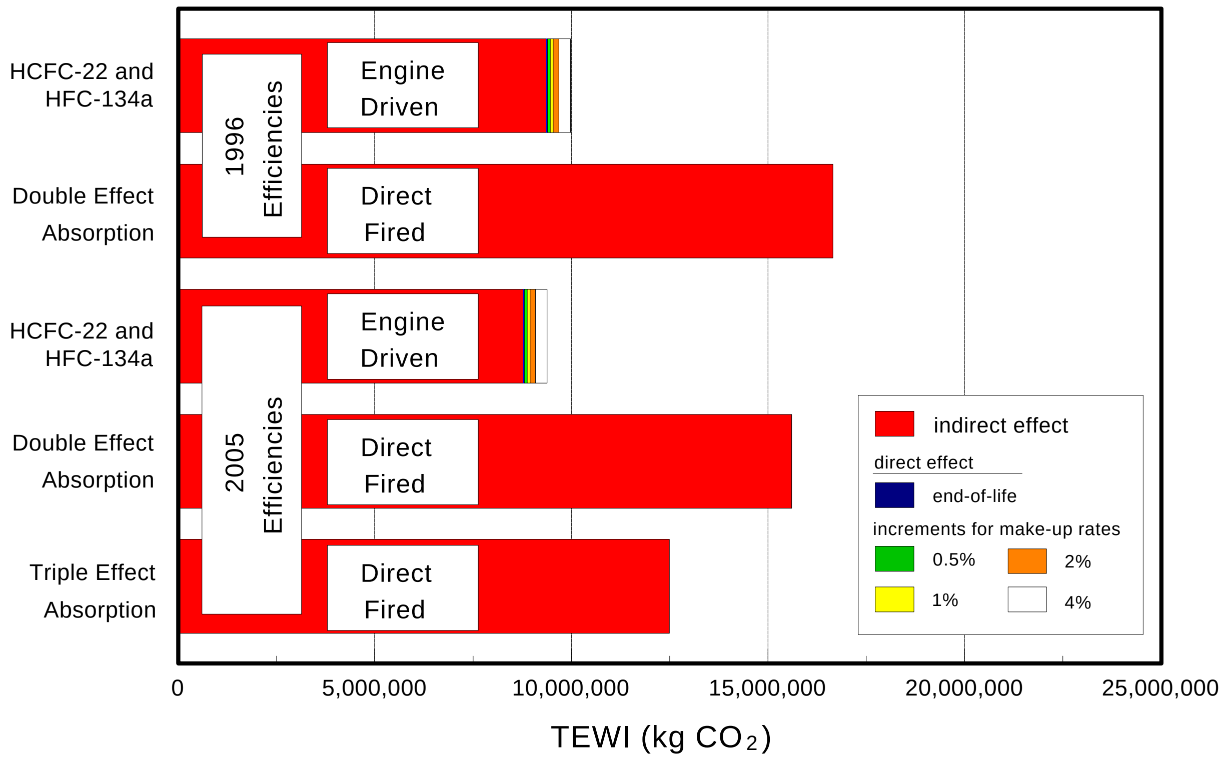

Figure 25. TEWI for 1200 kW (350 ton) gas-fired chillers in Atlanta, Georgia USA

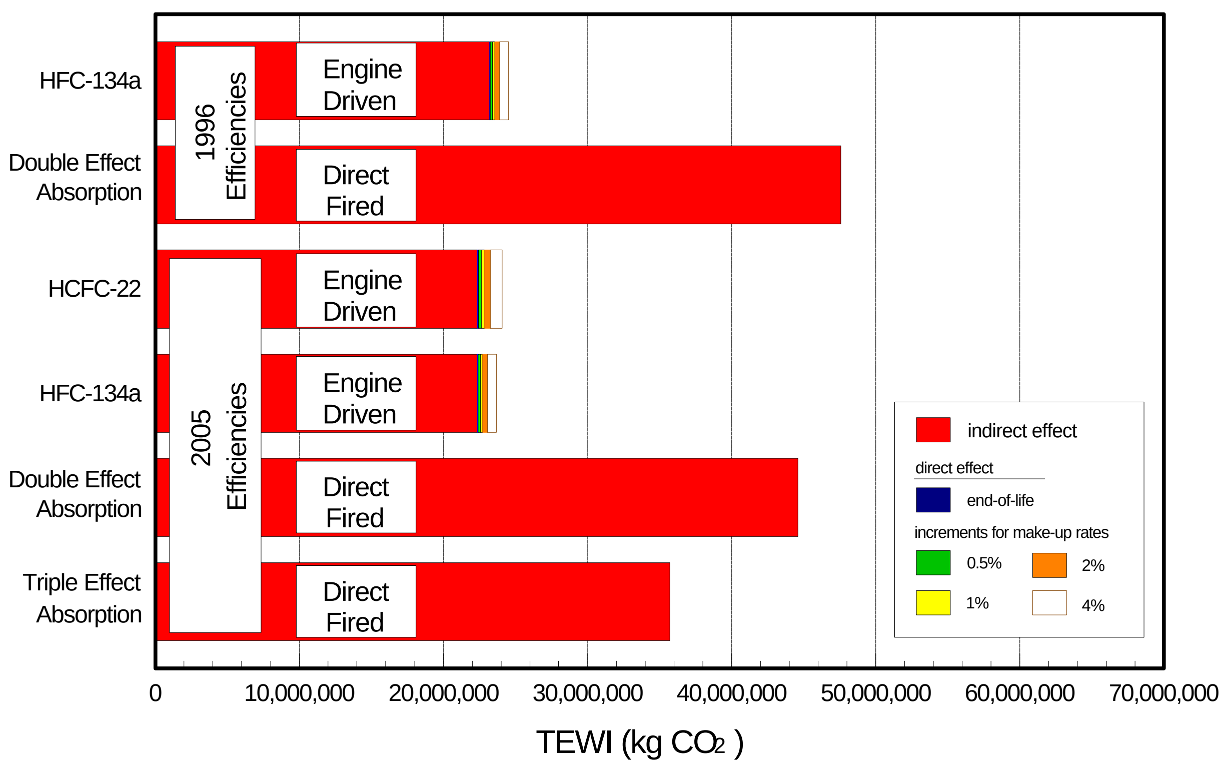

Figure 26. TEWI for 3500 kW (1000 ton) gas-fired chillers in Atlanta, Georgia USA.

considered.

Table 12. Chiller integrated part load values (IPLVs) for analysis of commercial chillers (kW/RT).

Equipment	1200 kW (350 RT) 1996	1200 kW (350 RT) 2005	3500 kw (1000 RT) 1996	3500 kw (1000 RT) 2005
Engine Driven Chillers				
HCFC-22 Screw	1.95	2.10	2.30	2.40
HFC-134a Centrifugal	1.95	2.10	2.30	2.40
Absorption Chillers				
Direct fired, double effect	1.07	1.15	1.07	1.15
Direct fired, triple effect	NA	1.50	NA	1.50

gas COPs are given for part-load gas chiller IPLVs, defined as the delivered cooling effect divided by the heat content (HHV) of the gas consumed (no waste heat recovery); NA signifies equipment/refrigerant combination is not available.

5.7 DISCUSSION

Figures 20 and 21 illustrate the environmental benefits of HCFC-123 as a refrigerant for this application. It does contain chlorine which gives it an ODP rating of 0.02. However, it shows the best cycle efficiency of all the alternative refrigerants currently being used in large chillers with flooded evaporators, and it has the lowest GWP (93) of all the non-flammable refrigerants. The one HFC alternative which has been suggested and preliminarily evaluated for use in low pressure chillers is HFC-245ca which has a GWP estimated at 560 and is slightly flammable in humid air.

TEWI for chillers is unlikely to improve markedly from 1996 to 2005 because annual refrigerant leakage and recovery rates are reaching their lowest practical limits, and dramatic improvements in chiller cycle efficiency within the last 10 years has left very little room for substantial improvements in this area (Glamm 1996).

Natural gas engine-driven chillers with COPs or IPLVs of 2.0 or greater have total TEWIs indistinguishable from comparable electric equipment in the United States when an annual average CO_2 emission rate of 0.65 kg CO_2/kWh is used.

Several industry experts have suggested that the TEWI definition needs to include all the energy consumed in extraction, transportation, and processing of the fuel *before* it arrives at the power plant. It has also been suggested that time-of-day CO_2/kWh should be used for calculation of air conditioning TEWIs because different plants are used to meet the “peak loads” associated with summer air conditioning use (Beggs 1996). These two questions are partially addressed in Appendix A.

Direct-fired, triple-effect absorption shows a substantial TEWI advantage over current direct-fired, double effect equipment, especially for the lower, 350 ton, capacity ranges.

Cooling tower burdens for chiller technologies can be decreased by increasing the temperature difference for the condensing water from 5.6° C (10° F) to 8.3° C (15° F). A 5.6° C (10° F) ΔT was used for both the electrical and gas-driven technologies because the condenser or absorber/condenser water flow rates specified in ARI 550-92 and 560-92 Standards are consistent with a 5.6° C (10° F) condenser ΔT. The new 560 Standard which is out for comment calls for a lower absorber/condenser water flow rate with would raise the condenser water ΔT for direct fired absorption. Industry sources indicate that gas and newer, more efficient electric chillers are being installed in the field with condenser flow rates considerably less than those specified in the appropriate ARI Standard to help minimize life cycle costs.

No equitable method was found which permitted including a heat recovery benefit from engine driven equipment and/or a CO_2 credit for direct-fired absorption equipment. These b/enefits are quite site specific and depend heavily on the type of building, its geographic location, and the type of activity in the building. Heat recovery is most beneficial when it can be used to avoid consumption of additional fossil fuel for heating or regeneration or when there is a fixed ratio of cooling and heating demands in an application. Figure 27 shows the relative TEWI reductions possible with fixed percentages of waste heat recovery.

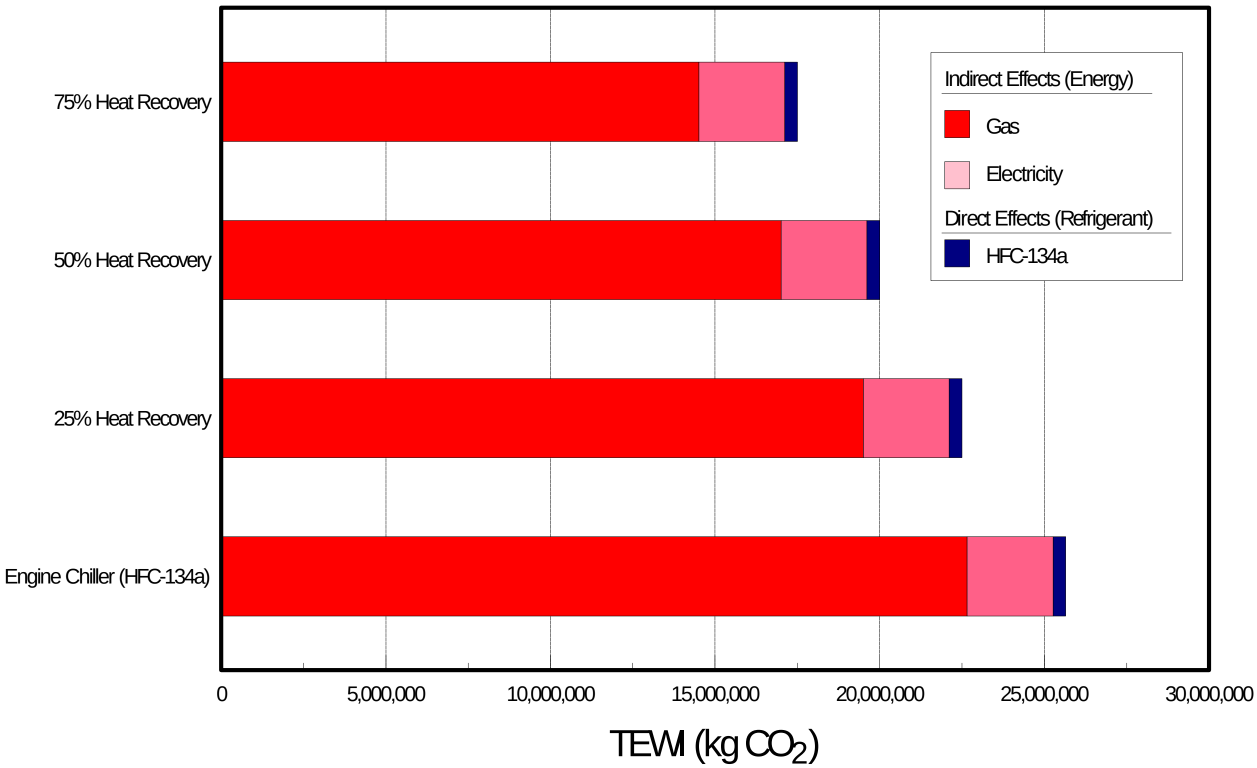

Figure 27. Benefits of heat recovery for chiller plants.

5.8 CONCLUSIONS

From the data presented in Figs. 20 and 21, it is clear that TEWI values for new electric chillers have improved by about 20 to 30% since the early 1990's. Approximately 75% of this improvement is due to a change from the CFC refrigerants which substantially reduced the *direct* TEWI. Substantial improvements in refrigerant handling and recovery practices have also contributed significantly to this *direct* TEWI decrease (Smithart 1996). Consequently, there is no significant TEWI advantage from using zero GWP refrigerants in a chiller.

The choice of working fluid among those currently being used as refrigerants for vapor compression chillers makes only a minor difference in direct TEWI. This is especially true now that low leakage rates, low service loss rates, and high efficiency purge systems have been implemented on this equipment. However, the differences in chiller efficiencies for various refrigerant options can make significant differences in the *indirect* TEWI. The *indirect* TEWI term remains dominant in this high energy use, long-life application with HCFC-123 machines showing a 10 to 15 percent advantage in total TEWI.

Longer annual operating hours decrease the relative direct TEWI contributions of refrigerants.

Direct-fired absorption chillers can be a preferred choice where they have greater value in their ability to use lower cost fuels and/or provide high quality waste heat for a useful application. In many chiller installations, absorption equipment is used for a much shorter time to handle peak air conditioning loads and avoid high electrical peak demand charges.

Gas fired technologies can show lower TEWI when simultaneous heating and cooling are required and heat recovery is used. Double bundle condensers used with electric chillers for waste heat recovery can also result in lower TEWI values.

6. COMMERCIAL REFRIGERATION SYSTEMS

6.1 INTRODUCTION

Commercial refrigeration is very a broad category which can include many diverse applications. Among these are supermarket refrigeration systems using twenty or more display cases for dairy products, meat and fish, frozen foods, and ice cream. These systems typically employ large racks of compressors in a machine room removed from the sales floor and condensers on the roof or in back of the store. Commercial refrigeration also includes “stand-alone” display cases at convenience and grocery stores that contain the entire mechanical package (e.g. compressor, condenser, evaporator), walk-in refrigerators and freezers at restaurants and hospitals, and ice makers in hotels, among others. Supermarket refrigeration systems, unlike the other applications mentioned, have long refrigerant lines which are susceptible to leaks and require very high refrigerant charges. Consequently, the direct effect of refrigerant emissions can be very high for supermarkets with distributed cases and long refrigerant lines; on the other hand, stand-alone display cases, walk-in freezers, and ice makers have smaller charges and much lower emission rates and are more like household refrigerators and unitary air conditioning in this regard. This study focuses exclusively on supermarket systems, and it includes the current standard technology as well as two alternative refrigeration systems.

6.1.1 Direct Expansion Systems

The traditional design for supermarket refrigeration systems is referred to as “direct expansion” because the high pressure refrigerant is circulated throughout the store directly to display cases where it is expanded and absorbs heat (maintaining the cold product temperature). Thousands of meters of piping can be required to connect the display cases to the remote compressor. The large internal volumes resulting from all of the piping require very large refrigerant charges, while the number of welded and brazed joints provide a great many sources for leaks. Also, the refrigerant itself must meet very strict safety requirements covering toxicity and flammability since it is being circulated throughout the retail sales floor and is in close proximity to customers and employees.

6.1.2 Secondary Loop Systems

Efforts to reduce the refrigerant charge required by direct expansion systems led to some test installations of refrigeration systems which use secondary heat transfer loops to pump cold brine solutions to the display cases on the sales floor. Replacing the piping of refrigerant with piping of brine allows the refrigeration system to operate with a refrigerant charge that can be as small as 10% of what is required for a comparable direct expansion system. Besides reducing the charge, secondary loop systems are less prone to leak because there are fewer welded and brazed joints in pipes containing refrigerant. A second potential advantage of these systems results from isolating the refrigerant from the sales floor; the refrigerant is no longer in proximity with the customers and less restrictive health and safety requirements may permit the use of flammable or toxic refrigerants. The introduction of a secondary loop, however, introduces a)T between the refrigerant and the brine which may require a lower evaporating temperature than is required for direct expansion systems. The lower evaporating temperature means lower relative efficiency and higher power consumption. Secondary loop systems also require a pump to circulate the brine, which is another addition to power consumption and affects the amount of heat that must be rejected by the condensers.

6.1.3 Distributed Systems

A second approach to reducing refrigerant charge and emissions has resulted in the development and marketing of what is termed here as “distributed systems.” In these installations, one or more compressors on small racks are distributed throughout the supermarket near the display cases they are serving. The compressors can be in attractive cabinets on the sales floor or they can be installed in unused space in back rooms or on top of walk in freezers and refrigerators (Broccard 1995). Commercially available distributed systems rely on a water loop to connect all of the compressor installations with a single cooling unit (e.g. cooling tower, evaporative cooler) on the roof or outside behind the store to reject the waste heat of the system. As with the secondary loop systems, this water loop introduces an additional)T and thermodynamic loss and pumping power not associated with direct expansion systems.

While TEWI are calculated and presented for all three classes of equipment, direct expansion, secondary loops, and distributed systems, insufficient information is provided to make absolute comparisons between the different technologies. Care was taken to select thermodynamic cycle conditions that were representative so that energy use calculations could then be compared; however, no effort was directed toward installation and maintenance costs comparisons for the three systems. This is a significant omission which needs to be covered before concluding that one system or another is “better” than another.

6.2 ASSUMPTIONS

6.2.1 Temperature Ranges

Refrigeration needs for supermarkets break down into three broad categories:

! high temperature refrigeration for air conditioning and cooling of prep rooms, typically providing cold air around 10EC (50EF),

! medium temperature refrigeration for meat and fish and dairy cases and walk-in coolers for meats and produce, with air temperatures from -2E to 7EC (28E to 45EF), and

! low temperature refrigeration for freezers and ice cream cases and walk-ins, -18E to -

32EC (0E to -25EF).

High temperature refrigeration is not considered in this study, since any analysis needs to consider interactions between the air conditioning system, the refrigerated display cases, and moist air entering the store. That degree of analysis is beyond what can be done in this study. Medium temperature refrigeration is simplified by assuming a single fixed evaporating temperature for all the display cases; -7EC (20EF) for direct expansion systems. A single evaporating temperature is also used to characterize all of the low temperature refrigeration; -32EC (-25EF) for direct expansion systems. Evaporating temperatures for the secondary loop calculations are assumed to be 3EC (5.5EF) lower than the corresponding temperatures for direct expansion systems (Likes 1996).

The high-side temperatures where heat is rejected depend on the outdoor ambient temperature, naturally, as well as the type of heat rejection unit used; an evaporative cooler will have lower temperatures than will an air-cooled unit. These calculations are based on a single condensing temperature, 36EC (97EF), for an entire year of operation and the same temperature is used for Europe, Japan, and North America. The condensing temperature for the distributed systems is assumed to be 3EC (5.5EF) higher to accommodate the condenser water loop mentioned earlier (Ares 1996). These assumptions are significant simplifications that vastly reduce the amount of work required in this analysis, but as a result small differences in calculated energy use should not be viewed as being significant.

6.2.2 Refrigeration Loads

Baseline refrigeration loads are assumed for Europe, Japan, and North America. It is assumed that a typical supermarket in North America requires 88 kW (300,000 Btu/h) of low temperature refrigeration and 264 kW (900,000 Btu/h) of medium temp (Hourahan 1996). A store in Japan is assumed to have 24 kW (82,000 Btu/h) and 127 kW (433,000 Btu/h) of low and medium temperature loads (Hara 1996). Store loads for Europe are assumed to be 50% of those in North America; 44 kW (150,000 Btu/h) low temp and 132 kW (450,000 Btu/h) medium temp.

6.2.3 Alternative Refrigerants

Historically, CFC-12, HCFC-22, and R-502 were used in supermarket refrigeration systems. Each of these either has been, or is being, phased out of use as a result of the Montreal Protocol. Many different refrigerants have been promoted as replacements for these three fluorocarbons, and manufacturers have gravitated toward marketing equipment using only two or three of the alternative refrigerants. The analysis presented here, however, includes more than just those two or three alternatives in an effort to determine if any other refrigerant has a clear advantage in reducing TEWI. Direct expansion and distributed systems are considered which include:

! five alternatives for medium temp; R-404A, R-507, R-134a, R-407A, and R-410A, and

- four alternatives for low temperature refrigeration; R-404A, R-507[1], R-407A, and R-407C.

Each of these refrigerants is also included in the analysis of secondary loop refrigeration systems, but those calculations and results also include ammonia, R-717, as an alternative. While the use of ammonia can be limited by building codes and ordinances at all levels of government, it is a viable alternative in this application and secondary loop systems with ammonia are being built and used in Europe (Haaf 1996). The compressor discharge temperature for ammonia can be much higher than it is for other refrigerants, and effort must be made in the design and application to keep temperature safely below the point where the oil would be damaged or destroyed (Ares 1996). These are important considerations in applying ammonia in commercial refrigeration and in comparing systems using ammonia with those using refrigerants with lower discharge temperatures, but they are primarily cost and design consideration and are not included in this analysis.

6.2.4 Refrigerant Charge

Assumptions relative to refrigerant charge and leakage rates are critical to any calculation of TEWI for this application, yet these numbers are either not well known and documented or they are widely variable. A general rule of thumb in American industry is that the number of pounds of refrigerant in a supermarket will be from 8% to 12% of the floor space in square feet (Broccard 1995); 40% to 60% of the floor space in kg refrigerant/m^2. Some of the large "mega-stores" being built can have refrigerant charges as high as 4000 kg (9000 lb). A recent survey showed the average store in a supermarket chain to be about 2300 m^2 (25,000 ft^2) (Progressive Grocer 1990); the average independent supermarket is smaller 1300 m^2 (14,000 ft^2) .

6.2.4.1 Direct Expansion

The baseline systems used to define the analysis are assumed to use 400 kg (880 lb) R-502 and 464 kg (1020 lb) HCFC-22 for North America; 200 kg (440 lb) R-502 and 232 kg (510 lb) HCFC-22 for Europe; and 79 kg (174 lb) R-502 and 277 kg (610 lb) HCFC-22 for Europe. The North American data correspond to an average sized store in a major supermarket chain based on the rule of thumb mentioned earlier (Progressive Grocer 1990); European data are arbitrarily set at 50% that of North America. A recent study by an industry trade group reported that the average supermarket in the U.S. uses 2700 kg (6000 lb) of HFCs to provide 530 kW (150 tons) of refrigeration (Bittner 1995); while this refrigeration load is consistent with the data used in this analysis,

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[1] ASHRAE has assigned the official designation of R-507A to the 50/50 mixture of HFC-125 and HFC-143a; that composition of 125/143a was commonly referred to as "R-507" prior to 1997 and is marketed in commercial products with that designation. The R-507 designation consequently is used throughout this report.

the refrigerant charge is significantly higher. This discrepancy has not been reconciled, but may be the result of the store size reflected by the survey and it may include refrigerant needs for air-conditioning and prep rooms. Refrigerant charges for alternative refrigerants are estimated using the baseline charge and the ratio of the liquid density of the alternative refrigerant at 21EC (70EF) to the density of the baseline refrigerant (i.e. R-502 or HCFC-22).

6.2.4.2 Secondary Loop Systems

The refrigerant charge for secondary loop systems ranges from 8% to 14% of the charge for a comparable direct expansion system (Hourahan 1996, Kruse 1993). An average value of 11% of the corresponding direct expansion charge is used in this analysis. The selection of a refrigerant charge for secondary systems is not crucial in the analysis because the combination of lower charge and lower emission rates results in a relatively insignificant direct effect in the TEWI.

6.2.4.3 Distributed Systems

Refrigerant charges for distributed systems can be reduced by 75% or more from what they would be for comparable direct expansion systems (Broccard 1995). This analysis assumes that the refrigerant charge for distributed systems is 25% of that for direct expansion systems.

6.2.5 Refrigerant Emission Rates

6.2.5.1 Direct Expansion Systems

Historically direct expansion supermarket refrigeration systems have had emission rates of 30% of the total charge per year or higher. These high rates were the result of using flared fittings in the piping to expedite service work on the sales floor, mixed metal welds, thermal shocks at flared fittings due to defrosting, and poorly or improperly supported pipes in long runs (Richey 1995). These rates were also the consequence of inexpensive refrigerant and expensive labor. Significantly higher refrigerant costs and enhanced environmental awareness have led to design changes in display cases and maintenance practices that are resulting in lower emission rates (Broccard 1996). Major supermarket chains have found that aggressive preventive maintenance programs can reduce emissions to 10% of the charge per year or less (Richey 1995). Haaf also reported a 10% emission rate for Europe (1996) and the Air Conditioning and Refrigeration Institute (ARI) reported current ranges of 12% to 15% of the charge which can be reduced to 4% to 8% of the charge per year in 5 to 10 years (Hourahan 1996). Calculations are reported using the averages of the ranges given by ARI for current and near term emission rates, 13½% and 6% respectively.

6.2.5.2 Secondary Loop Systems

Secondary loop systems are known to have lower leakage rates than direct expansion systems because there are fewer brazed and welded fittings on pipes containing refrigerant and the piping connections are more accessible to maintenance personnel for leak checking and repair. Leakage rates are also lower because there is not a refrigerant fitting at the display case where it would be exposed to the thermal shocks associated with defrosting the case evaporator. TEWI calculations are based on emission rates of 4% for current equipment and 2% for near term (5 to 10 years) systems (Hourahan 1996).

6.2.5.3 Distributed Systems

Distributed systems can also have lower leakage rates than direct expansion systems, but not as low as can be achieved in secondary loop systems. TEWI calculations for current technology are based on a 5% per year loss rate and 2% for near term (Hourahan 1996).

6.2.6 COPs

6.2.6.1 Direct Expansion Systems

Assumptions about the relative efficiencies of equipment using one refrigerant or another are probably the most controversial choices made in this analysis because there is an apparent advantage to one supplier's product over another. There are theoretical differences between the refrigerants, and differences have been demonstrated in laboratory and instrumented store testing. However, there are so many factors involved in store operation that the theoretical and laboratory differences in efficiency for refrigerants do not necessarily show up as differences in energy consumption comparisons between stores.

Different efficiencies, COPs, are assigned for each alternative refrigerant in this analysis for low and medium temperature refrigeration based on data reported in the open literature. First, a baseline COP is selected for R-502 for low temp and HCFC-22 for medium

Table 13. Low temperature refrigeration COPs.

Refrigerant	Direct Expansion	Secondary Loops	Distributed Systems
R-502	1.798	1.636	1.726
R-404A	1.762	1.603	1.692
R-407A	1.708	1.554	1.640
R-407C	1.654	1.505	1.588
R-507	1.762	1.603	1.692
R-717		1.638	

Table 14. Medium temperature refrigeration COPs.

Refrigerant	Direct Expansion	Secondary Loop	Distributed Systems
HCFC-22	3.605	3.281	3.389
HFC-134a	3.317	3.018	3.118
R-404A	3.425	3.117	3.219
R-410A	3.569	3.248	3.355
R-507	3.515	3.199	3.304
R-717		3.205	

temp, and then literature values for performance relative to those baselines are applied to get a COP for each alternative (Shiflett 1994, ASW 1994, Borhanian 1994, Haaf 1996). These data are summarized in Tables 13 and 14; COPs for ammonia, R-717, are shown <u>only</u> for the secondary loop systems because it is not considered a viable choice of refrigerant for applications where the refrigerant is in close proximity with customers or employees (note that R-502 and HCFC-22 are listed in Tables 13 and 14, but they are only used as baseline refrigerants to define COPs for the HFC mixtures and ammonia).

6.2.6.2 Secondary Loop Systems

While there are many comparisons of the performance of HFC mixtures in direct expansion systems, there is only a small body of information available for secondary loop systems and it is inadequate for comparing different refrigerants. Consequently this analysis relies on a theoretical degradation from the performance of direct expansion systems due to assumed differences in evaporating temperatures to estimate performance of refrigerants in secondary loop systems. The estimated COPs for secondary loop systems are also listed in Tables 13 and 14. As mentioned previously, it is assumed that the evaporating temperatures for systems using secondary loops are 3EC (5.5EF) lower than evaporating temperatures for direct expansion systems. The relative performance of secondary loop and direct expansion systems were determined using cycle calculations based on:

! low temp conditions of 36EC (97EF) condensing, 2.8EC (5EF) evaporator superheat, 4.4EC (40EF) return gas, 18EC (65EF) liquid line, and -32EC (-25EF) evaporating for direct expansion and -35EC (-30.5EF) for secondary loops to determine an average theoretical degradation of 9% for secondary loops relative to direct expansion, and

! medium temp conditions of 36EC (97EF) condensing, 2.8EC (5EF) evaporator superheat, 18EC (65EF) return gas, 18EC (65EF) liquid line, and -7EC (20EF) evaporating for direct expansion and -10EC (14.5EF) for secondary loops to determine an average theoretical degradation of 9% for secondary loops relative to direct expansion.

These factors are applied to the COPs for direct expansion to estimate the compressor power of secondary loop systems.

6.2.6.3 Distributed Systems

Similar calculations were performed to estimate degradation factors for distributed systems, although in this case the 3EC (5.5EF))T is applied to the condensing temperature of 36EC (97EF). Theoretically, based on these calculations, low temp distributed systems will have COPs 4% lower than direct expansion systems; medium temp COPs are 6% lower.

6.2.7 Compressor On-Time

Energy use calculations, as explained later, are based on compressor operation for some estimated equivalent full-load hours. Field measurements on installed equipment and annual energy use reported by Hara (1996) and Oas (1991) both correspond to the compressors running at full load 40% of the time.

6.2.8 Condenser Fan Power

Each of the alternative systems is required to meet its own corresponding set of thermal loads. The loads are comprised of the refrigeration load of the display cases, the compressor power, and pumping power for the secondary loop and distributed systems. The condenser fan power necessary to reject this heat is calculated for each system based on 18.3 W per kW of heat rejected.

6.2.9 System Lifetime

The assumption on equipment lifetime is important in evaluating the results presented later on an absolute basis, but perhaps surprisingly it is not important in the calculations. Power consumption and refrigerant emissions are all computed on an annual basis, and the assumed system lifetime is no more than a scaling factor applied to the annual results. This simplified analysis is a consequence of omitting any refrigerant losses at the end of product life; it is assumed that essentially all of the refrigerant charge is recovered when the refrigeration system is replaced. Estimates of equipment lifetime could be based on the frequency of remodeling the store and replacing the display cases (approximately seven years) or the approximately 20 year useful lifetime of the compressors and condensing unit (Fischer, et al 1991). The calculations and results presented in this report are based on an equipment lifetime of 15 years (Rosenstock 1997).

6.3 METHODOLOGY

TEWI is computed separately for low and medium temperature refrigeration for each of the three alternative technologies using each of the alternative refrigerants. The indirect effects are computed by estimating the annual and lifetime power consumption for the compressor and auxiliary fans and pumps; the total power consumptions are converted to lifetime CO_2 emissions using a single regional average factor for emissions from electrical power production. Direct effects from refrigerant emissions are computed using the annual emission factors for each technology, the equipment lifetime, and the GWPs for each refrigerant.

6.3.1 Power Consumption

Annual power consumption is computed as the sum of estimated compressor power, pumping power if any for the secondary loop or condenser loop, and condenser fan power. Power consumption for the evaporator fans has not been included. This omission is not significant in comparing TEWI for alternative refrigerants because including evaporator fan power affects each of the alternatives equally; including evaporator fan power would reduce the fraction of TEWI from direct effects slightly.

6.3.1.1 Compressor

Compressor powers are calculated by dividing the assumed refrigeration load by the COP corresponding to the application (i.e. low or medium temp; direct expansion, secondary loop, distributed system) for each alternative refrigerant. The annual energy inputs are then determined by multiplying these ratios of load and COP by the assumed number of equivalent full load hours, 3504 hours (i.e. 40% run-time and 8760 hours per year).

6.3.1.2 Secondary Loop Pumping Power

An effort is made to include an estimate of the secondary loop pumping power in the TEWI calculations, but it must be recognized that while attempting to be “realistic” that this effort is at best cursory. An accurate calculation of pumping power requires a detailed design of the secondary loop and an economic evaluation of trade-offs between materials and construction costs and operating costs. This analysis, while necessary for greater precision, is beyond the scope of this report. Whether good or bad, assumptions are made for these calculations in order to include an estimate of pumping power. These include:

- an equivalent length of piping between the display cases and the machine room (different lengths are assumed for Europe, Japan, and North America),
- the ΔT in the secondary loop acceptable at the display case to provide the cooling,
- the numbers of equal parallel loops servicing the low temperature and the medium temperature display cases, and
- the diameters of the piping used.

The allowable ΔT and the refrigeration load determine the flow rate necessary to meet the load on each parallel loop. The equivalent length, diameter, and flow rate determine the pressure drop and pumping power (for given coolant properties, several coolants were considered including both organic and inorganic salt solutions). In reality, the pipe dimensions and number of parallel circuits would be determined by cost factors; in this analysis these variables were chosen to approximate a European design value for a ΔP of 2 to 2.5 bar (30 to 36 psi) (Haaf 1996). Systems were selected for North America and Japan which gave pumping powers which are about the same fraction of compressor power as determined for the European systems. The details for these calculations are included in

6.3.1.3 Distributed System Condenser Loop Pumping Power

The calculational procedure for estimating pumping power for the condenser water loop of distributed systems is similar to that used for computing the pumping power for secondary loops.

6.3.2 Direct Effect

The portion of TEWI for each refrigerant in each application is a straightforward calculation using the equipment lifetime, the annual emission rate (as a percentage of refrigerant charge), the refrigerant charge, and the GWP for each refrigerant. It is assumed that essentially all of the refrigerant is recovered at the end of product life.

6.4 RESULTS

Tabular results and bar charts of all of the calculated TEWI are listed in Appendix I. One set of results is shown in Fig. 28 which shows groups of bars for the three alternative technologies; direct expansion systems, secondary loops, and distributed systems. Within each group, there are separate bars for the alternative refrigerants considered, and each bar is broken into three segments. Each of these segments corresponds to one factor in the global warming from supermarket refrigeration:

- the leftmost, heavily shaded segment, corresponds to indirect global warming from CO_2 emissions resulting from energy use,

- the center segment (where it can be seen) corresponds to the direct global warming effects of refrigerant emissions at the reduced rates that may be achieved in 5 to 10 years (2002 to 2007), and

- the lightly shaded segment on the right also due to refrigerant emissions, it is the increment over the near term rate due to the current (1997) higher emission rate.

The total length of each bar, then reflects the TEWI at current emission rates and the left two segments the TEWI at the lower future emission rates.

In a broad sense the information contained in Fig. 28 and Appendix I represent a reasonable comparison of the three technologies and alternative refrigerants considered, but it relies on many simplifying assumptions that affect the results. First, all of the energy use calculations are based on a single, average condensing temperature to estimate annual energy use. Second, although the assumed ΔTs and calculated pumping powers for the secondary loop systems are comparable to what has been

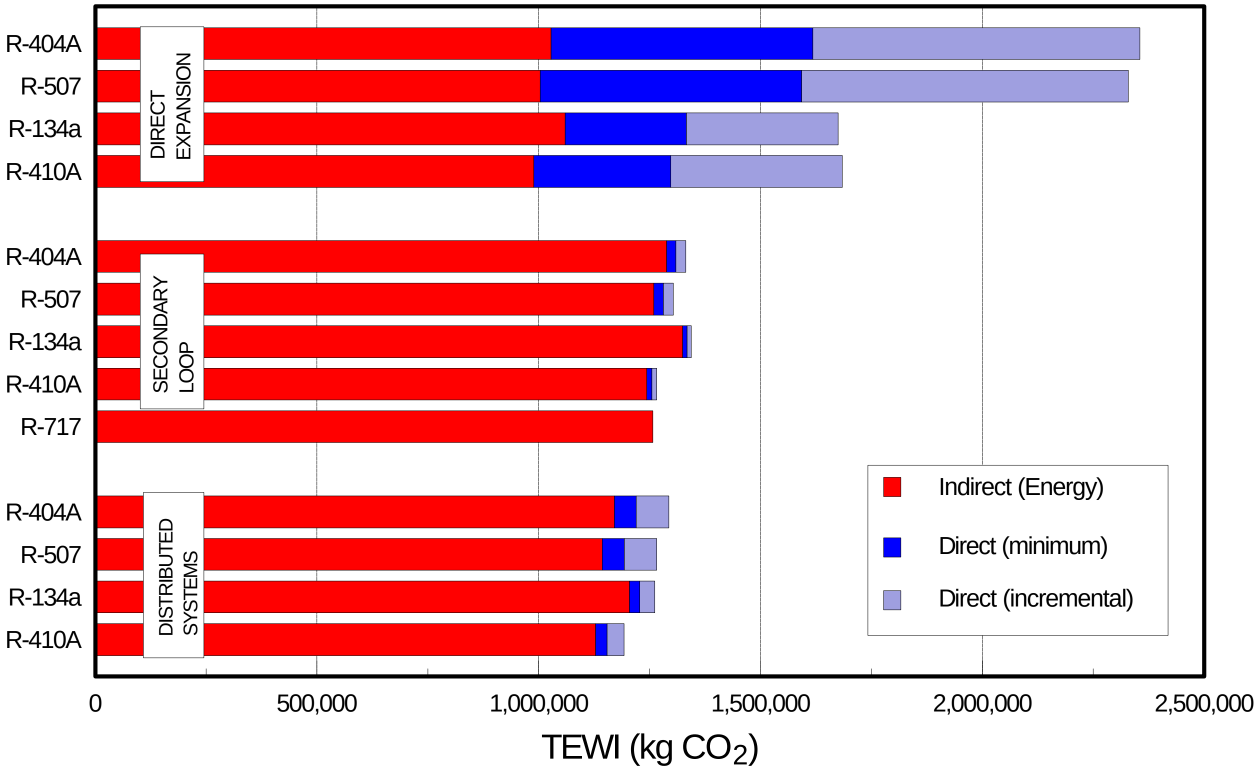

Figure 28. Medium temperature refrigeration in Europe.

experienced in actual installations (Boyko 1997), poor selection of heat transfer fluid could lead to higher pumping powers (Ares 1997, Boyko 1997) and design optimization for energy efficiency could reduce or eliminate the increases in compressor power (Boyko 1997, Gage 1997). While a more rigorous analysis and more optimal designs can lead to more precision in the estimated TEWI for each alternative, they are unlikely to change the relative comparisons shown for the three technologies or the different refrigerants.

6.5 CONCLUSIONS

Several conclusions can be drawn from the data presented in Appendix I, but it is essential to remember that a major factor has been omitted in this analysis. There are different equipment, installation, and maintenance costs associated with each of the three technologies presented (not to mention factors affecting the quality of the refrigerated products). The figures and tables show that:

! *direct expansion systems:* reduction of the emission rates from the currently achievable rates to what is believed to be achievable in 5 to 10 years (2002 to 2007) represents a significant reduction in TEWI,

! *direct expansion systems:* the major factor affecting differences in TEWI between alternative refrigerants is the GWP of each refrigerant,

- *secondary loop systems:* significant reductions in TEWI from current levels might be able to be achieved using secondary loop systems, but with a possible 13 to 26% increase in energy use relative to direct expansion systems (as shown in Appendix I). A more detailed analysis is required that includes construction, maintenance, and energy costs in order to evaluate the value of this technology in reducing TEWI, based on the assumptions used in this report,

- *secondary loop systems:* the use of a zero GWP refrigerant, ammonia (R-717), in a secondary loop system could result in lower TEWI than using a fluorocarbon refrigerant; TEWI for the fluorocarbons are 0 to 15% higher than the TEWI with ammonia,

- *distributed systems:* significant reductions in TEWI may also be achieved from current levels for direct expansion through the use of distributed refrigeration systems, but with an 8 to 15% increase in energy use. Evaluations of the equipment, installation, and operating costs and the possible loss of useful space on the sales floor is necessary to assess the value of this technology in reducing greenhouse gas emissions, and

- *secondary loop and distributed systems:* differences in TEWI for distributed and secondary loop systems are small, regardless of refrigerant choice, and in most instances are insignificant.

Three different methods of achieving lower TEWI are shown in these figures and information concerning the costs of each approach is needed to determine its merits as a technology for reducing emissions of greenhouse gases. The costs in equipment and preventive maintenance necessary to reduce refrigerant emissions from the currently achievable levels to those believed possible in 5 to 10 years are different from the costs associated with secondary loop systems. Unfortunately, a cost based analysis is beyond the scope of the current project and it is only possible to conclude that alternative technologies have the potential for dramatic reductions in TEWI for this application.

7. AUTOMOBILE AIR CONDITIONERS

7.1 INTRODUCTION

Automobile air conditioning was identified in previous studies as one of the few applications of fluorocarbon refrigerants where the direct effects of refrigerant emissions is a significant fraction of the TEWI (Fischer, et al 1991, Fischer, et al 1994). While not contesting that conclusion, those studies have been criticized because they relied on efficiency data at a single design point and an estimated equivalent full-load hours of operation. Those two simplifying assumptions cannot account for varying performance over a range of operating conditions or the effects of different climates. The present analysis addresses these concerns; it incorporates efficiency differences across a wide range of operating conditions, regional variations in outdoor ambient temperature, and changes in air-condition on-time with outdoor temperature. While not perfect, this more detailed analysis should provide a better comparison of fundamentally different cooling cycles.

Three different cooling systems are considered in this analysis; a conventional system using HFC-134a, a conventional system employing a hydrocarbon refrigerant, and the transcritical CO_2 (R-744) system. All three systems are assumed to consist of an orifice tube, a positive displacement compressor with a constant displacement, an accumulator, and a plate fin evaporator. The HFC-134a and hydrocarbon systems have a tube and fin condenser while the CO_2 system uses a high-side gas cooler. Flammable refrigerants, and hydrocarbons in particular, have not attracted a great deal of support in the automobile industry because of the increased hazards from circulating the refrigerant through the evaporator in the passenger compartment. Hydrocarbon refrigerants have been specifically excluded from use for automobile air conditioning in the U.S. and are looked on with disfavor in other countries. The proposed hydrocarbon system in this study includes the use of a secondary heat transfer loop which would isolate the flammable refrigerant in the engine compartment so there would be no significant increased risk to the passengers. This essential safety feature affects the system thermodynamics, reducing its efficiency relative to traditional direct expansion systems, adds parasitic power consumption to pump the heat transfer fluid, and increases the overall system weight. Advocates of hydrocarbon air-conditioning systems have considered both propane (HC-290) and isobutane (HC-600a) as refrigerants; there are very minor differences between these two refrigerants, as far as this study is concerned, and only HC-600a is considered here.

Although the analysis includes much more detailed information than earlier studies, overall it relies on the same approach of evaluating energy use for air conditioning, energy use for transporting the system, and the direct effect from refrigerant emissions. Results are computed for thirteen different countries

7.2 ASSUMPTIONS

7.2.1 Cycle Efficiencies

Cycle efficiencies are calculated using refrigerant properties at specified operating conditions. These include evaporating temperature, high-side heat transfer temperature (i.e. condensing temperature, gas cooler exit temperature), liquid subcooling, and compressor suction temperature. The transcritical cycle also includes high-side to low-side heat transfer.

Previous calculations of TEWI for automobile air conditioning were based on equivalent full load hours of operation at a single design condition (Fischer, et al 1991, Fischer, et al 1994). The conclusions based on this simplified approach may overstate the effects of energy use because the efficiency at design conditions is lower than it is for most of the operating hours. A more detailed calculation is performed in this analysis that accounts for changes in the evaporating and condensing temperatures, and subcooling based on the vehicle speed and ambient temperatures.

7.2.1.1 Evaporating Temperatures

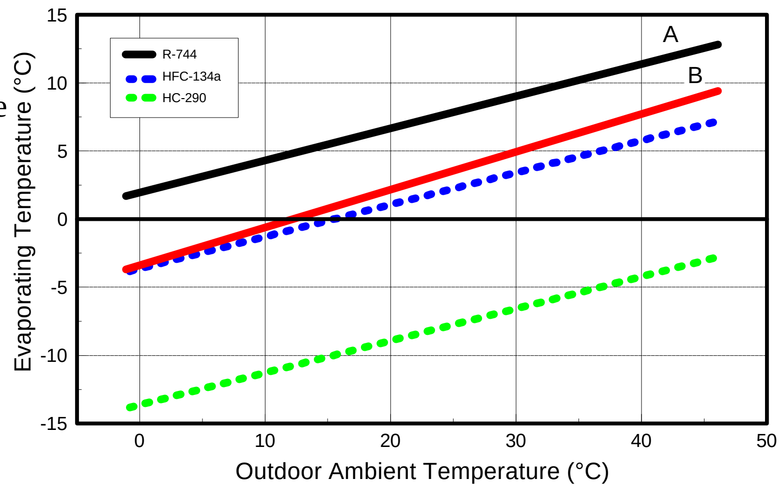

Figure 29. Evaporating temperatures.

The outdoor ambient temperature affects the refrigerant temperature in the evaporator and the evaporating temperatures are not the same for HFC-134a, hydrocarbons, and CO_2 because of heat transfer characteristics and additional)T's imposed by secondary loops. Evaporating temperatures for HFC-134a vary between -4EC and 7EC (25EF and 45EF) for the lowest and highest ambient temperatures (Patti 1996); additionally, typical evaporating temperatures are -1EC, -11EC, and 4EC (30EF, 12EF, and 40EF) for HFC-134a, hydrocarbons, and CO_2, respectively (Köhler 1996). These data are shown in Fig. 29 where the dotted line represents the evaporating temperatures for an air conditioner using a hydrocarbon refrigerant (propane), the dashed line for HFC-134a, and the two solid lines for CO_2. While computer simulations using the temperatures in Fig. 29 show similar evaporator air-side performance for the hydrocarbon and HFC-134a systems, this is not the case for the CO_2 air conditioner using the evaporator temperatures for prototype systems (line A in Fig. 29). Calculated evaporator discharge air temperatures in that instance are 1.7E to 4.4EC (3E to 8EF) higher than those computed for the HFC-134a and hydrocarbon systems. These differences in supply air temperatures, and accompanying differences in latent capacity, may not seem significant in mild climates, but they could have large impacts on passenger comfort in climates with high humidity and/or large air conditioning loads. A second set of evaporating temperatures was defined for the CO_2 air conditioner that would provide simulated air-side performance equivalent to the HFC-134a system (based on evaporator effectivenesses of 85% and 90% for the HFC-134a and CO_2 systems, respectively). These evaporating temperatures for CO_2 are shown as line B in Fig. 29.

7.2.1.2 Approach Temperatures

This report defines the approach temperature as the difference between the ambient air temperature and the temperature of the refrigerant (vapor or liquid, saturated or subcooled) leaving the condenser or the gas cooler. The approach temperature affects the high-side operating pressure for all three air conditioning systems. It determines the condensing temperature, and hence the condensing pressure, for the subcritical HFC-134a and hydrocarbon systems. The high-side pressure is chosen for the transcritical CO_2 system to give the maximum compressor COP for each gas cooler exit condition, and hence for each desired ambient temperature.

There are differences in the approach temperatures when vehicles are moving and when they are idling in a stationary position. While moving, the approach temperature is essentially the)T between the ambient air and the refrigerant. When stationary, there is a significant amount of engine heat recirculated back through the radiator and condenser as well as heat radiated from the pavement. Approach temperatures are also different depending on the refrigerant because of their individual thermophysical properties and heat transfer characteristics. The assumed approach temperatures are listed in Table 15 (Pettersen 1997).

Table 1.. Automobile air-conditioning high-side approach temperatures.

Refrigerant	Idle	Highway
HFC-134a	20EC (36EF)	7EC (13EF)
hydrocarbons	20EC (36EF)	7EC (13EF)
CO_2	20EC (36EF)	7EC (13EF)

7.2.1.3 Subcooling And Return Gas Temperatures

The subcritical systems either have subcooled refrigerant or saturated liquid leaving the condenser. This exit state depends on the operating conditions as well as the refrigerant charge; a system designed to have subcooling with a full charge may not have subcooling at lower charges experienced as refrigerant leaks from the system (Burk 1996). This analysis uses the simplified assumption that HFC-134a and hydrocarbons operate with no subcooling under idle conditions and 8EC (15EF) subcooling at highway speeds. Subcooling is not a factor in transcritical systems. The compressor suction gas temperature is 18EC (65EF) for the HFC-134a and hydrocarbon air conditioners; it is 5EC (9EF) below the gas cooler exit temperature for the CO_2 system because of the assumed high/low-side heat exchanger. There is an assumed pressure drop for the CO_2 system of 60 kPa (8.7 psia) at idle conditions and 150 kPa (21.7 psia) under highway conditions for the accumulator and high/low side heat exchanger.

7.2.2 Compressor Efficiency

Compression efficiencies vary with the shaft speed and between refrigerants. Only two shaft speeds are considered in this analysis; a low speed corresponding to idle and vehicle speeds below 16 km/h (10 mph) and a high speed for all other operating conditions. These two shaft speeds have been arbitrarily chosen to be 900 and 2000 rpm, respectively. These values affect the compressor efficiency, condenser subcooling, approach temperatures, and volumetric efficiencies. The assumed compressor efficiencies are listed in Table 16 (Köhler 1997).

Table 2.. Compressor isentropic efficiencies.

Refrigerant	Idle	Highway
HFC-134a	65%	60%
hydrocarbons	65%	60%
CO_2	70%	65%

7.2.3 Clutch / Transmission / Belt Efficiency

Each of the air-conditioning systems considered requires some sort of drive mechanism to operate the compressor from the engine drive shaft. Traditionally, a clutch and belt system is used for subcritical HFC-134a air conditioners and a similar system could be employed for hydrocarbons. This equipment is inappropriate to modulate the operation of a transcritical CO_2 system because of the high operating pressure and the relatively constant volumetric compressor efficiency. This analysis does not consider alternatives to traditional drive systems that may be suitable for CO_2; the calculations are based on a drive efficiency of 95% for all three types of air conditioners.

7.2.4 Auxiliary Power Consumption

Electric parasitics degrade system performance from the compressor only efficiency or COP. It is assumed that each of the systems considered uses an evaporator blower drawing 250 W and that the blower has the same operating time as the compressor, which is not strictly true. It is also assumed that a pump is used to circulate the secondary fluid for the hydrocarbon systems that draws 150 W, inclusive of a 50% pump efficiency. The alternator efficiency is 50%.

7.2.5 COP At Idle And Highway Conditions

Cycle efficiencies are calculated using the assumptions listed above with refrigerant properties calculated by commercially available computer software (F-Chart 1996, NIST 1996). Figures 30 and 31 show the system COPs at highway and low speed/idle conditions, respectively. Two curves are shown for CO_2 corresponding to the evaporating temperatures for prototype equipment and equivalent air-side performance discussed in Section 7.2.1.1. Requiring equivalent evaporator discharge air temperatures reduces the COPs for CO_2 by 9% to 15% from those computed using the evaporating temperatures from line A in Fig. 29. As mentioned earlier, the high-side pressure of the CO_2 system was selected in order to obtain the highest compressor only COP within limits of 7,600 to 15,000 kPa (1100 to 2175 psia). The lower pressure limit is set in order for the refrigerant to remain above the critical point, the upper limit is from the design of a prototype system (Pettersen 1994, Pettersen 1997, Fernqvist 1997, Denso 1997).

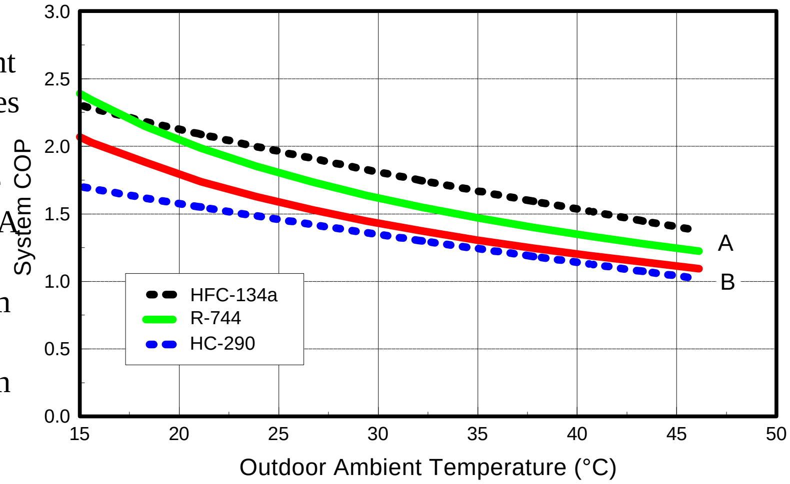

Figure 30. Air-conditioner system efficiency at low engine speeds.

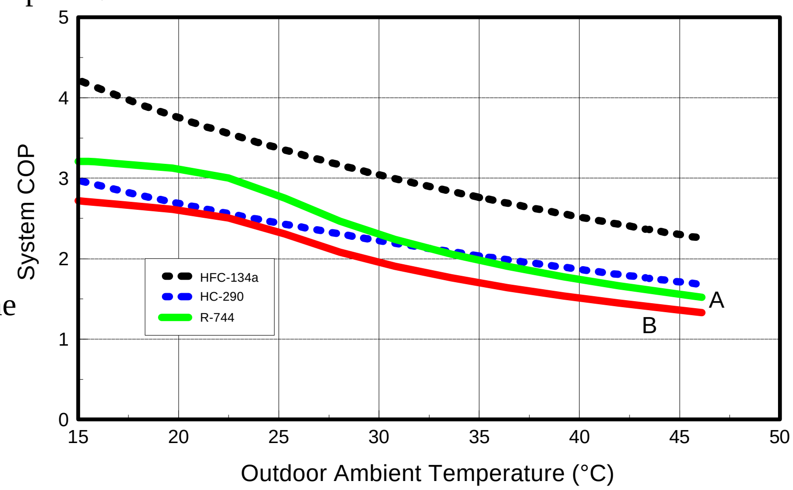

Figure 31. Air-conditioner system efficiency at high engine speeds.

7.2.6 Air-Conditioner Weight

Fuel is burned throughout the year, discharging CO_2 into the atmosphere, simply as a result of transporting the weight of the air conditioner and associated equipment. Heavier air conditioners result in greater amounts of CO_2 being discharged than lighter systems. The HFC-134a air conditioner is assumed to weigh 11.4 kg (25 lb) (Taylor 1996); while the hydrocarbon and CO_2 air conditioners is assumed to weigh 15.4 kg (39 lb). The additional weight for the hydrocarbon air conditioner is due to the refrigerant-to-brine and brine-to-air heat exchangers, the intermediate brine itself, and the pump (Pettersen 1996). The additional weight for the CO_2 system is believed to be necessary in order to provide the cooling capacity and COPs used in the TEWI calculations (Fernqvist 1997). The system weights used should exclude components of the heating and ventilating systems which are not specific to the air-conditioning system (e.g. blower motor) and add 1.5 to 2 kg (3.3 to 4.4 lb) to the vehicle weight (Nonnenmann 1997)

The weight effect of CO_2 emissions is directly proportional to annual vehicle usage (km or miles). The average distances for 1996 are assumed to be 16,100 km/y for Europe, 10,300 km/y for Japan, and 21,900 km/y for North America based on trends for the past 10 years in each region (Davis 1994).

7.2.7 Refrigerant Emissions And Direct Effect

The emission of HFC-134a from automobile air conditioners through leaks from hoses, fittings, and shaft seals, servicing, and accidents can contribute to global warming because of the relatively high GWPs of this gas; the corresponding direct effect from hydrocarbons or CO_2 would be negligible because of the very small GWPs of these gases. The magnitude of the direct effect from HFC-134a depends directly on how well the charge is contained and how it is handled during servicing. Historically, a single automobile air conditioner that used CFC-12 resulted in the release of approximately five full charges of refrigerant during its operating lifetime:

- systems required servicing every three years, on an average,
- any charge remaining in the air conditioner was vented to the atmosphere during servicing, and
- a full charge of refrigerant was used to diagnose the problem with the air conditioner before repairing the system.

The redesign of air conditioners to use HFC-134a included improvements to hoses and fittings which dramatically reduced refrigerant emissions and lengthened the service interval. It is estimated that vehicle owners will bring their cars in for servicing after the air conditioner has lost 40% of its charge and that systems will average 1½ to 2 service visits during their operating lifetime (Baker 1997).

Service practices, as mentioned, were also a major source of refrigerant emissions in the past when refrigerants were inexpensive and their environmental effects were unknown. Economics and attitudes toward the deliberate venting of refrigerants have changed significantly since the Montreal Protocol was signed and most major developed countries have either adopted or are considering regulations prohibiting the deliberate release of refrigerants to the atmosphere. These regulations are also likely to mandate the recovery of refrigerant remaining in air-conditioning systems when they are scrapped. Changing service practices should limit emissions of refrigerant to only what is required for recharging (40% of the original equipment charge) and recovery should limit end-of-life emissions to no more than 10% of the charge remaining in the system when it is scrapped.

Three different scenarios are considered for the direct effect of refrigerant emissions based on the assumptions listed above:

- a minimum value calculated using leakage of 35 g/y for a new vehicle (Zietlow 1997, Hara 1996, Fernqvist 1997),
- emissions from 1½ recharges during the operating lifetime of the air conditioner plus recovery of 90% of the remaining charge when the vehicle is scrapped, and
- emissions from 2 recharges during the lifetime plus recovery of 90% of the remaining charge

when the vehicle is scrapped.

These scenarios are illustrated in Figs. 48, 49, and 50 in Appendix J; they correspond to lifetime emissions of 415 g, 860 g, and 1080 g of HFC-134a, respectively, for an original equipment charge of 1000 g (Baker 1996, Nonnenmann 1996). No effort is directed in this analysis toward the technical problems associated with containing hydrocarbon or CO_2 refrigerants in systems using flexible hoses; these refrigerants have smaller molecules than HFC-134a and may operate at significantly higher pressures (as much as 5 times as high for CO_2) and leakage rates could be quite higher than they are projected to be for HFC-134a. Higher emissions of these low GWP refrigerants would have a strong effect on system performance, maintenance requirements, and customer satisfaction, but not on TEWI.

7.3 METHODOLOGY

7.3.1 Ambient Temperature Distributions

Ambient temperature profiles are used for locations; twelve in Europe (two in the U.K., four in Germany, two in Greece, two in Italy, and two in Spain), four in Japan, and four in North America (USAF 1978). Data for Europe are averaged to obtain five national values which are used in the TEWI calculations. Automobiles are manufactured for broad markets, and consequently the air-conditioning systems are designed according to the requirements necessary to satisfy customers in the hottest, most humid climates where the cars will be sold. TEWI are presented only for the most demanding temperature profiles for Europe, Japan, and North America.

Figures 51 to 53 in Appendix K show the number of hours per year that the outdoor ambient temperature is in a 5.6EC (10EF) temperature bin for a typical meteorological year. The temperate climate of Greece in Fig. 51 shows a distribution shifted toward lower temperatures while the profiles for Okinawa, Japan and the Southwestern U.S. show more hours at higher temperatures. Figure 53 shows a significant number of hours at the design condition of 35EC (95EF).

7.3.2 Compressor On-Times

The outdoor temperature is related to compressor on-times using a curve derived from manufacturer’s fleet data in Phoenix, Arizona USA (Fernqvist 1996). The compressor percent on-time is correlated against the daily average temperature, as shown in Fig. 32.

7.3.3 Idle and Highway Operation

The final results of the TEWI calculation incorporate a weighted average of energy consumption for idle and low vehicle speeds (e.g. 900 rpm) and highway speeds (e.g. 2000 rpm). It is assumed that driving patterns for Europe and North America are similar and that 15% of vehicle use is at low engine speeds (and low compressor shaft speeds) and 85% at high engine speeds (Siewert 1983). Vehicle usage in Japan is significantly different than in Europe or North America. Approximately 47% of vehicle operation is at idle conditions and 53% at highway speeds (Denso 2 1996).

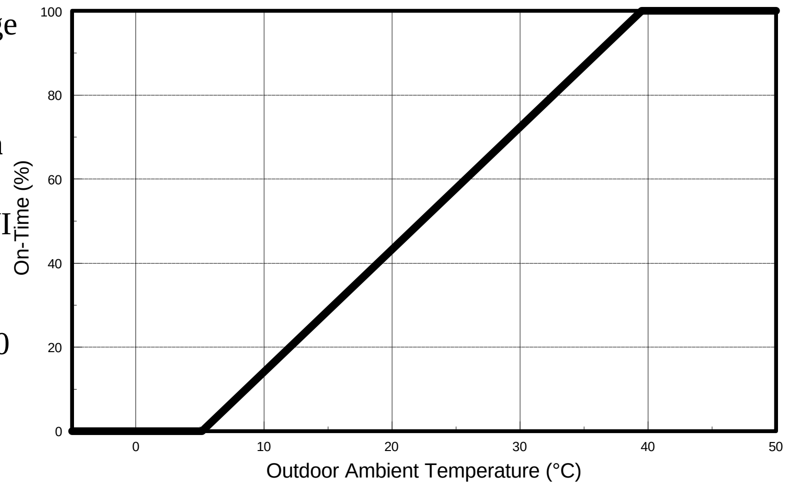

Figure 32. Compressor operation and average daily temperature.

7.3.4 Cooling Capacity

Compressor power at each ambient temperature is computed using the COPs shown in Figs. 30 and 31, the compressor on-time as a function of ambient temperature shown in Fig. 32, and constant cooling capacities of 7 kW (24,000 Btu/h) and 3.27 kW (11,000 Btu/h) at highway and idle conditions, respectively. The compressor on-time determines the cooling load as a function of the ambient temperature, in these calculations, and the analysis does not differentiate between the sensible and the latent loads. A more rigorous evaluation is required to account for the effects of the latent load (Bhatti 1997).

7.3.5 TEWI Calculations

7.3.5.1 Power Consumption For Air Conditioning

System power consumption is calculated at each ambient temperature using the temperature distributions (e.g. Figs. 51 to 53), annual driving distance, and a correspondence between driving distance and hours of vehicle operation. Davis (1994) reported data for average annual vehicle driving distances for the U.S., Japan, and many European countries. This information is used in conjunction with data on hours of vehicle use to determine air-conditioning power consumption (Zietlow 1997, Hara 1996, Siewert 1983). It is assumed that the annual hours of operation have the same distribution as the ambient temperature in order to estimate the vehicle hours at each ambient temperature. Annual cooling output at each ambient is calculated at low and high speeds using the number of driving hours, compressor percent on-time, and cooling capacities at low and high speeds. Annual power consumption is then the sum of the cooling output at each ambient divided by the corresponding system COP.

7.3.5.2 Incremental engine efficiency

Overall the thermal efficiency of gasoline and diesel engines is between 25% and 30% for operating the vehicle under full load conditions (Fischer, et al 1991, Nonnenmann 1996, Denso 1996). Although used in the past (Fischer, et al 1991, Fischer, et al 1994) this range of values is considered inappropriate for accounting for the incremental energy use resulting from operating the air conditioner; values of 40% (Nonnenmann 1996) and 50% (Wertenbach 1996) are cited. TEWI calculations in this analysis use an incremental engine efficiency of 40%.

7.3.5.3 CO_2 Emissions From Air-Conditioning Energy Use

The total energy requirement for air-conditioner operation is divided by the incremental engine efficiency to obtain the energy input to the vehicle engine for air conditioning. The CO_2 emissions associated with this energy input (kWh) are determined based on 0.243 kg CO_2 per kWh of input (Fischer, et al 1994). The annual value is multiplied by the assumed air-conditioner lifetime to obtain a lifetime value for CO_2 emissions from air conditioning.

7.3.5.4 CO_2 Emissions From Transporting The Air Conditioner

The fuel use necessary for transporting the air conditioner is calculated for each of the three systems by multiplying the assumed weight by 57×10^{-6} liters/kg/km times the regional annual vehicle use (km) (Fischer, et al 1991). The fuel use is converted to CO_2 emissions based on 2.32 kg CO2 / liter of gasoline (Fischer, et al 1994). This value is also multiplied by the lifetime to get total CO_2 emissions for system weight.

7.3.5.5 Direct Effect Of Refrigerant Emissions

Direct effects of refrigerant emissions are calculated based on the estimated lifetime emission of the refrigerant and the refrigerant GWP. The direct effects of the hydrocarbon and CO_2 systems are considered negligible and no effort is given to refining estimates of emission rates.

7.4 RESULTS

TEWI are calculated for the HFC-134a air conditioner using the three refrigerant emission scenarios identified earlier and for the hydrocarbon and CO_2 systems with emissions relative to two system recharges for HFC-134a. These results are summarized in Table 17 for three climates with high air-conditioning demand (i.e. the southwestern U.S., Greece, and Okinawa, Japan). Results for CO_2 are shown for calculations based on prototype equipment (column A) and equivalent discharge air temperatures (column B). All four sets of results differ substantially between the regions (shown) as they do within each region (not shown). This information is also summarized in Fig. 33; detailed results for all thirteen climates are in Appendix L.

Table 17. TEWI for HFC-134a, hydrocarbon, and transcritical CO_2 air conditioners.

	Refrigerant			
	HFC-134a	Hydrocarbon	CO_2	
Region			A	B
Europe				
A/C Energy	1255	1838	1547	2048
Weight	245	331	331	331
Direct Effect	540 to 1404	4	1	1
TEWI	2040 to 2904	2173	1878	2380
Japan				
A/C Energy	972	1408	1147	1694
Weight	154	167	167	167
Direct Effect	507 to 983	3	0	0
TEWI	1633 to 2109	1578	1314	1861
North America				
A/C Energy	2889	4160	3759	4902
Weight	363	491	491	491
Direct Effect	540 to 1404	4	1	1
TEWI	3792 to 4656	4655	4240	5393

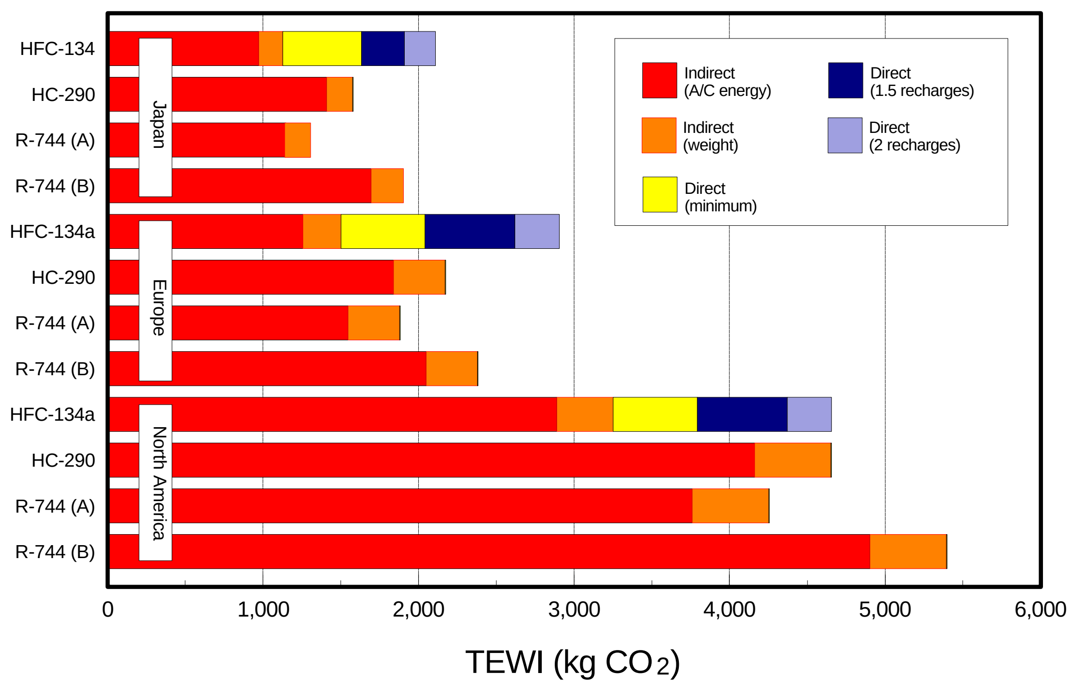

Figure 33. TEWI for HFC-134a, HC-290, and CO_2 (R-744) air conditioners in selected design climates.

The bar chart shows results for all three emissions scenarios, although the direct effects are so small for HC-600a and R-744 that they are not visible. Groups of bars are shown for Japan, Europe, and North America; each group contains bars for the TEWI of air conditioners using HFC-134a, HC-600a (isobutane), and R-744 (CO_2). The heavily shaded segment of each bar corresponds to the CO_2 emissions from fuel use necessary to drive the compressor. This segment is shortest for HFC-134a for each region. The second segment of each bar corresponds to fuel use to transport the weight of the air conditioner. The three segments that appear on the right of the bars for HFC-134a correspond to the three emissions scenarios; a minimum value based on estimates of leakage for systems straight from the factory, and increment corresponding to how much more refrigerant would be lost assuming 1½ recharges, and a second increment assuming 2 recharges during the lifetime of the air conditioner. In other words, the bars for HFC-134a show the TEWI assuming 2 system recharges during the lifetime; reducing emissions to 1½ recharges eliminates the rightmost segment; the lowest possible TEWI for HFC-134a is shown by eliminating the two segments on the right (one air conditioner manufacturer believes they can cut emissions lower than the "as manufactured" rates assumed in this analysis). It should be noted that different interpretation of the assumptions used in this analysis and the incorporation of latent cooling loads can alter the results from those presented here (Bhatti 1997).

7.5 CONCLUSIONS

The information in Table 17 and the detailed results in the appendices allow some general conclusions:

- ! under most of the scenarios considered the TEWI of CO_2 air conditioners is lower than hydrocarbon or HFC-134a air conditioners; it is comparable to to higher than that of HFC-134a in the regions of North America with high air-conditioning loads depending on assumptions for the emission rates of HFC-134a and the evaporating temperatures (and corresponding) discharge air temperatures for CO_2.

- ! fuel consumption is lower for air conditioners using HFC-134a than for either of the two alternative refrigerants under all of the climate and driving conditions considered. The transcritical CO_2 system requires 14 to 66% more energy, and fuel, for power and transport the air conditioner than HFC-134a and hydrocarbons 35 to 45% more than HFC-134a. It should be noted that fuel use for air conditioning has not been a major issue in the past, and the assumptions used here are for an HFC-134a system that has not been designed to provide maximum efficiency.

- ! estimated TEWI of the hydrocarbon system is generally lower than that of the HFC-134a systems in cool climate regions; it is comparable to or higher than that of HFC-134a in areas with high air conditioning loads.

8. RESIDENTIAL GAS HEATING/COOLING OPTIONS

8.1 INTRODUCTION

In regions where natural gas is available as an energy source, additional technology options and combinations are possible for unitary heating and air conditioning. Natural gas is used in commercial and residential applications because of its convenience and affordability. The U.S. Department of Energy (DOE) ranks natural gas as having the lowest average energy cost per therm when compared with refined petroleum products, liquefied propane (LPG), and electricity. Natural gas forced air or hot water furnaces in combination with an electrically powered air conditioner is a popular option for residential comfort conditioning. Unitary air conditioning units and heat pumps that use gas as the primary source of energy are currently available and under development.

8.2 UNITARY GAS TECHNOLOGIES

Gas furnaces in combination with a centralized, vapor-compression air conditioner is evaluated to provide a baseline for TEWI comparisons. In appropriate locations an electric resistance heat option is assessed.

8.2.1 Gas Engine Driven Heat Pumps

Engine-driven heat pumps rely on the same vapor compression cycle for heating and cooling as conventional electric-powered systems except the electric motor is replaced by an internal combustion engine which is usually powered by natural gas. Space heating capacity is supplemented by engine and exhaust heat recovery. Local market conditions and factors such as the relative costs of gas and electricity and local utility incentives rather than total TEWI are the main criteria used to choose one fuel over the other. The seasonal heating performance of a gas engine heat pump available since 1994 is listed at 126% AFUE[1] by the Natural Gas Cooling Equipment Guide (AGCC 1996). TEWI values were computed for this system using its published efficiencies.

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[1] AFUE - Annual Fuel Utilization Efficiency - appliance heating efficiency calculated by assuming 100% of the fuel is converted to thermal energy and then subtracting losses for exhausted sensible and latent heat, cyclic effects, infiltration, and pilot losses over the whole year. AFUE does not include electrical energy used for fans, pumps, ignition, exhaust, or blowers.

8.2.2 Gas-Fired Absorption Heat Pumps

Absorption heat pumps using ammonia-water are under development to provide heating and cooling for residential and light commercial applications. One product, based on the generator absorber heat exchange (GAX) cycle, has reached the proof-of-concept stage under a project sponsored by the U.S. Department of Energy and major American manufacturers have seriously considered hardware introduction (AGCC 1996). A small number of ammonia-water GAX heat pump prototypes have been built in the U.S. for testing and, optimistically, a product could enter the market in 1998; a more realistic date for market entry is 2000 (Fiskum 1996, Erickson 1996).

The GAX heating efficiency exceeds that of a gas furnace by 20% to 80% depending on the furnace efficiency used for the comparison. Conservative target steady-state efficiencies are 1.4 COP heating (gas-fired) at 8.3EC (47EF) and 0.9 COP cooling (gas-fired) at the DOE/B (82EF) 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 heat and mass transfer surfaces for the absorber and generator which are required to maintain the desired temperature and concentration profiles needed to achieve the full performance potential 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 (8.3EC, 47EF) and 0.75 cooling (28EC, 82EF), including flue losses but excluding electric parasitics (DeVault 1994).

There are strong prospects for successful commercialization of the GAX concept for heat pump applications if the remaining technical obstacles are overcome. Masala (1993) 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-condensable gases in the refrigerant/absorbent loop,

! reducing parasitic electric power consumption, and

! developing cost effective manufacturing techniques and quality control.

TEWIs are calculated for a GAX gas absorption heat pump in this section based on the same projected heating and cooling COPs used in the previous AFEAS/DOE study which included parasitic loads (Fischer 1994).

8.3.3 Desiccants

Desiccant materials which absorb or adsorb water vapor from the air can be used effectively in building air-conditioning systems. Sprayed-liquid desiccants or solid desiccants that are bonded to a porous substrate are used to dehumidify building ventilation air rather than cooling it below its dew point which is the approach used by conventional air-conditioning equipment. This process produces dry, but hot air since the water vapor gives up its latent heat of vaporization in the process. At this point of a desiccant cycle, no useful work has been accomplished because the enthalpy of the “wet” and that of the “dry” air are essentially the same (Collier 1996). So, some means for sensibly cooling the air after it has been dehumidified is needed before it can be used in a conditioned space. The energy efficiency of desiccant cooling applications is facilitated when this cooling can be accomplished via low energy input processes like direct or indirect evaporative coolers or an air-to-air heat exchanger. Use of waste heat to regenerate the desiccant also improves net efficiency.

Various hybrid cycles and combinations of HVAC equipment are currently used with desiccants to provide low humidity air in hospital operating rooms, supermarkets, and hotels. Dehumidification with desiccants allows building owners to install lower capacity air conditioning systems (because the desiccant is handling the latent building load), exchange gas energy or waste heat for electrical energy, and provide ventilation air with better indoor air quality. Because desiccant systems tend to be designed on a custom basis and since they are usually used in combination with a large variety of conventional air conditioning equipment, they were not evaluated in this report.

8.3 ASSUMPTIONS

National annual averages are used for the power plant emission rates in these calculations. The averaged electrical power plant emission rates are 0.650 kg CO_2/kWh for North America and 0.470 kg CO_2/kWh for Europe (see Appendix A). These emission rates are compiled from the open literature rather than calculated from the fundamental heat contents of fuels, fuel mix used in power production, plant efficiencies, and transportation and distribution losses. Total CO_2 production is divided by delivered kWh so all of the power plant conversion and distribution losses are accounted for.

The heat content and carbon dioxide emission rate for natural gas which were used for the gas powered technologies were 38,200 kJ/m^3 and 51.1 g/MJ, respectively. A 96.5% distribution efficiency was assumed for natural gas which raised the CO_2 emission rate to 53.0 g/MJ (55.9 g CO_2/1000 Btu) at its point of use (EIA 1997).

Fifteen year lifetimes are assumed for U.S. and European unitary equipment. Based on information assembled from ARI member companies air conditioner annual leak rates of 4% of the charge for 1996 equipment were used for the TEWI calculations. An end-of-life (E.O.L.) Charge loss rate of 15% was rationalized for residential air conditioning units on the basis of recovering 90% of the charge from 95% of the field units, but allowing for a 100% charge loss from about 5% of field stock (Hourahan 1996a). The 1996 annual leak or make up rate and 15% E.O.L. loss were also used for the European equipment calculations.

8.4 METHODOLOGY

8.4.1 Central Heating and Air Conditioning

Total equivalent warming impacts were calculated for baseline 10.5 kW (36,000 Btu/h) central air conditioners with SEERs of 10 and 12 in combination with an 80% or 92% efficient gas furnace. Seasonal energy use is computed based on a "typical" 1800 square foot residence with a 74.7×10^6 Btu/h (78.8×10^6 kJ/y) heating load and 16.1×10^6 Btu/y (17.0×10^6 kJ/y) cooling load in Pittsburgh, Pennsylvania, USA; a 34.8×10^6 Btu/h (36.7x106 kJ/y) heating load and 33.8×10^6 Btu/y (35.7×10^6 kJ/y) cooling load in Atlanta, Georgia, USA; and 0 Btu/y (0 kJ/y) heating load and 82.2×10^6 Btu/y (86.7×10^6 kJ/h) cooling load in Miami, Florida, USA (Ballou 1981).

For premium technology residential equipment, the baseline SEER of a central air conditioning unit was increased to 14. The gas powered GAX absorption heat pump that is being developed by DOE is shown as a 2005 option for Pittsburgh, Atlanta, and Europe.

Residential heating-only TEWI calculations were performed for Europe using averaged loads and seasonal performance factors (SPFs), Appendix D. An average heating load for all European countries was used. The European average of 0.470 kg CO_2/kWh was used to convert electric use to CO_2 emissions.

8.5 RESULTS AND DISCUSSION

Total equivalent warming impacts for various residential heating/cooling options incorporating natural gas were calculated for Pittsburgh, Pennsylvania; Atlanta, Georgia; and Miami, Florida in the United States, and the results are shown in Figs. 33 to 35. These results are computed using the efficiency data in Table 18. Each figure has two sections, the upper portion shows "standard technologies," or heating/cooling options that represent baseline costs for a residential system in each of these cities, while the lower indicates a "premium technologies" section which is a heating/cooling option that is significantly more expensive than the baseline technology as in Chapter 4 of this report.

Table 18. Residential gas options.

System	Efficiencies Cooling/Heating
Electric A/C and Gas Furnace	
minimum efficiency	SEER-10 / 80% furnace
high efficiency	SEER-12 / 92% furnace
Premium Technologies	
electric A/C and gas furnace	SEER-14 / 92% furnace
engine driven heat pump (HCFC-22)	gCOP-1.30 / gCOP-1.28
2005 Technologies	
GAX absorption heat pump	gCOP-1.0 / gCOP-1.5

Each segment of the bar graphs plotted in these figures indicates TEWI contributions from different sources. The initial, darker gray section of most of the bar graphs is the indirect contribution from electric power used for the vapor compression cooling process. The lighter gray section of bars shown in Figs. 34 to 36 with a lower density of fill dots, is the indirect contribution from gas combustion. The lightest section on some bars show the indirect contribution to TEWI from auxiliary electric parasitics such as pumps or fans that are not included in the SEER or HSPF ratings of the equipment. The moderate gray section on the ends of most of the bar graphs is the direct contribution to TEWI caused by refrigerant losses.

Figures 34 through 36 show the advantages of increased efficiency in both vapor-compression air conditioning and natural gas powered options.

In Pittsburgh and Atlanta which have substantial heating loads, newer gas-driven technologies like the engine-driven heat pump and the GAX absorption heat pump show a significant decrease in TEWI when compared to options using a conventional gas furnace. The TEWI advantage is also seen in the heating only data for residential European applications in Fig. 37.

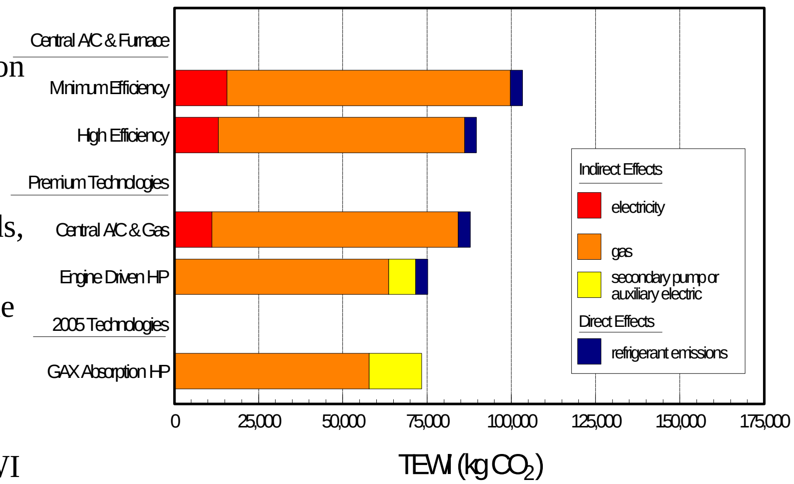

Figure 34. TEWI for residential gas heating/cooling options in Pittsburgh, Pennsylvania, USA.

In all the cases presented, the direct contribution of refrigerant to the TEWI was no larger than 6% of the total. Results presented for alternative refrigerants R-407C and R-410A in Chapter 4 of this report indicate they show efficiencies and GWPs similar to refrigerant HCFC-22 in the electric air conditioning unit and, therefore, a similar total TEWI.

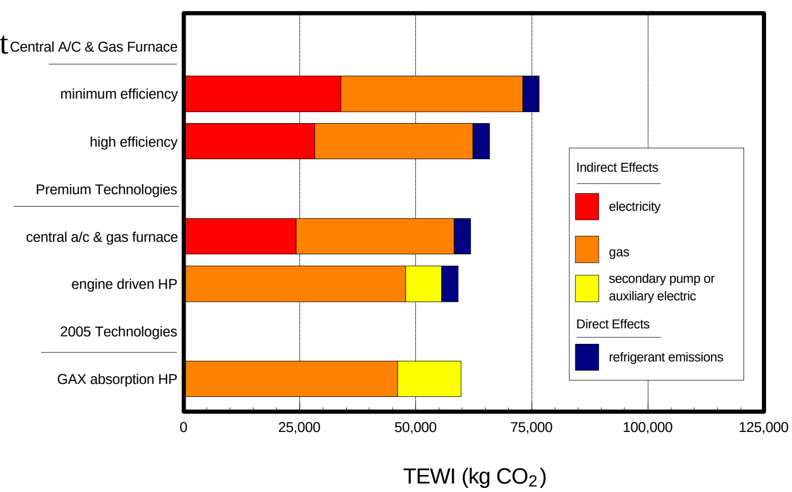

Figure 35. TEWI for residential gas heating/cooling options in Atlanta, Georgia USA.

As with heat pumps, nearly 80% of the direct TEWI effect for vapor compression air conditioning systems is due to the assumption on annual emissions from leakage, accidents, and maintenance practices. 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.

8.6 CONCLUSIONS

In climates with appreciable heating loads, the gas-fired heat pump options have a significantly smaller TEWI than combined gas and electric air conditioning options. This advantage decreases as the balance shifts to lower heating and higher cooling loads. The comparisons presented in Figs. 34 to 36 and Fig. 37 are based on laboratory measurements for a GAX absorption heat pump and ratings data for a commercially available engine-driven heat pump. First cost, climate, and projected operating costs rather than TEWI are likely to be the main criteria for selection of a residential heating/cooling system.

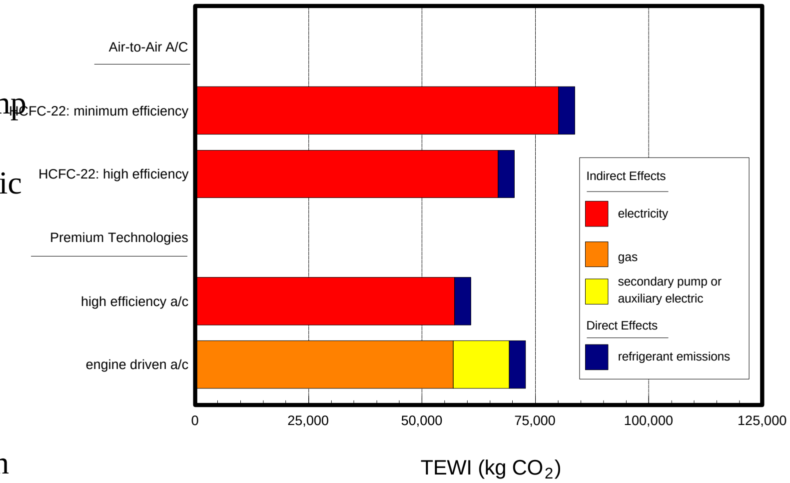

Figure 36. TEWI for residential cooling options in Miami, Florida USA.

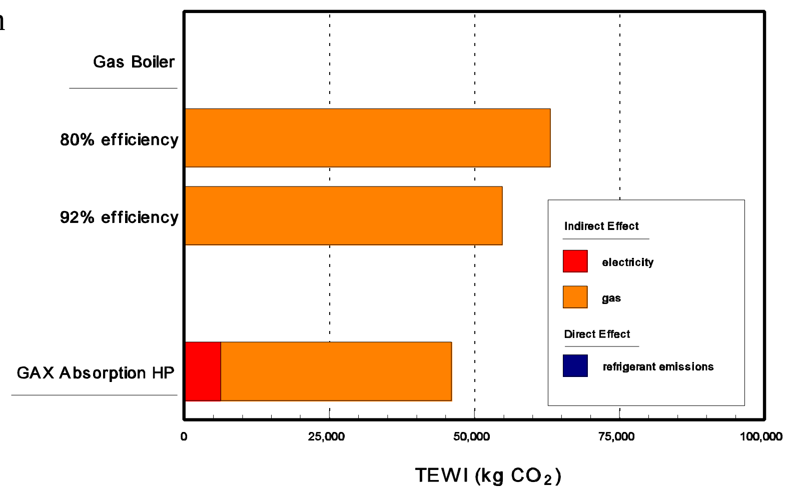

Figure 37. TEWI for residential heating only options in Europe.

9. CONCLUSIONS

The TEWI of the technology options considered in this report have been estimated for several end-use applications using 100 year ITH global warming potentials for each of the relevant greenhouse gases. This choice of time horizon primarily affects alternative systems based on fluorocarbon refrigerants and blowing agents and to a lesser extent those alternatives based on hydrocarbons. The direct emissions effect of refrigerant and blowing agents is more heavily emphasized by using the 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). Ultimately however, 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 ITH as was done in the 1991 AFEAS/DOE study. Scientists have also stated that while long term change reflects the cumulative effects, there is also a real concern that 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 the 100 year ITH. Therefore this study, like the 1994 project, presents the principal comparison results on the 100 year time horizon basis.

As in the previous studies, a consistent conclusion drawn from using either a 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 to ensure that their direct contribution to global warming does not outweigh the efficiency benefit derived from the use of fluorocarbons as refrigerants and blowing agents when considering at TEWI for most applications.

9.1 GENERAL CONCLUSIONS

Several broad conclusions can be drawn from the study.

- TEWI evaluations emphasize the combined environmental effect of the direct emission of greenhouse gases with the indirect effects of CO_2 emissions from energy use by equipment using these fluids as refrigerants or blowing agents. This is only one criterion in selecting between technology options. System costs, operating costs, regional energy costs, ease of maintenance, continuing technology improvements, etc., are equally important factors to consider in selecting *the most appropriate* technology for any specific application.

- Reductions in TEWI through the use of ammonia or hydrocarbons as refrigerants are insignificant for refrigeration systems with low emissions and may lead to an increase in energy use when applications of these fluids must meet the same safety design criteria currently defined as acceptable for fluorocarbon refrigerants.

- Ammonia and some hydrocarbon refrigerants have thermophysical properties comparable to (and for some applications superior to) those of HCFC or HFC refrigerants. They also have system irreversibilities and system design features necessary for safe products (e.g., secondary loops) which reduce their overall efficiency. Such changes often offset much of the TEWI benefit claimed for non-fluorocarbon refrigerants.
- Insignificant TEWI differences for most applications occur when design and service requirements, for low refrigerant emissions and safe operation of equipment using flammable or toxic refrigerants, are applied to systems engineered for non-flammable or non-toxic refrigerants.

! The direct contribution of HFC refrigerant and blowing agent emissions in refrigeration and insulation applications is given greater emphasis by using shorter-term GWP values in calculating the TEWI (the 100 year GWP values used herein are approximately three times the 500 year values used in the 1991 TEWI report).

! Actions to reduce fluorocarbon system refrigerant losses will result in lower TEWIs for supermarket refrigeration and automobile air-conditioning systems using these fluids.

! Non-fluorocarbon technologies may penetrate into mainstream refrigeration and air conditioning application areas, but it is unlikely that they will significantly displace conventional fluorocarbon technologies in the near future. Changing to HFC and non-fluorocarbon technologies both demand some technician and servicing personnel retraining, but procedures associated with HFC applications are more consistent with current practices associated with HCFC refrigerants and blowing agents.

! Efficiencies of conventional technologies are likely to increase as electric and gas-driven equipment and insulating foam formulations are further optimized for replacement refrigerants and blowing agents.

! Innovative design and modifications of standard practice can lead to significant reductions in TEWI for refrigeration systems using ammonia, fluorocarbon, or hydrocarbon refrigerants. These include mandatory refrigerant recovery and recycling, distributed refrigeration systems, charge reduction, elimination of flared fittings and reduced numbers of brazed connections, highly efficient purge units, improved heat transfer surfaces, high-efficiency compressors, etc.

! Average annual CO_2 emissions from electricity generation vary widely for individual regions and countries - from 0.0 to over 1.0 kg CO_2/kWh compared to the 1993 World average of 0.58. Emission rates also vary with season and time of day depending on how the generation fuel mix changes. Overall TEWI values in any particular location will be peculiar to the local electrical power generating efficiency and seasonal and time of day generating characteristics. The direct contribution can range from all (or nearly all) of total TEWI in areas with low CO_2 emission rates [using mostly nuclear or hydro power] to a minor fraction of TEWI for areas with high rates [using mostly coal].

9.2 Individual Applications

9.2.1 Household Refrigeration

The phase-out of HCFCs affects TEWI for household refrigerator/freezers through the choice of refrigerant and insulating material. No clear conclusions are possible at this time for foam blowing agents to replace HCFC-141b; the use of hydrocarbon and HFC blown foam insulations are under active development in an effort to improve their thermal and mechanical properties. Hydrocarbon blown foams continue to have higher thermal conductivity than HCFC-141b and HFC foams and, consequently, exhibit higher energy use with increased impact on CO_2 emissions. The increased energy use must be balanced against any direct impact caused by the HFC blowing agent itself.

TEWI estimates from this analysis for household refrigerator-freezers using HC refrigerants and foams are about 4-5% lower (in North America and Japan) and about 13% lower (in Europe) than those of HFC-134a refrigerant/HFC foam units assuming refrigerant recovery at end-of-life disposal. Energy use estimates for HC-based refrigerators are about 10% greater than that of the HFC units in all regions. Use of HCs in refrigerators raises safety concerns and has resulted in higher unit costs. Applying this cost differential to HFC designs (to incorporate vacuum panel insulation in the cabinet walls, for instance) could yield a product with potentially superior TEWI characteristics.

9.2.2 Automobile Air Conditioners

The direct effect of refrigerant emissions for HFC based automobile air conditioners is a significant part of the TEWI. The automobile manufacturers have responded aggressively with efforts to reduce charge size and emissions. Research and laboratory development of air-conditioning systems based on transcritical CO_2 compression show a potential to reduce TEWI for this application. Estimated TEWIs for CO_2 and hydrocarbon systems are lower than those for HFC-134a systems in regions with cool climates; TEWI are comparable to higher in climates with high cooling loads. Energy consumption estimates for HFC systems are consistently lower than those of CO_2 systems. The long term performance, lifetimes, viability, and TEWIs of both the alternative systems must be proven through extensive prototype and field trial testing. Energy consumption and TEWIs for HC-based systems are negatively affected in all regions due to use of an indirect loop with attendant efficiency penalties to keep the flammable refrigerant out of the passenger compartment.

9.2.3 Chillers

TEWI for this class of equipment has fallen significantly since the early 90's. New electric chillers have 25% to 30% lower TEWIs than models of 4-5 years ago due to replacement of CFC refrigerants with HCFC and HFC alternatives and to significant improvements in energy efficiency and reductions in refrigerant loss rates. The choice of refrigerant makes only a minor difference in direct TEWI in new equipment. Differences in chiller efficiencies for various refrigerant options can have a significant impact on the indirect contribution to TEWI, however, which is dominant in this application.

Significant advances have also been made in gas-fired chiller technologies. Triple-effect absorption chillers now under development show potential for 25-30% reductions in TEWI compared to existing double-effect machines. Engine-driven chillers are now available with rated efficiencies more than 25% higher than the value used in the TEWI-II report. Estimated TEWIs for these machines are around 25% lower than those for the triple-effect absorption equipment.

TEWI estimates given herein for gas-fired engine driven and absorption chillers are *not* directly comparable to those for electric driven chillers. It is difficult to make accurate interfuel comparisons using TEWI *as computed in this report*. At the very least “local” CO_2 emission factors need to be used for electric power generation instead of the broad regional averages used in this study, and some technologies may in fact require time of year or time of day factors to account for differences in emissions due to intermediate and peak power. These differences are in part due to the types of generating equipment brought on line for peak demand, changes in the local fuel mix, and lower transmission and distribution efficiencies during peak generating periods.

9.2.4 Unitary Equipment

Transition away from HCFC-22 in this equipment appears to be achievable with either no change or a slight reduction in the estimated TEWIs. The HFC-400 blends R-407C and R-410A are the principal alternatives being considered as HCFC-22 substitutes. Laboratory and limited field testing indicates that R-407C has equivalent performance compared to R-22 while R-410A-based equipment has potential for slightly better efficiency, lowering the indirect contribution to TEWI. Geothermal heat pumps and premium grade air-to-air heat pumps can significantly reduce TEWI for this application, albeit with higher purchase costs.

Gas engine-driven and gas-fired absorption heat pumps for space heating and cooling show potential to reduce TEWI in climates dominated by heating requirements. Long term performance and reliability of the gas-driven technologies have not been demonstrated.

9.2.5 Commercial Refrigeration

Supermarket refrigeration systems have had high direct contributions because of historically high refrigerant charges and leakage rates. Equipment manufacturers have worked to reduce refrigerant leaks at the display cases and in the brazed and welded joints in refrigerant lines. The lower emission rates have reduced TEWI significantly from the values reported previously. New system design concepts (secondary loop and distributed compressor approaches) also dramatically reduce the direct effect of refrigerant emissions and result in lower overall TEWI estimates for this application. The differences in TEWI between the HFC mixtures that have been considered are due primarily to the

GWPs of the refrigerants; the differences in energy use are not considered significant. Ammonia with secondary heat transfer loops has been shown to be a viable alternative for HFCs in this application, but there can be an energy penalty associated with necessary secondary heat exchangers. In many areas system designs will have to comply with regulation and permit requirements intended to ensure safe use in retail and commercial areas. Some European and developing countries have fewer regulations and are more open to using ammonia. Refrigerant containment measures necessary for ammonia and hydrocarbons could also be used with HFCs, resulting in essentially identical TEWI for these alternatives.

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APPENDIX A: ELECTRIC POWER CO_2 EMISSION RATE DATA

A.1 REGIONAL AVERAGES AND RANGES

In the first report (Fischer, et al 1991) CO_2 emission rates were calculated based on data for the carbon content of fossil fuels used for electricity generation (coal, oil, and natural gas) and the average mix of generation capacity for a given region (Japan, Europe, or North America). For the present study it was decided to use publicly available data on CO_2 emissions from each major region wherever possible. Accordingly much effort was devoted to soliciting data from utilities, utility organizations, and other sources. Considerable data were received, primarily from European sources, and are summarized in Tables 19 and 20. The data sources for these tables are listed as follows:

1. Jean-Yves Caneill, Electricité de France, 1996; data give "order of magnitude" CO_2 emissions for current plants of 0.99 kg CO_2/kWh for coal, 0.87 for oil, and 0.67 for gas; also gives projections for future power plants (2010) of 0.73 for coal, 0.76 for oil, and 0.35 for gas. Caneill also included a paper by W. Ruijgrok (KEMA), presented at a 1993 UNIPEDE/IEA conference in Hamburg. This paper gives 1989 data on average emissions for EU-12 and America of 0.39 and 0.54 kg CO_2/kWh, respectively.
2. S. Fischer, et al. 1994. *Energy and Global Warming Impacts of Not-In-Kind and Next Generation CFC and HCFC Alternatives*, AFEAS and DOE.
3. Bert Stuij, IEA/Paris, 1996; gives average emissions for European OECD countries of 0.50 (range of 0.00 to 1.23 kg CO_2/kWh), a value of 0.71 for the U.S., and an OECD average of 0.58, all based on 1990 data.
4. P. Göricke, RWE Energie, Germany, 1996; provided a paper "Fuel Cycle Analysis" by H. J. Laue giving 1990 average value for EU-12 of 0.50 (range of 0.08 to 0.96); also provided a 1995 value for Germany of 0.61 from VDEW (German association of electric power generation and distribution companies).
5. K. Hara, Japan Industrial Conference for Ozone Layer Protection (JICOP), 1996; provided an average value of 0.473 kg CO_2/kWh for 1995 based on consolidated input from Japanese electricity producers.
6. Jurgen Michorius, Dutch Electricity Generating Board, 1996; gives 1992 average emission data for EU-12 of 0.47 kg CO_2/kWh with country averages ranging from 0.00 to 1.08; major data source (for all countries except Germany) was *Annual Energy Review, Special Issue-June 1994, Directorate General for Energy (DGXVII)*; data for Germany include emissions from plants in the former East Germany.
7. Energy Information Administration (EIA), U. S. Department of Energy, December, 1996a; yields 1994 average rate for electric utilities of 0.65 (0.64 for 1995) which includes 6% transmission and distribution losses. Including nonutility generation, the annual average is about 8% higher.

CO_2 emissions numbers given in the "Best" Value column in Tables 19 and 20 were generally taken from the reference providing the most recent data. In most cases for Europe this was taken from Michorius' input. The emissions values for coal, oil, and gas plants (1.11, 0.77, and 0.55, respectively)

Table 19. Carbon dioxide emissions from power plants.

Emissions Source	CO_2 Emission Rate (kg / kWh)							
Reference	1	2	3	4	5	7	Range	"Best" Value
Coal Plant	0.993	1.25	0.90 - 1.33	0.89 - 1.11			0.89 - 1.33	1.11
Oil Plant	0.866	0.96	0.65 - 0.89	0.70 - 0.88			0.65 - 0.96	0.77
Gas Plant	0.665	0.583	0.42 - 0.61	0.46 - 0.58			0.42 - 0.67	0.55
future coal plant	0.73 - 0.80						0.73 - 0.80	
future oil plant	0.76						0.76	
future gas plant	0.35 - 0.40						0.35 - 0.40	
Australia	0.82						0.82	0.82
Canada	0.22		0.26				0.22 - 0.26	0.24
Japan	0.39	0.582	0.46		0.473		0.39 - 0.58	0.473
New Zealand	0.13						0.13	0.13
U.S.	0.54	0.650	0.71			0.64-0.65	0.54 - 0.71	0.65

are averages of the data provided by Caneill, Stuij, and Göricke. Hara's input was used for the Japanese average. In comparison, an average value of 0.45 was calculated for Japan for 1993 using International Energy Agency (IEA) data (OECD/IEA, 1995b) and the average plant emissions factors above. The average for North America was taken as 0.65 kg CO_2/kWh. This is equal to the 1994 EIA average and to that used in the 1994 AFEAS/DOE report (Fischer). It is approximately equal to the average of the values provided by Caneill and Stuij and also agrees well with the 0.674 value determined by Calm (1993). Calm also determined average values for six regions with the U.S., shown in Table 21.

The 1995 U. S. CO_2 emissions rate (0.64) could have been used for the North American average. However, this data only became available during the final stage of this report's preparation and would have the effect of decreasing the indirect TEWI contributions for all electric technologies considered in North America by only about 1.5%. A change of this magnitude would not cause a significant change in relative total TEWI between technologies nor to any of the conclusions drawn from the analyses and comparisons. For this reason it was decided to stay with the 0.65 value for this report.

Table 20. Carbon dioxide emissions from power plants in Europe.

Emissions Source	CO_2 Emission Rate (kg / kWh)							
Reference	1	2	3	4	5	6	Range	"Best" Value
Austria	0.22	0.00	0.37			0.220	0.00 - 0.37	0.22
Belgium	0.30	0.47	0.37	0.35		0.288	0.29 - 0.47	0.29
Denmark	1.08	1.01	0.80	0.96		0.844	0.80 - 1.08	0.84
Finland	0.30	0.58	0.36			0.235	0.24 - 0.58	0.24
France	0.11	0.27	0.13	0.08		0.093	0.08 - 0.27	0.09
Germany	0.54	0.70	0.67	0.61		0.597	0.54 - 0.70	0.61
Greece	0.92	0.77	1.23	0.86		0.976	0.77 - 1.23	0.98
Iceland		0.15	0.00				0.00 - 0.15	0.00
Ireland	0.78	0.80	0.91	0.80		0.700	0.70 - 0.91	0.70
Italy	0.55	0.59	0.60	0.63		0.586	0.55 - 0.63	0.59
Luxembourg			0.72	0.29		1.083	0.29 - 1.08	1.08
Netherlands	0.58		0.67	0.66		0.637	0.58 - 0.67	0.64
Norway	0.005	0.01	0.00				0.00 - 0.005	0.00
Portugal	0.59		0.67	0.51		0.638	0.51 - 0.67	0.64
Spain	0.43		0.53	0.40		0.483	0.40 - 0.53	0.48
Sweden	0.04	0.23	0.04			0.044	0.04 - 0.23	0.04
Switzerland	0.04	0.05	0.13				0.04 - 0.13	0.08
Turkey	0.55		0.70				0.55 - 0.70	0.62
U.K.	0.67	0.89	0.84	0.75		0.640	0.64 - 0.89	0.64
European Average		0.513		0.50		0.427-0.469	0.43 - 0.51	0.47
OECD Europe			0.50					
OECD			0.58					

However, historical data from this source indicate that carbon emissions from fossil-fueled power plants in North America are falling gradually. As older, less efficient plants are retired this trend should continue in all regions.

The range of values for the North American (from EIA, 1996a) and European annual averages suggest that an uncertainty band of ±3-4% is appropriate for the regional estimates used in the bulk of the comparisons in this study.

Representative values for other regions for 1993 are given in Table 22. These data were determined by using the average plant emissions factors from Table 18 and average generation mixes based on data compiled by IEA (OECD/IEA 1995a, 1995b). In general, these averages and ranges are of the same order of magnitude as those for Europe, Japan, and the U.S.

Table 21. U.S. regional average CO_2 emissions from power plants (Calm 1993).

Region	CO_2 Emissions (kg/kWh)
East Central	0.939
South Central	0.737
West Central	0.672
Southeast	0.671
West	0.497
Northeast	0.489

Table 22. 1993 average electric power plant CO_2 emissions for other regions.

Region	CO_2 Emissions (kg/kWh)	Region	CO_2 Emissions (kg/kWh)
Africa Average	0.77	Former USSR Average	0.44
Egypt	0.53	Russia	0.47
South Africa	1.03	Tajikistan	0.02
Zaire	0.02	Ukraine	0.48
Asia Average (w/o China)	0.66	Non-OECD Europe	0.79
Australia	0.82	Bulgaria	0.59
China	0.88	Hungary	0.51
India	0.83	Poland	1.07
New Zealand	0.13	Romania	0.64
Singapore	0.77		
Latin America Average	0.14	Middle East Average	0.63
Argentina	0.24	Israel	0.83
Brazil	0.04	Saudi Arabia	0.68
Columbia	0.22	Syria	0.32
World Average	0.58		

A.2 SENSITIVITY OF TEWI TO CO_2 EMISSION RANGES

As the preceding section indicates, individual country average emission rates vary widely around the regional averages. Figure 38 illustrates the range of average country emission rates for several western European nations and compares those with relative percentages of electricity produced by each country. Indirect (and overall) TEWI values depend directly on the assumed CO_2 emission rate and conclusions drawn can vary. As an example, Fig. 39 shows TEWIs for two representative European refrigerators, one installed in a country with no fossil-fuel electricity dependence (0.0 kg CO_2/kWh) and the other in a country heavily dependent on coal-fired generation (0.98 kg CO_2/kWh). In the latter case, improvements in electricity generation efficiency and/or a switch to different fuel sources would be much more effective for TEWI reduction than a change in refrigerant or blowing agent. The direct TEWI does not change in either case and is about an order of magnitude less than the indirect TEWI for the coal-generation dominated situation.

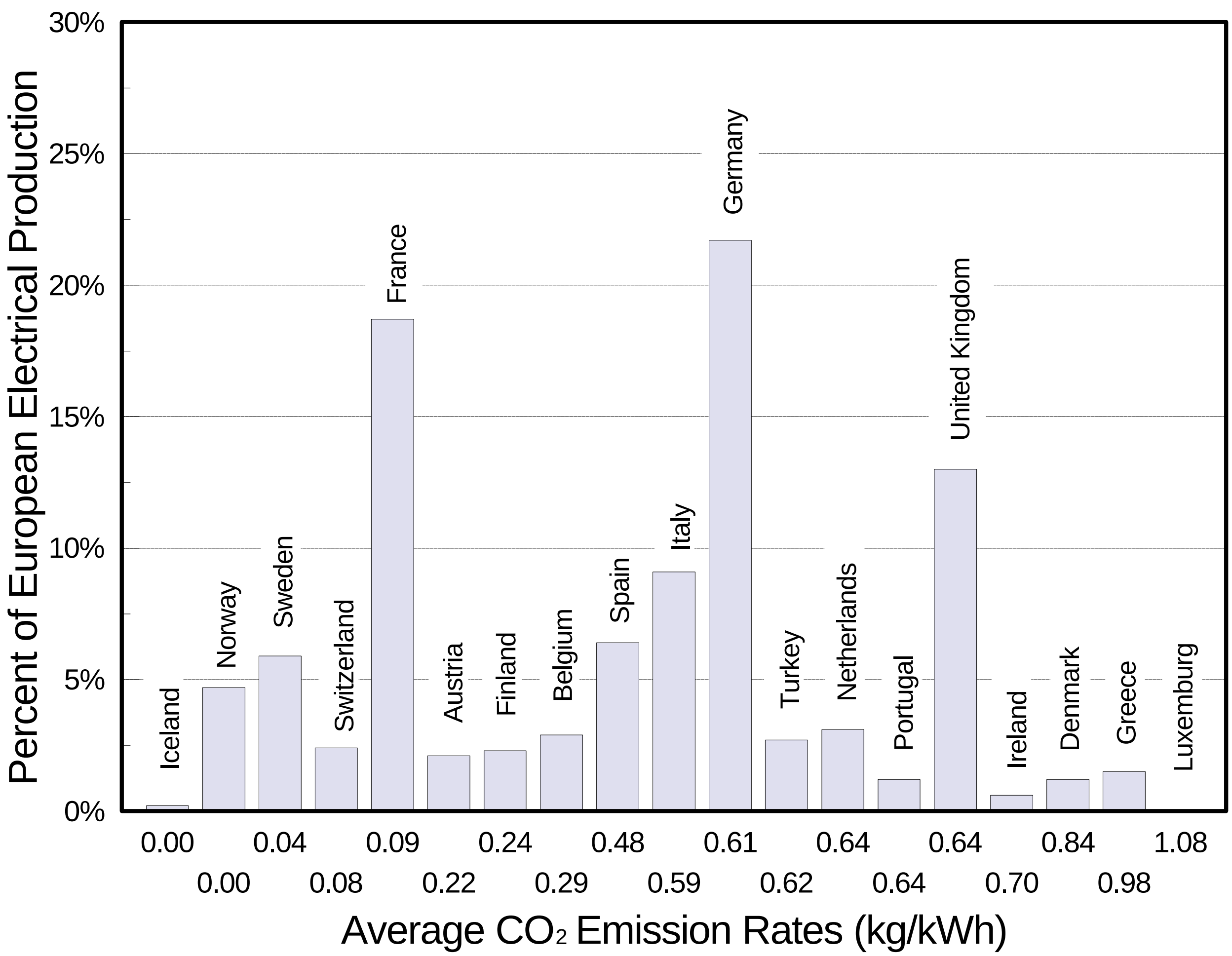

Figure 38. Average power plant emission rates and electricity production for European nations.

A.3 IMPACT OF FUEL PROCESSING AND TRANSPORTATION ENERGY CONSUMPTION ON CO_2 EMISSIONS

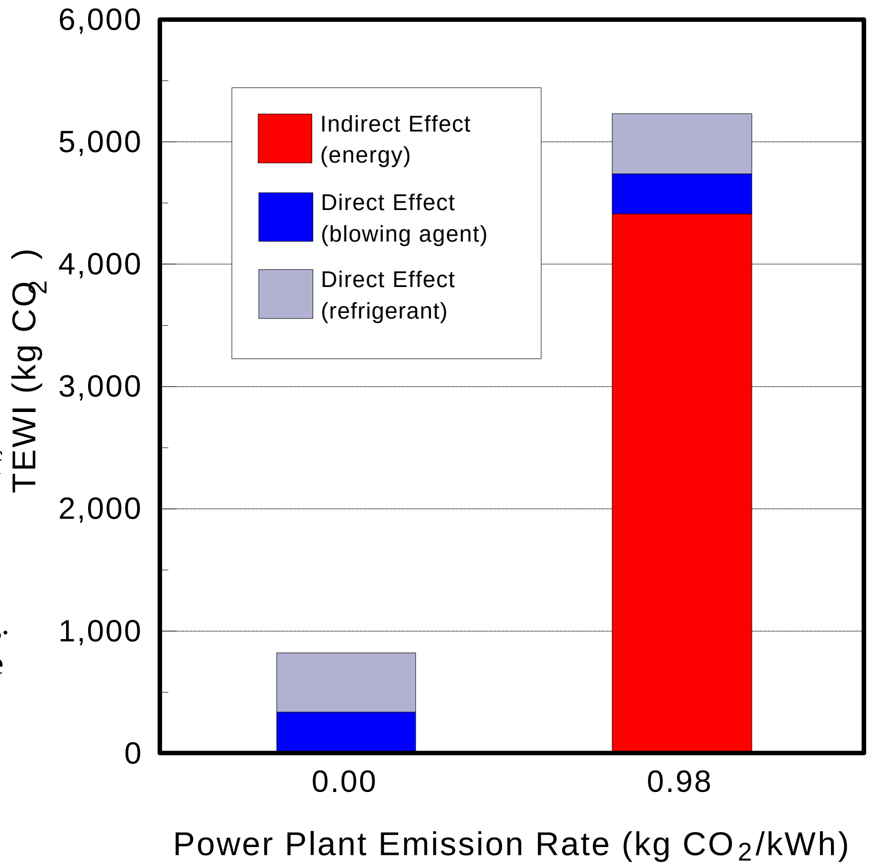

Figure 39. TEWI for European refrigerator/freezer using high (0.98 kg CO_2/kWh) and low (0.0 kg CO_2/kWh) emission rates for power plants (HFC-134a, HCFC-141b, 15 year lifetime, 200 kWh/y).

The previous two studies did not consider this effect in making the indirect TEWI estimates because it was felt that the magnitude of the change to CO_2 emission rates would be small and would, thus, not impact the comparisons we were making. However, if an absolute value for the TEWIs of competing alternatives was desired for a particular application, this impact would of course need to be quantified. Data from a paper by Beggs (1996) was used to attempt to estimate the impact of fuel processing and transportation energy on the average CO_2 emissions from electricity generation for the United Kingdom (UK). Using the data presented by Beggs it is possible to come up with the following estimates for increases in plant emission rates:

coal plants:	add 4.5% to the plant rate;
gas plants:	add 8.9% to the plant rate;
oil plants:	add 10.9% to the plant rate; and
nuclear plants:	use 0.005 kg CO_2/kWh.

Using these factors and applying them to the 1993 UK electricity supply mix from IEA (1995b) yields an overall increase in the average CO_2 emission rate of about 5.5%. This would raise the average value for the UK in Table 20 from 0.64 to 0.675 kg CO_2/kWh. The absolute impact would be smaller for countries with a heavier dependence on nuclear or hydro power and would be larger where use of coal, oil, or gas is greater. Adding this effect to indirect contributions from electric power would slightly increase TEWIs for all options which use electric power. However, the relative differences in TEWI between any two options would not change significantly. Therefore, omitting this effect from the TEWI estimates does not affect the overall conclusions significantly.

A.4 IMPACT OF AVERAGE, SEASONAL, AND TIME-OF-DAY VARIATIONS IN CO_2 EMISSION RATES

Electricity consumption for certain applications considered in this study tends to occur more heavily at specific times of the year and during certain periods of each day. Residential and commercial building air-conditioning and heating are the prime examples. Clearly, CO_2 emission rates from electric power production are not constant. They vary with region, time-of-day, and season as the generation mix changes. In the UK for example, Beggs (1996) cites a 30% to 47% reduction in CO_2 emissions in summer nighttime hours relative to afternoon hours and an 11% to 22% reduction in winter for the UK. He also shows that electric generation needs and CO_2 emissions are highest in winter and, further, his time-of-day average emissions estimates range from 0.23 to 0.63 kg CO_2/kWh which are lower than the 1992 overall average in Table 20 above (0.64). For the U.S., data for 1996 electricity generation indicated that the utility generation mix remained fairly constant with natural gas generation peaking in the spring and summer, coal in fall and winter, hydro in winter and spring, and oil and nuclear staying relatively flat, Table 23 (EIA 1997). Historical data for 1994/1995 from EIA (1996b) mirror these trends.

Table 23. U.S. electricity generation mix (1996).

Fuel Source	Quarter				Year
	1st	2nd	3rd	4th	
Coal	56.1%	54.2%	55.7%	59.6%	56.4%
Oil	2.9%	1.7%	2.2%	1.9%	2.2%
Natural Gas	5.8%	9.6%	11.6%	6.8%	8.5%
Nuclear	22.9%	21.9%	21.3%	21.6%	21.9%
Hydroelectric and Other	12.3%	12.6%	9.1%	10.1%	11.0%

Source: EIA 1997.

These variations notwithstanding, a yearly average CO_2 emission value was sufficient for the primary purposes of the present analysis -- a relative comparison between the overall global warming impacts of HCFC refrigerants and blowing agents and those of alternative fluids and technologies. As noted in the Executive Summary, absolute TEWI comparisons, particularly between options that use different fuels, are difficult. If a detailed and rigorous evaluation of TEWI for competing air-conditioning or refrigeration technology options is required for a particular location, then an hourly simulation of the systems coupled with an hourly schedule for the local electricity source generation mix and CO_2 emissions would be needed. Conclusions from such an analysis would be applicable only to the specific location considered. In order to meet one of the major goals of the project, to identify opportunities for CO_2 emissions reduction that are broadly applicable across regions, these detailed analyses would have to be done for several locations in each region and for each technology option considered to fairly illustrate how the comparisons would vary throughout the regions. Such analyses are well beyond the scope of and time available for this project. A less rigorous case study was conducted to illustrate the sensitivity of TEWI comparisons between electric and absorption chiller options to different assumptions about electric power CO_2 emissions.

Figure 40 illustrates TEWI values for 475 ton (1670 kW) electric powered centrifugal and direct-fired absorption chillers in three U.S. utility districts selected for the broad differences in their overall summertime average CO_2 emission rates and summertime average emission rates for just peak and intermediate power generation (Reid 1997). “Utility A” was chosen because it has a low average emission rate because of a high nuclear capacity for base power generation and a high intermediate and peak rate because of a high use of coal for intermediate power generation. “Utility C” was selected because it has a high summer average CO_2 emission rate (large coal base power) and lower intermediate and peak emission rate (predominantly natural gas). “Utility B” in Fig. 40 has summertime average and intermediate and peak power emission rates that are approximately the same as the regional average used in the body of this report for North America. A 475 ton (1670 kW) chiller was used for these calculations because it is the weighted average nominal size for the U.S. (Hourahan 1996a). Operating hours were adjusted to coincide with the utility locations.

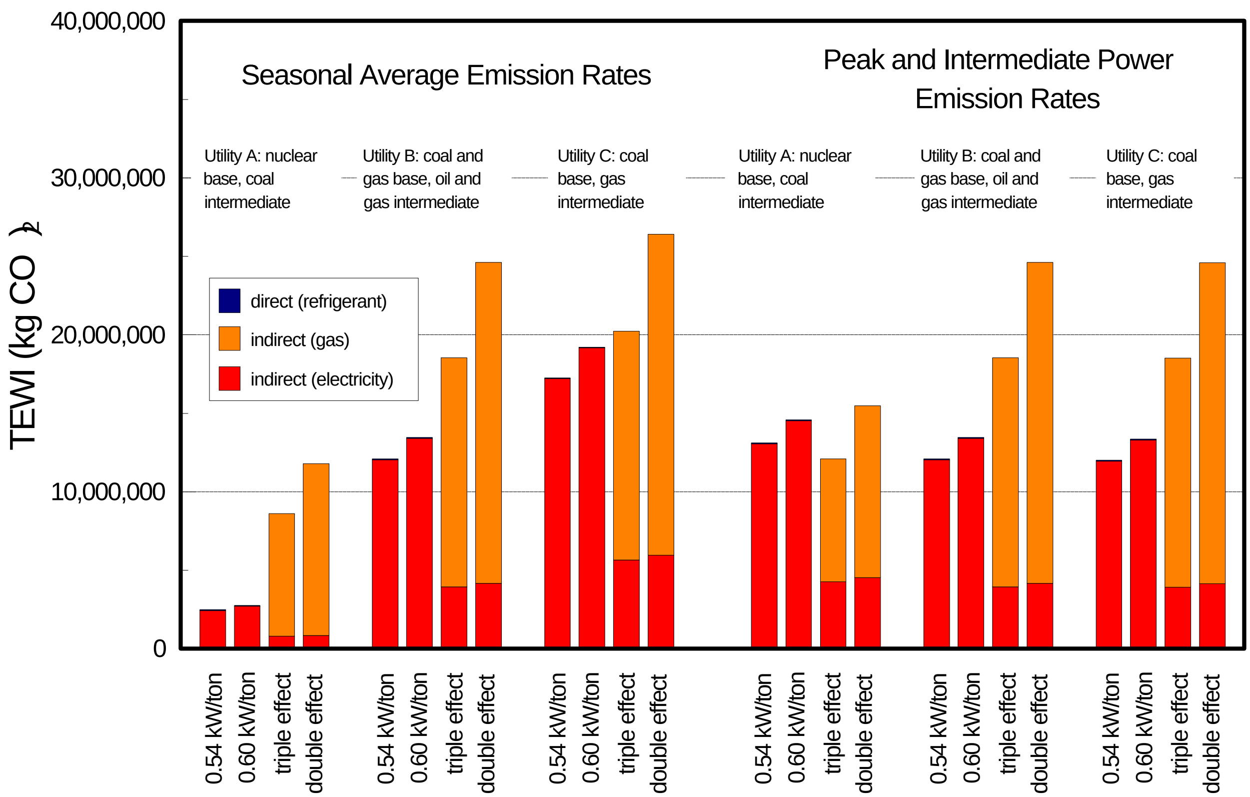

Figure 40. Sensitivity of TEWI for chillers to power plant CO_2 emission rates.

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APPENDIX B: ATMOSPHERIC LIFETIMES AND GWPs FOR REFRIGERANTS AND BLOWING AGENTS

Notes for Tables 23 and 24:

a. CFC and HCFC lifetimes and 100 year GWP values are taken from the 1995 Radiative Forcing of Climate Change Report to IPCC from the Scientific Assessment Working Group (WB1), Table 5.2 CFCs are not listed in the 1995 IPCC report.

b. When listed, the lifetimes and 100 year GWP values for HFC refrigerants and blowing agents and methane are taken from the 1995 Radiative Forcing of Climate Change Report to IPCC from the Scientific Assessment Working Group, Table 2.9.

c. The atmospheric lifetimes and 100 year GWPs given in Tables B.1 and B.2 for HFCs and HCs *not* specifically listed in the 1994 or 1995 reports are from the open literature or estimates provided by AFEAS member companies. This includes values for the zeotropic and azeotropic mixtures.

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Table 24. Atmospheric lifetimes and 100 year GWPs for refrigerants and blowing agents.

Refrigerant	Atmospheric Lifetime (years)	ODP	GPW (100 year ITH)		Refrigerant	Atmospheric Lifetime (years)	ODP	GWP (100 year ITH)
CFCs					**HFCs (cont)**			
CFC-11	50±5	1.0	4000		HFC-152a	1.5	0	140
CFC-12	102	1.0	8500		HFC-227ea	36.5	0	2900
CFC-113	85	0.8	5000		HFC-236ea	8	0	710
CFC-114	300	1.0	9300		HFC-236fa	209	0	6300
CFC-115	1700	0.6	9300		HFC-245ca	6.6	0	560
HCFCs					HFC-245ea [c]	5	0	320
HCFC-22[a]	13.3	0.055	1700		HFC-245eb	3.9	0	380
HCFC-123[a]	1.4	0.02	93		HFC-245fa	7.3	0	820
HCFC-124[a]	5.9	0.022	480		HFC-356mcf	1.3	0	125
HCFC-141b[a]	9.4	0.11	630		HFC-356mffm	7.0	0	760
HCFC-142b[a]	19.5	0.065	2000		HFC-365mfc	10.8	0	840
HCFC-225ca[a]	2.5	0.025	170		HFC-43-10mee	17.1	0	1300
HFCs					**HCs**			
HFC-23[b]	264	0	11,700		HC-50 (methane)	12.2±3	0	21
HFC-32[b]	5.6	0	650		HC-170 (ethane)	--	0	11
HFC-125[b]	32.6	0	2800		HC-290 (propane)	--	0	11
HFC-134a[b]	14.6	0	1300		HC-3(11)0 (butane)	--	0	11
HFC-143a[b]	48.3	0	3800		Cyclo-Pentane	--	0	11

Table 25.. Atmospheric lifetimes and 100 year GWPs for zeotropic and azeotropic mixtures.

Zeotropes and Azeotropes	Atmospheric Lifetime (years)	ODP	GWP (100 year ITH)
R-401A[c]	----	0.037	970
R-401B[c]	----	0.04	1060
R-401C[c]	----	0.03	760
R-402A[c]	----	0.021	2250
R-402B[c]	----	0.033	1960
R-403B[c]		0.031	3570
R-404A[c]	----	0	3260
R-405A[c]	----	0.028	4480
R-406A[c]	----	0.057	1560
R-407A[c]	----	0	1770
R-407B[c]	----	0	2290
R-407C[c]	----	0	1530
R-408A[c]	----	0.026	2650
R-409A[c]	----	0.048	1290
R-410A[c]	----	0	1730
R-411A[c]	----	0.048	1330
R-411B[c]	----	0.052	1410
R-500	----	0.74	6010
R-501	----	0.29	3150
R-502	----	0.33	5260
R-503	----	0.6	11,350
R-504	----	0.31	4890
R-507[c]	----	0	3300

APPENDIX C: SPREADSHEETS FOR REFRIGERATOR/FREEZER TEWI CALCULATIONS

Table 26.. TEWI calculations for a North American refrigerator/freezer.

Baseline Information		
Internal Volume	510	liters
Foam Volume	0.267	m^3
Refrigerant Charge	0.155	kg HFC-134a
HCFC-141b Blown Foam	12.0%	blowing agent
Annual Energy Use	646.5	kWh/year
CO_2 Emission Rate	0.65	kg CO_2/kWhe

	Refrigerant: HFC-134a											Refrigerant: HC-290/HC-600a	
Blowing Agent	HCFC-141b	Vacuum Panels	HCFC-22/ HCFC-142b	HFC-134a	HFC-365mfc	HFC-356mffm	HFC-245fa	HCFC-22/ HCFC-141b	HFC-236ea	Pentane Isomers	Carbon Dioxide	Cyclopentane	Carbon Dioxide
Foam													
Volume (m^3)	0.267	0.235	0.267	0.267	0.267	0.267	0.267	0.267	0.267	0.267	0.267	0.267	0.267
Density (kg/m^3)	32.60	32.60	33.99	37.80	32.90	37.10	33.00	32.83	31.40	36.90	33.64	36.90	33.60
Mass (kg)	9.02	7.94	9.08	7.85	7.48	8.98	8.96	8.77	8.96	8.96	8.98	7.16	8.97
Blowing Agent													
Weight %	11.84%	11.84%	11.00%	8.28%	10.50%	10.68%	12.60%	10.80%	13.08%	5.07%	2.00%	5.07%	2.00%
Mass (kg)	1.031	0.907	0.998	0.848	0.922	1.058	1.110	0.947	1.097	0.500	0.180	0.500	0.179
GWP	630	630	1700	1300	840	760	820	1165	710	3	0	3	1
Refrigerant													
Mass (kg)	0.155	0.155	0.155	0.155	0.155	0.155	0.155	0.155	0.155	0.155	0.155	0.05	0.05
GWP	1300	1300	1300	1300	1300	1300	1300	1300	1300	1300	1300	3	3
Direct Effect	884	802	1899	1304	861	1020	1127	1304	1003	203	202	2	0
Refrigerant	202	202	202	202	202	202	202	202	202	202	202	0	0
Blowing agent	649	571	1697	1086	775	804	910	1102	779	1	0	1.4	0.2
Energy Use													
Daily (kWh)	1.77	1.672	1.86	2.01	1.93	1.84	1.79	1.77	1.85	1.95	1.98	1.95	1.98
Annual (kWh)	646.5	591.7	679.4	734.2	704.9	672.1	653.8	646.5	675.1	712.2	723.2	712.2	723.2
Conversion Factors													
Lifetime (years)	15	15	15	15	15	15	15	15	15	15	15	15	15
CO_2 factor (kg/kWh)	0.65	0.65	0.65	0.65	0.65	0.65	0.65	0.65	0.65	0.65	0.65	0.65	0.65
Indirect Effect	6303	5769	6624	7158	6873	6553	6375	6303	6588	6944	7051	6944	7051
TEWI	7154	65.42	8446	8446	7849	7558	7415	7608	7568	7147	7253	6946	7051
% Direct Effect	11.89%	11.82%	22.28%	15.41%	12.44%	13.30%	14.99%	17.15%	12.95%	2.84%	2.78%	0.02%	0.00%

Table 27. TEWI calculations for a European refrigerator/freezer.

Baseline Information		
Internal Volume	230	liters
Foam Volume	0.1539	m^3
Refrigerant Charge	0.045	kg HC-600a
HCFC-141b Blown Foam	5.4%	blowing agent
Annual Energy Use	300	kWh/year
CO_2 Emission Rate	0.470	kg CO_2/kWhe

	Refrigerant: HC-600a (isobutane)				Refrigerant: HFC-134a			
Blowing Agent	Cyclopentane	HCFC-141b	HFC-245fa	HFC-134a	Cyclopentane	HCFC-141b	HFC-245fa	HFC-134a
Foam								
Volume (m^3)	0.1539	0.1539	0.1539	0.1539	0.1539	0.1539	0.1539	0.1539
Density (kg/m^3)	37.5	33.80	33.54	37.8	37.5	33.80	33.54	37.8
Mass (kg)	5.163	5.202	5.162	4.525	5.163	5.202	5.162	4.525
Blowing Agent								
Weight %	5.1%	10.0%	11.5%	8.3%	5.1%	10.0%	11.5%	8.3%
Mass (kg)	0.279	0.520	0.594	0.366	0.279	0.520	0.594	0.366
GWP	11	630	820	1300	11	630	820	1300
Refrigerant								
Mass (kg)	0.045	0.045	0.045	0.045	0.127	0.127	0.127	0.127
GWP	11	11	11	11	1300	1300	1300	1300
Direct Effect	3.6	328.2	487.3	476/9	168.2	492.8	651.9	641.5
Refrigerant	0.50	0.50	0.50	0.50	165.10	165.10	165.10	165.10
Blowing agent	3.22	327.7	486.8	627.7	3.22	327.7	486.8	627.7
Energy Use								
Daily (kWh)	0.90	0.82	0.82	0.85	0.90	0.82	0.82	0.85
Annual (kWh)	328	300	300	312	328	300	300	312
Conversion Factors								
Lifetime (years)	15	15	15	15	15	15	15	15
CO_2 Emissions (kg/kWh)	0.470	0.470	0.470	0.470	0.470	0.470	0.470	0.470
Indirect Effect	2312.4	2115	2115	2200	2312.4	2115	2115	2200
TEWI	2316	2443	2602	2828	2481	2608	2767	2992
% Direct Effect	0.15%	15.52%	23.04%	28.56%	7.29%	23.30%	30.82%	36.04%

Table 28. TEWI calculations for a Japanese refrigerator/freezer.

Baseline Information		
Internal Volume	355	liters
Foam Volume	0.1794	m^3
Refrigerant Charge	0.140	kg HFC-134a
HCFC-141b Blown Foam	11.0%	blowing agent
Annual Energy Use	522	kWh/year
CO_2 Emission Rate	0.473	kg CO_2/kWhe

	Refrigerant: HFC-134a								Refrigerant: HC-600a
Blowing Agent	HCFC-141b	HCFC-22/ HCFC-142b	HFC-152a	HFC-236ea	HFC-245ca	HFC-245fa	HFC-356mffm	Pentane Isomers	Pentane Blowing Agents
Foam									
Volume (m^3)	0.1794	0.1794	0.1794	0.1794	0.1794	0.1794	0.1794	0.1794	0.1794
Density (kg/m^3)	36	34	29.4	33.54	33.54	32	33.63	37	37
Mass (kg)	6.46	6.10	5.27	6.02	6.02	5.74	6.03	6.64	6.64
Blowing Agent									
Weight %	11.00%	11.00%	8.10%	11.90%	11.90%	12.60%	11.30%	5.45%	5.45%
Mass (kg)	0.710	0.671	0.427	0.716	0.716	0.723	0.682	0.362	0.362
GWP	630	1820	140	710	560	820	760	3	3
Refrigerant									
Mass (kg)	0.140	0.140	0.140	0.140	0.140	0.140	0.140	0.140	0.080
GWP	1300	1300	1300	1300	1300	1300	1300	1300	3
Direct Effect	630	1403	242	690	583	775	700	183	1
Refrigerant	182	182	182	182	182	182	182	1	0
Blowing agent	448	1221	60	508	401	593	518	182	1
Energy Use									
Daily (kWh)	1.43	1.43	1.67	1.49	1.50	1.43	1.48	1.57	1.57
Annual (kWh)	522	522	611	543	548	522	540	572	572
Conversion Factors									
Lifetime (years)	15	15	15	15	15	15	15	15	15
CO_2 Factor (kg/kWh)	0.473	0.473	0.473	0.473	0.473	0.473	0.473	0.473	0.473
Indirect Effect	3704	3704	4335	3853	3888	3704	3831	4058	4058
TEWI	4333	5107	4577	4543	4471	4479	4531	4241	4060
% Direct Effect	14.5%	27.5%	5.3%	15.2%	13.0%	17.3%	15.5%	4.3%	0.0%

APPENDIX D: AVERAGED HEATING AND COOLING LOADS AND HEAT PUMP PERFORMANCE FACTORS FOR NEW SINGLE-FAMILY HOMES AND COMMERCIAL BUILDINGS IN EUROPE

Table 29. European residential building data.

Housing Unit	1991 new stock (units)	average load (kWh/y)
Single-family homes -- heating		
France	170,200	14,700
Germany	122,100	24,000
Greece	42,800	6,300
Italy	182,000	16,000
Netherlands	65,700	8,700
Norway	7,000	19,000
Sweden	28,800	19,000
United Kingdom	210,100	17,500
Total	828,800	(avg) 16,400
Single-family homes -- cooling		
Southern Europe (Greece)	42,800	7,500
Northern Europe (Germany)	122,100	100

Source: *International Heat Pump Status and Policy Review*, Report HPC-AR3, IEA Heat Pump Centre, September 1994 (Part 1, Tables 2.5 and 2.6; supplemented with information from Part 2A, National Position Papers -- Italy and The United Kingdom).

Table 30. Average seasonal performance factors for HCFC-22 based heat pumps in Europe.

Country	Electric		Absorption	
	Air-Source	Geothermal	Air-Source	Geothermal
<u>Residential Building Heat Pumps</u>				
France	2.0	2.5 - 3.0		
Germany	2.1 - 2.3	2.2 - 3.0	1.1 - 1.3	1.1 - 1.3
Greece	3.2	3.4		
Italy	2.7		1.1	
Netherlands	2.5 - 3.0	3.0 - 3.5		
Norway	2.0 - 2.5	2.5 - 3.5		
Sweden	2.1	2.6		
United Kingdom	2.0			
Average	2.5	3.0	1.2	1.2
<u>Commercial Building Heat Pumps</u>				
Germany	2.1 - 2.3			
Greece	3.1			
Italy	2.9			
Netherlands	2.0 - 2.5			
Norway	2.5 - 3.5			
United Kingdom	2.4			
Average	2.45			

Source: *International Heat Pump Status and Policy Review*, Report HPC-AR3, IEA Heat Pump Centre, September, 1994 (Part 1, Tables 4.1 and 4.2).

Note: The reference listed above made no distinction between cooling and heating seasonal performance factors (SFP). For purposes of the analyses undertaken for this study, the SPFs for the electric systems above are assumed to be the same whether heating or cooling. The absorption system SPFs are considered to be for heating only. A value of 0.70 is used for a single-stage absorption cooling system to evaluate this technology for unitary cooling in the European climates.

Table 31. European commercial building data.

Country	New Building Stock: 1991 (units)	Average Building Heating/Cooling Load (kWh/y)
Commercial Building Heating Data		
Germany	35,200	80,000
Greece	1,100	22,100
Italy	20,000	145,000
Netherlands	900	8,800
Norway	200	1,300,000
United Kingdom	55,900	120,000
Total	113,200	112,500
Commercial Building Cooling Data		
Germany	35,200	200,000
Greece	1,100	30,100
Netherlands	900	6,500
Norway	200	10,000
United Kingdom	55,900	50,000
Total	93,200	105,800

APPENDIX E: UNITARY AIR CONDITIONING TEWI RESULTS

Table 32. TEWI calculations for unitary space conditioning equipment: Pittsburgh, Pennsylvania USA.

Equipment Lifetime	15	years				
Building Description	1800	ft^2	CO_2 Conversions:	0.0559	kg CO_2 / 1000 Btu gas input
	74,700,000	Btu/y heating		0.65	kg CO_2/kWh electricity
	16,100,000	Btu/y cooling		0.19	kg CO_2 / 1000 Btu I^2R

		Seasonal Efficiency		Indirect Effect			Direct Effect	
Equipment Description	Refrigerant	Cooling	Heating	Electricity	Gas	Parasitics	Refrigerant	TEWI
1996 Equipment	refrigerant emissions: make-up rate: 4% of charge per year, end-of-life loss: 15% of charge							
Air-to-Air Heat Pumps	HCFC-22	SEER - 10	HSPF - 7	122,376	0	0	3570	125,946
	HCFC-22	SEER - 12	HSPF - 8	107,742	0	0	3570	111,312
Premium Heat Pump Options								
Electric Air-to-Air	HCFC-22	SEER - 14	HSPF - 9	96,526	0	0	3570	100,096
Geothermal	HCFC-22	SEER -16.8	HSPF -12	70,038	0	0	1785	71,823
2005 Equipment	refrigerant emissions: make-up rate: 2% of charge per year, end-of-life loss: 15% of charge							
Air-to-Air Heat Pumps								
Direct Expansion	HCFC-22	SEER - 12.0	HSPF -8.0	107,742	0	0	2142	111,312
	R-407C	SEER - 12.0	HSPF - 8.0	107,742	0	0	1928	109,670
	R-410A	SEER - 12.6	HSPF - 8.4	103,113	0	0	1791	104,904
Secondary Loops	HC-290	SEER - 12	HSPF - 8					
5.0EC)T				121,514	0	4583	2	117,880
10EC)T				137,953	0	4583	2	132,398
Premium Heat Pump Options								
Electric Air-to-Air	HCFC-22	SEER - 16	HSPF - 9.5	88,957	0	0	2142	91,099
Geothermal	HCFC-22	SEER - 17.2	HSPF -12.8	66,027	0	0	1071	67,098

Table 33. TEWI calculations for unitary space conditioning equipment: Atlanta, Georgia USA.

Equipment Lifetime	15	years		
Building Description	1800	ft^2	CO_2 Conversions 0.0559	kg CO_2 / 1000 Btu gas input
	34,800,000	Btu/y heating	0.65	kg CO_2/kWh electricity
	33,800,000	Btu/y cooling	0.19	kg CO_2 / 1000 Btu I^2R

		Seasonal Efficiency		Indirect Effect			Direct Effect	
Equipment Description	Refrigerant	Cooling	Heating	Electricity	Gas	Parasitics	Refrigerant	TEWI
1996 Equipment	refrigerant emissions: make-up rate: 4% of charge per year, end-of-life loss: 15% of charge							
Air-to-Air Heat Pumps	HCFC-22	SEER - 10	HSPF - 7	81,426	0	0	3570	84,996
	HCFC-22	SEER - 12	HSPF - 8	69,875	0	0	3570	73,445
Central A/C and Electric Furnace	HCFC-22	SEER - 12	100%	31,036	0	99,440	3570	130,476
Premium Heat Pump Options								
Electric Air-to-Air	HCFC-22	SEER - 14	HSPF - 9	61,239	0	0	3570	64,809
Geothermal	HCFC-22	SEER - 16.8	HSPF - 12	47,891	0	0	1785	49,676
2005 Equipment	refrigerant emissions: make-up rate: 2% of charge per year, end-of-life loss: 15% of charge							
Air-to-Air Heat Pumps								
Direct Expansion	HCFC-22	SEER - 12	HSPF - 8	69,875	0	0	2142	72,017
	R-407C	SEER - 12	HSPF - 8	69,875	0	0	1928	71,803
	R-410A	SEER - 12.6	HSPF - 8.4	66,548	0	0	1791	6,8339
Secondary Loops	HC-290	SEER - 12	HSPF - 8					
5.0EC)T				75,149	0	3,466	2	78,615
10EC)T				87,102	0	3,466	2	90,568
Premium Heat Pump Options								
Electric Air-to-Air	HCFC-22	SEER - 16	HSPF - 9.5	56,313	0	0	2142	58,455
Geothermal	HCFC-22	SEER -17.2	HSPF - 12.8	45,666	0	0	1071	46,739

Table 34. TEWI calculations for unitary space conditioning equipment: Miami, Florida USA.

Equipment Lifetime	15	years		
Building Description	1800	ft^2	0.0559	kg CO_2 / 1000 Btu gas input
	0	Btu/y heating	0.65	kg CO_2/kWh electricity
	82,200,000	Btu/y cooling	0.19	kg CO_2 / 1000 Btu I^2R

Equipment Description	Refrigerant	Seasonal Cooling Efficiency	Indirect Effect: Electricity	Indirect Effect: Gas	Indirect Effect: Parasitics	Direct Effect: Refrigerant	TEWI
1996 Equipment	refrigerant emissions: make-up rate: 4% of charge per year, end-of-life loss: 15% of charge						
Air-to-Air Heat Pumps	HCFC-22	SEER - 10	80,145	0	0	3570	83,715
	HCFC-22	SEER - 12	66,788	0	0	3570	70,358
Premium Heat Pump Options							
Electric Air-to-Air	HCFC-22	SEER - 14	57,246	0	0	3570	60,816
Geothermal	HCFC-22	SEER - 16.8	47,705	0	0	1785	49,490
2005 Equipment	refrigerant emissions: make-up rate: 2% of charge per year, end-of-life loss: 15% of charge						
Electric Air Conditioning							
Direct Expansion	HCFC-22	SEER - 12	66,788	0	0	2142	68,930
	R-407C	SEER - 12	66,788	0	0	1928	68,716
	R-410A	SEER - 12.6	63,607	0	0	1791	65,398
Secondary Loops							
5.0EC)T	HC-290	SEER - 12	78,575	0	4,154	2	82,729
10EC)T	HC-290	SEER - 12	92,762	0	4,154	2	96,916
Premium Air-Conditioner Options							
Electric Air-to-Air	HCFC-22	SEER - 16	56,313	0	0	2142	58,455
Geothermal	HCFC-22		46,596	0	0	1071	47,667

Table 35. TEWI for 7.5 ton roof top air conditioners (RTU): cooling only.

Equipment Lifetime	15 years			
Refrigerant Emissions:			Refrigerant Charge:	
1996	1.50% of charge per year		HCFC-22	6.90 kg
2005	1.00% of charge per year		R-407C	6.90
End-of-Life	15% of charge		R-410A	5.70
Equivalent Full Load Cooling Hours		CO_2 Conversion Factor:	0.65 kg CO_2/kWh	
Pittsburgh	600			
Atlanta	1400			
Miami	2700			

Cooling Climate	Technology Base	Equipment Efficiency (EER)	Indirect Effect (kg CO_2)	Direct Effect (kg CO_2)	TEWI (kg CO_2)
Pittsburgh, Pennsylvania					
HCFC-22 RTU	1996	10	52,650	4399	57,049
HCFC-22	2005	11	47,834	3519	51,383
R-407C	2005	11	47,834	3167	50,801
R-410A	2005	11.6	45,584	2958	48,542
Atlanta, Georgia					
HCFC-22 RTU	1996	10	122,850	4399	127,249
HCFC-22	2005	11	111,682	3519	115,201
R-407C	2005	11	111,682	3167	114,849
R-410A	2005	11.6	106,364	2958	109,322
Miami, Florida					
HCFC-22 RTU	1996	10	236,925	4399	241,324
HCFC-22	2005	11	215,386	3519	218,905
R-407C	2005	11	215,386	3167	218,553
R-410A	2005	11.6	205,130	2958	208,088

Table 36. TEWI for European residential space heating (data from the *International Heat Pump Status and Policy Review Report*, PHC-AR3, IEA Heat Pump Centre, September 1994, Part 1.

Equipment Specification:							
Lifetime	15	years	Refrigerant Loss Data:		CO_2 Conversion Factors:		
Heat pump capacity	10	kW	Make-up rate	5% of charge/year	Electricity Generation	0.469	kg/kWh
Annual heating load	16,400	kW/year	End-of-Life Loss	15% of charge	Natural Gas Combustion	0.190	kg/kWh

		Electricity			Refrigerant Emissions						
Technology	Efficiency (SPF)	Heating (kWh/y)	Auxiliary (kWh/y)	Lifetime CO_2 (kg)	Charge (kg)	Leakage (g/y)	End-of-Life (g)	Lifetime Loss (kg)	Refrigerant GWP (kg CO_2/kg)	Direct Effect (kg CO_2)	TEWI (kg CO_2)
<u>Air-to-Air Heat Pump</u>											
HCFC-22	2.50	6560	0	46150	3.1	155	465	2.79	1700	4743	50,893
HFC-134a	2.50	6560	0	46150	2.7	135	405	2.43	1300	3159	49,309
R-407C (95% HCFC-22)	2.38	6891	0	48476	3.1	155	465	2.79	1530	4269	52,745
R-407C (100% HCFC-22)	2.50	6560	0	46150	3.1	155	465	2.79	1530	4260	50,418
R-410A (105% HCFC-22)	2.63	6560	0	46150	2.2	155	330	1.98	1730	3425	47,377
<u>Geothermal Heat Pumps</u>											
HCFC-22	3.00	5467	0	38458	3.1	155	465	2.79	1700	4743	43,021
HFC-134a	3.00	5467	0	38458	2.7	135	405	2.43	1300	3159	41,617
R-407C (95% HCFC-22)	2.85	5745	0	40482	3.1	155	465	2.79	1530	4269	44,751
R-407C (100% HCFC-22)	3.00	5467	0	38458	3.1	155	465	2.79	1530	4269	42,727
R-410A (105% HCFC-22)	3.15	5206	0	36627	2.2	155	330	1.98	1730	3425	40,052
Electric Resistance Heat	1.00	16,400	0	115,374	0	0	0	0	0	0	115,374

Table 37. TEWI for European residential cooling only (Greece). Source for loads and Seasonal Performance Factors (SPF): *International Heat Pump Status and Policy Review Report*, HPC-AR3, IEA Heat Pump Centre, September 1994.

Equipment Specification:						
Lifetime	15 years	Refrigerant Emission		CO_2 Conversion Factors:		
Heat pump capacity	10 kW	Data:	5% of charge/year	Electricity Generation	0.976	kg/kWh
Annual cooling load	7,500 kW/year	Make-up rate	15% of charge	Natural Gas Combustion	0.190	kg/kWh
		End-of-Life Loss				

		Electricity		Refrigerant Emissions						
Technology	Efficiency (SPF)	Cooling (kWh/y)	Lifetime CO_2 (kg)	Charge (kg)	Leakage (g/y)	End-of-Life (g)	Lifetime Loss (kg)	Refrigerant GWP (kg CO_2/kg)	Direct Effect (kg CO_2)	TEWI (kg CO_2)
Central Air Conditioner										
H FC-22	3.20	2344	34,313	3.1	155	465	2.79	1700	4743	39,056
HFC-134a	3.20	2344	34,313	2.7	135	405	2.43	1300	3159	37,472
R-407C (95% HCFC-22)	3.04	2467	36,118	3.1	155	465	2.79	1530	4269	40,387
R-407C (100% HCFC-22)	3.20	2344	34,313	3.1	155	465	2.79	1530	4260	38,581
R-410A (105% HCFC-22)	3.36	2232	32,679	2.2	155	330	1.98	1730	3425	34,182
Geothermal Heat Pumps										
HCFC-22	3.40	2206	32,294	3.1	155	465	2.79	1700	4743	37,037
HFC-134a	3.40	2206	32,294	2.7	135	405	2.43	1300	3159	35,453
R-407C (95% HCFC-22)	3.23	2322	33,994	3.1	155	465	2.79	1530	4269	38,263
R-407C (100% HCFC-22)	3.40	2206	32,294	3.1	155	465	2.79	1530	4269	36,563
R-410A (105% HCFC-22)	3.57	2101	30,756	2.2	155	330	1.98	1730	3425	34,182

Table 38. TEWI for unitary heating and cooling equipment in Japan.

	Refrigerant Emission Scenarios														
	½% annual; 15% end-of-life			½% annual; 10% end-of-life			1% annual; 15% end-of-life			1% annual; 5% end-of-life			½% annual; 5% end-of-life		
	R-22	R-407C	R-410A	R-22	R-407C	R-410A	R-22	R-407C	R-410A	R-22	R-407C	R-410A	R-22	R-407C	R-410A
Direct Effect															
refrigerant charge (kg)	1.00	0.93	1.00	1.00	0.93	1.00	1.00	0.93	1.00	1.00	0.93	1.00	1.00	0.93	1.00
leak rate (kg/y)	0.005	0.005	0.005	0.005	0.005	0.005	0.010	0.009	0.010	0.010	0.009	0.010	0.005	0.005	0.005
end-of-life loss (kg)	0.15	0.1395	0.15	0.10	0.093	0.100	0.150	0.1395	0.15	0.05	0.0465	0.05	0.05	0.0465	0.05
lifetime emissions (kg)	0.21	0.195	0.21	0.16	0.149	0.16	0.270	0.251	0.27	0.17	0.158	0.17	0.11	0.102	0.11
GWP	1700	1530	1730	1700	1530	1730	1700	1530	1730	1700	1530	1730	1700	1530	1730
Direct Effect (kg CO_2)	357	299	363	272	228	277	459	384	467	289	242	294	187	157	190
Indirect Effect (cooling)															
COP	2.67	2.67	2.67	2.67	2.67	2.67	2.67	2.67	2.67	2.67	2.67	2.67	2.67	2.67	2.67
operating time (hours/y)	350	350 2.80	350	350	350 2.80	350	350	350 2.80	350	350	350 2.80	350	350	350 2.80	350
system capacity (kW)	2.80	367.0	2.80	2.80	367.0	2.80	2.80	367.0	2.80	2.80	367.0	2.80	2.80	367.0	2.80
power input (kWh/y)	367.0	4405	367.0	367.0	4405	367.0	367.0	4405	367.0	367.0	4405	367.0	367.0	4405	367.0
lifetime input (kWh)	4405	2083	4405	4405	2083	4405	4405	2083	4405	4405	2083	4405	4405	2083	4405
Indirect Effect (kg CO_2)	2083		2083	2083		2083	2083		2083	2083		2083	2083		2083
Indirect Effects (heating)															
COP	3.20	3.20	3.20	3.20	3.20	3.20	3.20	3.20	3.20	3.20	3.20	3.20	3.20	3.20	3.20
operating time (hours/y)	450	450	450	450	450	450	450	450	450	450	450	450	450	450	450
system capacity (kW)	2.80	2.80	2.80	2.80	2.80	2.80	2.80	2.80	2.80	2.80	2.80	2.80	2.80	2.80	2.80
power input (kWh/y)	393.8	393.8	393.8	393.8	393.8	393.8	393.8	393.8	393.8	393.8	393.8	393.8	393.8	393.8	393.8
lifetime input (kWh)	4725	4725	4725	4725	4725	4725	4725	4725	4725	4725	4725	4725	4725	4725	4725
Indirect Effect (kg CO_2)	2235	2235	2235	2235	2235	2235	2235	2235	2235	2235	2235	2235	2235	2235	2235
TEWI (kg CO_2)	4675	4617	4681	4590	4546	4595	4777	4702	4785	4607	4560	4612	4505	4475	4508

Note: 2.8 kW split system, 12 year equipment lifetime; 0.473 kg CO_2 / kWh, heating and cooling hours for Tokyo

Table 39. TEWI for packaged air conditioners (PACs) for Japanese unitary applications.

	Emission Rates: 0.20% per year, 15% end-of-life					Emission Rates: 0.00% per year, 5% end-of-life				
	HCFC-22	R-407C		R-410A		HCFC-22	R-407C		R-410A	
Energy Use (Cooling)										
relative efficiency	100%	97%	100%	100%	105%	100%	97%	100%	100%	105%
COP	2.50	2.43	2.50	2.50	2.63	2.50	2.43	2.50	2.50	2.63
cooling capacity (kW)	14.0	14.0	14.0	14.0	14.0	14.0	14.0	14.0	14.0	14.0
power input (kW)	5.6	5.76	5.6	5.6	5.32	5.6	5.76	5.6	5.6	5.32
operating time (hours/y)	600	600	600	600	600	600	600	600	600	600
power input (kWh/y)	3360	3464	3360	3360	3200	3360	3464	3360	3360	3200
Energy Use(Heating)										
relative efficiency	100%	97%	100%	100%	105%	100%	97%	100%	100%	105%
COP	3.00	2.91	3.00	3.00	3.15	3.00	2.91	3.00	3.00	3.15
heating capacity (kW)	16.0	16.0	16.0	16.0	16.0	16.0	16.0	16.0	16.0	16.0
power input	5.33	5.49	5.33	5.33	5.08	5.33	5.49	5.33	5.33	5.08
operating time (hours/y)	900	900	900	900	900	900	900	900	900	900
power input (kWh/y)	4800	4948	4800	4800	4571	4800	4948	4800	4800	4571
Indirect Effect										
annual energy use (kWh/y)	8160	8412	8160	8160	7771	8160	8412	8160	8160	7771
lifetime energy use (kWh)	81,600	84,120	81,600	81,600	77,710	81,600	84,120	81,600	81,600	77,710
CO_2 emissions	38,597	39,791	38,597	38,597	36,759	38,597	39,791	38,597	38,597	36,759
Direct Effect										
charge size (kg)	4.45	4.45	4.45	4.45	4.45	4.45	4.45	4.45	4.45	4.45
refrigerant emissions (g/y)	8.9	8.9	8.9	8.9	8.9	0	0	0	0	0
end-of-life emissions (g)	668	668	668	668	668	223	223	223	223	223
lifetime emissions (kg)	0.757	0.757	0.757	0.757	0.757	0.223	0.223	0.223	0.223	0.223
GWP (kg CO_2/kg)	1500	1530	1500	1730	1730	1500	1530	1530	1730	1730
Equivalent CO_2 emissions	1135	1157	1157	1309	1309	334	340	340	385	385
TEWI (kg CO_2)	39,732	40,948	39,754	39,906	38,068	38,931	40,131	38,937	38,982	37,144

Note: equipment lifetime 10 years, power generation CO_2 emission factor 0.473 kg CO_2/kWh.

APPENDIX F: CALCULATION OF COOLING TOWER ELECTRICAL BURDEN FOR LARGE CHILLERS

F.1 CHILLER HEAT REJECTION

The total heat rejection for an electric chiller is the sum of the work <u>into</u> the compressor plus the refrigeration effect. The total heat of rejection (THR) can be expressed in terms of the coefficient of performance as follows:

$$THR = Qcool \cdot \left(\frac{1}{COP} + 1 \right)$$

If the chiller is hermetic, the COP is defined based on the electrical power supplied to the prime mover (the electric motor). If the chiller is an open-drive electric, the COP would be defined based on the shaft work into the compressor (THR is slightly lower since the electric motor heat is not rejected to the tower).

For absorption chillers, Fig. 41 shows the relevant energy flows where: *Qfue*l is the thermal energy associated with the energy source "firing" the chiller, Qaux is the energy associated with providing power to the chiller's auxiliaries (solution pumps, purge units, etc), *Qexhaust* is the thermal energy leaving the unit's exhaust via combustion byproducts (or exhaust steam for indirect fired), Qcool is the heat absorbed in the evaporator, *Qrej* is the total heat needed to be rejected to a tower. If Qaux ≈ Qexhaust (or if Qaux and Qexhaust are small, relative to the other energy flows), the relation for the total heat rejection, in terms of the COP and Qcool, is identical to the electric chiller case.

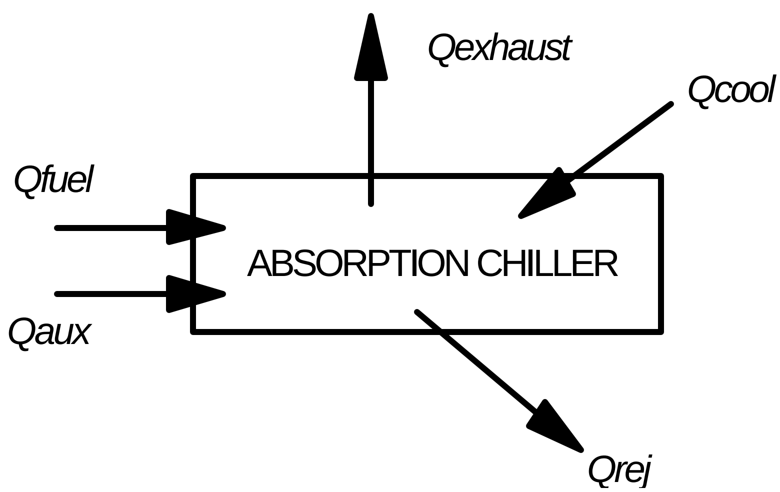

Figure 41. Heat flows for an absorption chiller.

$$THR = Qcool \cdot \left(\frac{1}{COP} + 1 \right)$$

The identical relation given above is true for an engine-driven chiller that rejects both the heat from the chiller as well as the engine jacket to the ambient environment through the cooling tower (assuming the COP is defined based on the fuel input).

Thus given the unit's COP and cooling capacity, the total heat rejection can be determined.

F.2 ELECTRICAL AUXILIARIES FOR HEAT REJECTION

The simplest approach is to determine the electrical energy (kWe) associated with condenser water pumps and cooling tower fans on a per unit heat rejection basis (kWt). The approach requires some encompassing assumptions but is more defensible than the previous approach which was based on a "rule-of-thumb".

F.2.1 Condenser Water Pumping

Assuming a 10°F range on the tower ($\Delta T_{condenser}=10°F$), the total heat rejection, on a kWt basis, will be a function of the condenser water flowrate (GPM) as given by:

$$\text{THR (kWt)} = 1.465\ \text{GPM}$$

Equating the above equation with the relation previously found for total heat rejection gives:

$$1.465\ \text{GPM} = \text{Qcool} \cdot \left(\frac{1}{\text{COP}} + 1 \right)$$

where: Qcool is given on a kW thermal basis.

The kWe for condenser water pumping can be determined, based on the water-side flow rate by assuming a condenser water pump efficiency, η_p, of 60%; motor efficiency, η_m, of 80%; and a water-side pressure drop, *head*, of 40 ft of $H_2 0$.

$$kWe_{pump} = \frac{GPM_{Head}}{5308 \cdot \eta_p \cdot \eta_m} = \frac{GPM \times 40}{(5308)(0.6)(0.8)} = \frac{GPM}{64}$$

or

$$GPM = 64 \times kWe_{pump}$$

Substituting the above expression into the heat rejection balance gives the pumping kWe as a function of the unit COP and cooling capacity.

$$kWe_{pump} = \frac{Qcool}{1.465 \times 64}\left(\frac{1}{COP} + 1\right)$$

F.2.2 Cooling Tower Fans

Several different configurations of cooling tower fans are available. The most common type for chiller applications is an induced draft type with either an axial or centrifugal fan. The data assumes a heat rejection factor of 1.25[6] and a 10 F temperature rise on the condenser.

The range of tower fan power is given in Table 40.

Table 40. Cooling tower fan power.

Fan Type	Tower Fan Ratio (kWe/kWt)
Axial	0.0128 - 0.0168
Centrifugal	0.0250 - 0.358

Treating all technologies equally, the tower fan power calculation can be simplified by assuming:

$$\frac{kWe_{fan}}{Qcool} = 0.0142\left(\frac{HRF}{1.25}\right) = 0.0114\left(\frac{1}{COP} + 1\right)$$

Thus, the total (pumping and tower fan) electrical energy associated with heat rejection equipment is given by,

$$kWe = \left(\frac{1}{93.76} + 0.0114\right) Qcool \cdot \left(\frac{1}{COP} + 1\right) = 0.022 \cdot (Qcool) \left(\frac{1}{COP} + 1\right)$$

where Q_{cool} is the unit cooling capacity in kW.

Table 41 uses the proposed method to calculate an appropriate cooling tower electrical burden

[6] The heat rejection factor is a dimensionless ratio of the total heat rejected to the cooling effect.

for each of the technologies assuming a 100 ton (352 kW Qcool) chiller.

Table 41. Cooling tower electrical burden.

	Electric Chiller	Engine Driven Chiller	Direct-Fired Double Effect Absorption Chiller	Direct-Fired Triple Effect Absorption Chiller
COP	6.40	2.00	1.05	1.45
Power				
Pump (kW)	4.33	5.62	7.31	6.34
Fan (kW)	4.64	6.02	7.83	6.78
Total	8.97	11.64	15.15	13.12

Acknowledgment: this method of assessing the additional loads associated with chiller tower operation was conceived and derived by D. Reindl at the University of Wisconsin-Madison.

APPENDIX G: SPREADSHEETS FOR CHILLER TEWI CALCULATIONS

Table 42. Parametric analysis of the indirect effects for chillers

IPLV (kW/RT	Cooling Tower Burden (kWe/RT)	Effective Efficiency (kW/RT)	Annual Energy Use (kWh/y/RT)	Lifetime Energy Use (kWh/RT)	Indirect Effect (kg CO_2)
0.45	0.087	0.54	1142	34,251	22,263
0.47	0.088	0.56	1185	35,554	23,110
0.49	0.088	0.58	1229	36,857	23,957
0.51	0.089	0.60	1272	38,160	24,804
0.53	0.089	0.62	1315	39,463	25,651
0.55	0.089	0.64	1359	40,766	26,498
0.57	0.090	0.66	1402	42,069	27,345
0.58	0.090	0.67	1424	42,721	27,788
0.59	0.090	0.68	1446	43,372	28,192
0.60	0.091	0.69	1467	44,024	28,615

Note: 2125 annual operating hours, 30 year equipment lifetime, CO_2 conversion factor 0.65 kg CO_2/kWhe

Table 42. Sensitivity of TEWI in chillers to refrigerant make-up rate.

Refrigerant	Refrigerant Emission Rate (% of charge/year/RT)					
	0.0	0.50	1.00	2.00	4.00	10.0
<u>CFC-12</u>						
lifetime refrigerant loss (kg/RT)	0.062	0.248	0.434	0.806	1.55	3.782
direct effect	527	2108	3689	6851	13,175	32,147
indirect effect	26,498	26,498	26,498	26,498	26,498	26,498
TEWI per RT	27,025	28,606	30,187	33,349	39,673	58,645
% direct effect	2.0%	7.4%	12.2%	20.5%	33.2%	54.8%
<u>CFC-11</u>						
lifetime refrigerant loss (kg/RT)	0.050	0.20	0.35	0.65	1.25	3.05
direct effect	200	800	1400	2600	5000	12,200
indirect effect	32,252	32,252	32,252	32,252	32,252	32,252
TEWI per RT	32,452	33,052	33,552	34,852	37,252	44,452
% direct effect	0.6%	2.4%	4.2%	7.5%	13.4%	27.4%
<u>HFC-134a</u>						
lifetime refrigerant loss (kg/RT)	0.041	0.162	0.284	0.527	1.013	2.471
direct effect	52.7	211	369	685	1316	3212
indirect effect	26,498	26,498	26,498	26,498	26,498	26,498
TEWI per RT	26,551	26,709	26,870	27,180	27,814	29,710
% direct effect	0.20%	0.79%	1.37	2.52%	4.73%	10.81%
<u>HCFC-123</u>						
lifetime refrigerant loss (kg/RT)	0.0525	0.210	0.368	0.683	1.313	3.203
direct effect	5	20	34	63	122	298
indirect effect	26,498	26,498	26,498	26,498	26,498	26,498
TEWI per RT	26,503	26,518	26,532	26,561	26,620	26,796
% direct effect	0.02%	0.07%	0.13%	0.24%	0.46%	1.11%
<u>HCFC-22</u>						
lifetime refrigerant loss (kg/RT)	0.041	0.16	0.28	0.53	1.01	2.47
direct effect	69	275	482	895	1721	4,200
indirect effect	26,498	26,498	26,498	26,498	26,498	26,498
TEWI per RT	26,667	26,773	26,980	27,393	28,219	30,980
% direct effect	0.26%	1.03%	1.79%	3.27%	6.10%	13.68%
<u>HFC-245ca</u>						
lifetime refrigerant loss (kg/RT)	0.038	0.15	0.26	0.49	0.94	2.29
direct effect	23	92	160	298	572	1396
indirect effect	26,498	26,498	26,498	26,498	26,498	26,498
TEWI per RT	26,521	26.590	26,658	26,796	27,070	27,894
% direct effect	0.09%	0.35%	0.60%	1.11%	2.11%	5.00%
<u>R-717</u>						
direct effect	0.00	0.00	0.00	0.00	0.00	0.00
indirect effect	26,498	26,498	26,498	26,498	26,498	26,498
TEWI per RT	26,498	26,498	26,498	26,498	26,498	26,498
% direct effect	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%

Notes: IPLV = 0.55 kW/ton, 2125 annual operating hours, 0.65 kg CO_2/kWhe

Table 44. TEWI for 3500 kW (1000 RT) chillers using data from Tables 42 and 43.

			Direct Effect				
			Refrigerant Make-Up Rate (% charge/year)				
1996 Technology	IPLV*	Indirect Effect	0%	0.5%	1.0%	2.0%	4.0%
Centrifugal Chillers:							
CFC-11	0.58	27,768,000	200,000	800,000	1,400,000	2,600,000	5,000,000
CFC-12	0.59	28,192,000	527,000	2,108,000	3,689,000	6,851,000	13,175,000
HCFC-123	0.47	23,110,000	5,000	20,000	34,000	63,000	122,000
HFC-134a	0.54	26,074,500	52,700	211,000	369,000	685,000	1,316,000
HCFC-22	0.54	26,074,500	69,000	275,000	482,000	895,000	1,721,000
Screw Chillers:							
HCFC-22	0.60	28,615,000	69,000	275,000	482,000	895,000	1,721,000
R-717	0.60	28,615,000	0	0	0	0	0
HFC-134a Engine Driven Chillers	2.30	23,193,000	52,700	211,000	396,000	685,000	1,316,000
Direct-Fired Double Effect Absorption	1.07	47,577,000	0	0	0	0	0
2005 Technology							
Centrifugal Chillers:							
HCFC-123	0.45	22,263,000	5,000	20,000	34,000	63,000	122,000
HFC-134a	0.48	23,533,500	52,700	211,000	369,000	685,000	1,316,000
HCFC-22	0.48	23,533,500	69,000	275,000	482,000	895,000	1,721,000
Screw Chillers:							
HCFC-22	0.58	27,788,000	69,000	275,000	482,000	895,000	1,721,000
R-717	0.57	27,345,000	0	0	0	0	0
Engine Driven Chillers:							
HCFC-22	2.40	22,360,000	69,000	275,000	482,000	895,000	1,721,000
HFC-134a	2.40	22,360,000	52,700	211,000	369,000	685,000	1,316,000
Direct-Fired Absorption Chillers:							
Double Effect	1.15	44,592,000	0	0	0	0	0
Triple Effect	1.50	35,712,000	0	0	0	0	0

*IPLVs given as kW/RT for electric powered chillers and gas COPs for thermally powered chillers, 3500 kW (1000 RT) cooling capacity, 30 year equipment lifetime

Table 45. TEWI for direct-gas fired absorption chillers.

	Double Effect Chiller IPL (gCOP)				Triple Effect Chiller IPL (gCOP)	
	1.00	1.05	1.07	1.15	1.45	1.50
Gas:						
g CO_2/1000 Btu	55.9	55.9	55.9	55.9	55.9	55.9
kg CO_2/12,000 Btu	0.671	0.671	0.671	0.671	0.671	0.671
gas COP	1.00	1.05	1.07	1.15	1.45	1.50
kg CO_2/hton	0.671	0.638	0.626	0.583	0.462	0.447
kg CO_2/year/ton	1,424	1,356	1,331	1,238	982	949
kg CO_2/ton	42,713	40,679	39,918	37,141	29,457	28,475
Electricity:						
kW/ton cooling	3.516	3.516	3.516	3.516	3.516	3.516
gas COP	1.00	1.05	1.07	1.15	1.45	1.50
peripherals (kW/ton)	0.035	0.035	0.035	0.035	0.046	0.046
cooling tower burden (kW/ton)	0.155	0.151	0.150	0.145	0.131	0.129
kW/ton - total	0.1899	0.1862	0.1848	0.1798	0.1764	0.1747
kWh/year/ton	403.5	395.7	392.8	382.1	374.9	371.1
kWh/ton lifetime	12,106	11,871	11,783	11,463	11,248	11,134
kg CO_2/ton	7,869	7,716	7,659	7,451	7,311	7,237
TEWI (per Rton)	50,581	48,395	47,577	44,592	36,768	35,712

Note: 2125 annual operating hours (1200 EFLH adjusted for IPLV) 30 year lifetime, 0.65 kg CO_2/kWhe, 55.9 g CO_2/1000 Btu (assumes 96.5% distribution efficiency).

Table 46. TEWI for natural gas engine driven chillers using HCFC-22, HFC-134a, or R-717 (screw and centrifugal compressors, HFC-134a).

	Primary Fuel IPLVs (COP)					
Source of Emissions	1.90	1.95	2.10	2.20	2.30	2.40
Indirect Effect (gas):						
g CO_2/1000 Btu	55.9	55.9	55.9	55.9	55.9	55.9
kg CO_2/12,000 Btu	0.671	0.671	0.671	0.671	0.671	0.671
kg CO_2/hr/RT w/ gCOP	0.353	0.344	0.319	0.305	0.292	0.280
kg CO_2/year/ton	750	731	679	648	620	594
Indirect Effect (kg CO_2/Rton)	22,507	21,930	20,364	19,438	18,593	17,818
Indirect Effect (electric)*:						
kW/RT	0.118	0.117	0.114	0.113	0.111	0.110
kWh/year/RT	251	249	243	239	236	233
kWh/ton-lifetime	7528	7462	7281	7174	7077	6987
kg CO_2/kWh	0.65	0.65	0.65	0.65	0.65	0.65
Indirect Effect (kg CO_2/Rton)	4893	4850	4733	4663	4600	4542
Total Indirect Effect (kg CO_2/Rton)	27,401	26,780	25,096	24,101	23,193	22,360

Note: 2125 annual operating hours, 30 year lifetime, 55.9 g CO2/1000 Btu natural gas (96.5% distribution efficiency)
*cooling tower contribution only

Table 47. Direct effects taken from Table 43.

Refrigerant Loss Data	HCFC-22	HFC-134a	R-717
Charge (kg/ton)	0.81	0.81	0.46
Make-Up Rate (kg/ton/year)	0.0162	0.0162	0.0092
Lifetime Emissions (kg/ton)			
make-up	0.486	0.486	0.276
end-of-life	0.041	0.041	0.023
total	0.527	0.527	0.299
GWP	1700	1300	0
Direct Effect (kg CO_2/ton)	895	685	0

Note: 2.0% of charge per year make-up rate, 30 year lifetime, end-of-life loss of 5% of charge

Table 48. TEWI for centralized chiller applications in Japan.

	HCFC-123	HFC-134a	Double Effect Absorption Chillers Direct Fired	Double Effect Absorption Chillers Indirect-Fired
Direct Effect				
refrigerant charge (kg)	255	255	0	0
loss rate (% charge/year)	1.0%	0.50%	0	0
annual losses (kg)	2.55	1.28	0	0
lifetime losses (kg)	63.75	31.87	0	0
end-of-life losses (kg)	12.8	12.8	0	0
lifetime emissions (kg)	76.5	44.6	0	0
GWP	93	1300	0	0
direct effect	7115	58,013	0	0
Indirect Effect: Electricity				
a. power consumption (kW/RT)	0.70	0.70	0.035	0.025
1055 kW (300 RT) chiller power (kW)	210	210	0	0
annual power consumption (kWh)	147,000	147,000	0	0
b. cooling tower power per ton (kW/RT)	0.14	0.14	0.22	0.22
cooling tower for 1055 kW (300 RT) (kW)	42	42	66	66
annual cooling tower power (kWh)	29,400	29,400	45,465	45,465
c. absorption auxiliaries 1055 kW chiller (kW)	0	0	10.5	7.5
annual absorption auxiliary (kWh)	0	0	7350	5,250
d. lifetime electrical energy use (kWh)	4,410,000	4,410,000	1,320,375	1,267,875
e. lifetime CO2 emissions	2,085,930	2,085,930	624,537	599,705
Indirect Effect: Natural Gas				
system COP	NA	NA	1.00	1.00
boiler efficiency	NA	NA	-----	80%
gas consumption 1055 kW (300 RT) chiller (kW)	NA	NA	1055	1319
annual gas consumption (kWh)	NA	NA	738,500	923,125
CO_2 emissions (kg/year)	NA	NA	140,868	176,085
lifetime CO_2 emissions (kg)	NA	NA	3,521,700	4,402,125
TEWI (kg CO_2)	2,093,045	2,143,943	4,146,237	5,001,183

Notes: equipment lifetime 25 years, CO_2 conversion factor 0.473 kg CO_2/kWhe, typical chiller capacity 1055 kW (300 RT), 0.671 kg CO_2/12,000 Btu gas input (0.190 kg CO_2/kWh gas input), 700 annual full load operating hours, 20% auxiliary power consumption

APPENDIX H: SECONDARY HEAT TRANSFER LOOP CALCULATIONS FOR COMMERCIAL REFRIGERATION

Table 49. Secondary loop pumping power for commercial refrigeration applications.

	North America		Europe		Japan	
Secondary Loop Parameters	low temp	medium temp	low temp	medium temp	low temp	medium temp
Refrigeration Load (kW)	88	264	44	132	24	127
Secondary Loop)T (EC)	2.8	2.8	2.8	2.8	2.8	2.8
<u>Pipe Parameters</u>						
equivalent length (m)	122	122	61	61	46	46
e/D	0.0012	0.0012	0.0012	0.0012	0.0012	0.0012
ID (mm)	53	53	53	53	41	53
<u>Fluid Properties</u>						
density (kg/m^3)	1218	1207	1218	1207	1218	1207
velocity (m/s)	1.3	1.0	2.0	2.0	1.8	1.2
dynamic viscosity (mPa-s)	0.942	5.87	19.2	5.87	19.2	5.87
kinematic viscosity (mm^2/sec)	23.3	7.10	23.3	7.10	23.3	7.10
thermal conductivity (W/mEC)	0.439	0.455	0.439	0.455	0.439	0.455
specific heat (kJ/kgEC)	2.976	3.036	2.976	3.036	2.976	3.036
<u>Fluid Characteristics</u>						
Reynolds number	3027	7369	4541	14,700	3162	8507
Prandtl number	6.4	39.2	130.0	39.2	130.0	39.2
Nusselt number	2.6	9.6	9.7	16.7	7.2	10.7
friction factor	0.0206	0.0206	0.0206	0.0250	0.0206	0.0206
Pressure Drop (kPa)	105.3	57.3	118.3	114.5	91.1	28.6
Pumping Power (kW)	1.2	2.0	0.7	2.0	0.3	0.5
Annual Energy Input (kWh)	10,700	17,300	6035	17,300	2520	4160

APPENDIX I: COMPLETE RESULTS FOR COMMERCIAL REFRIGERATION

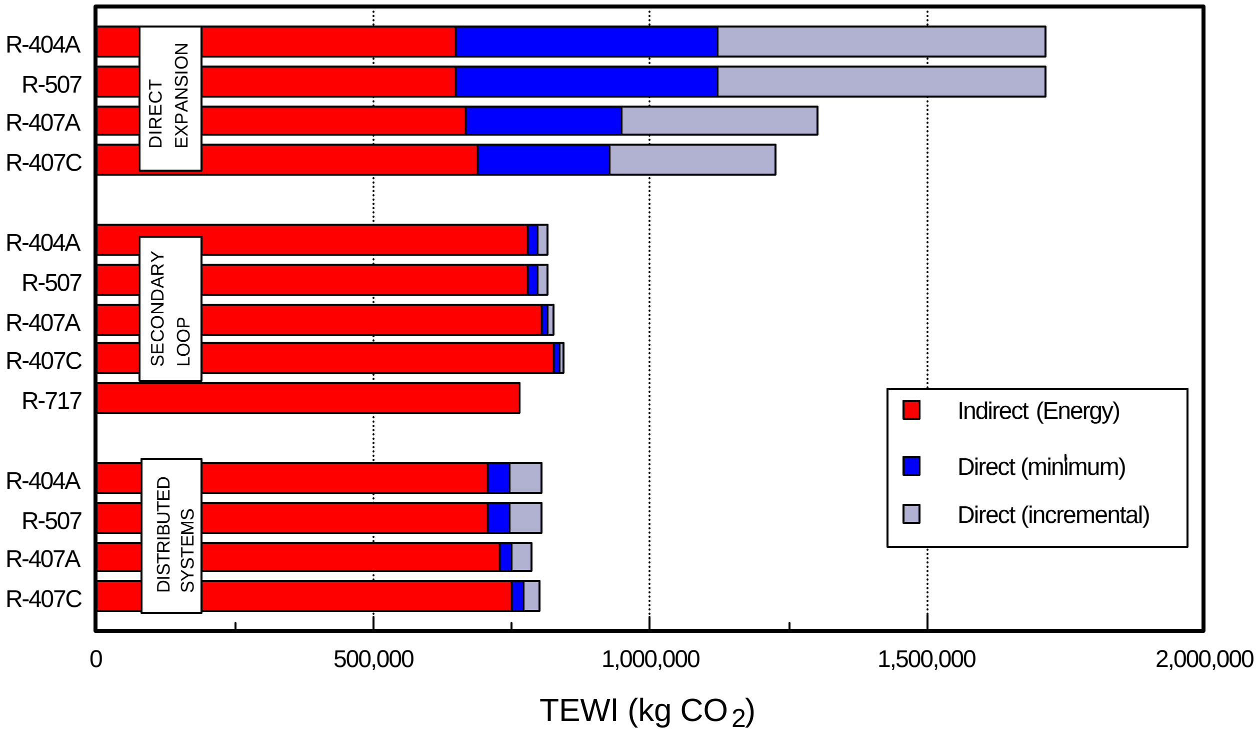

Figure 42. Low temperature refrigeration in Europe.

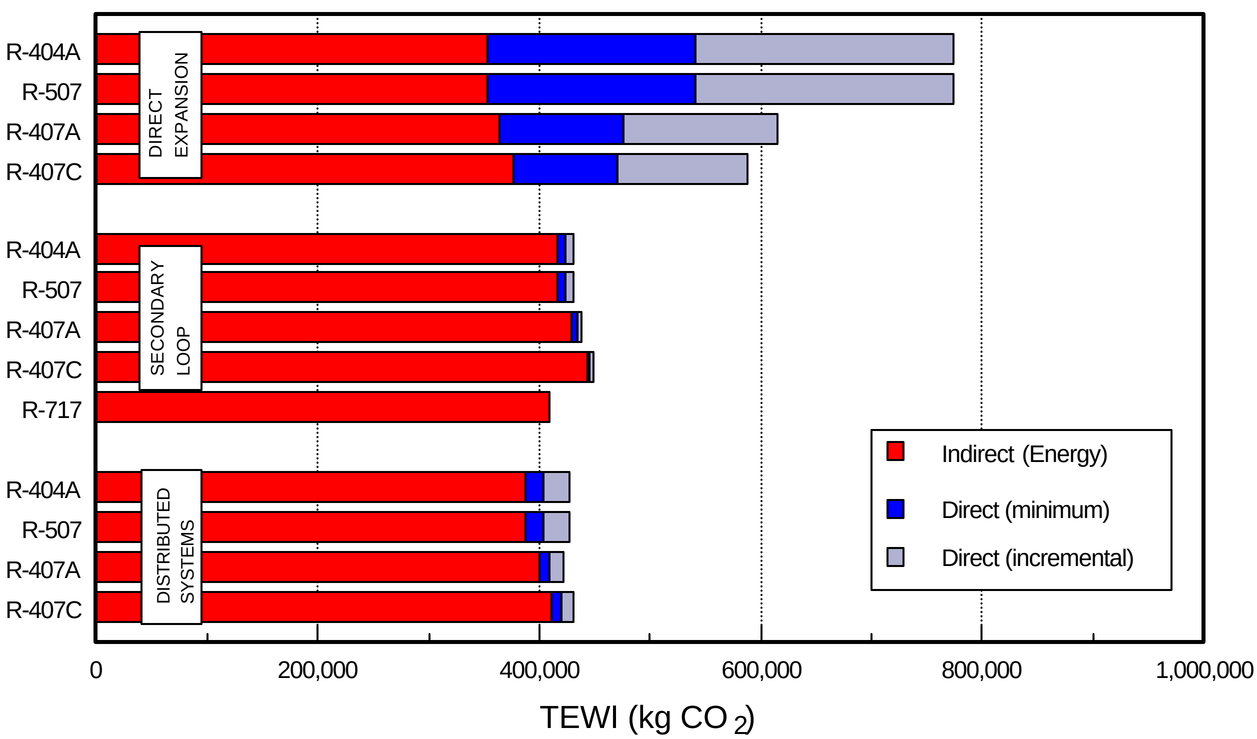

Figure 43. Low temperature refrigeration in Japan.

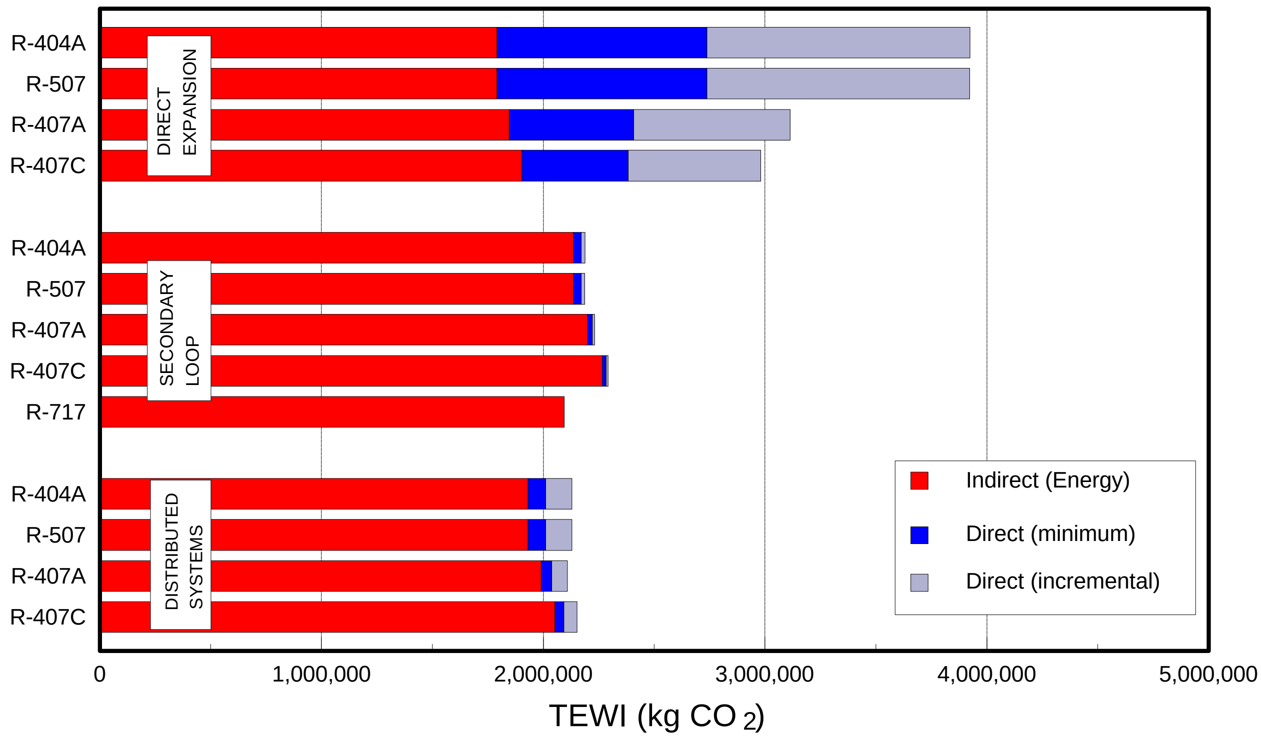

Figure 44. Low temperature refrigeration in North America.

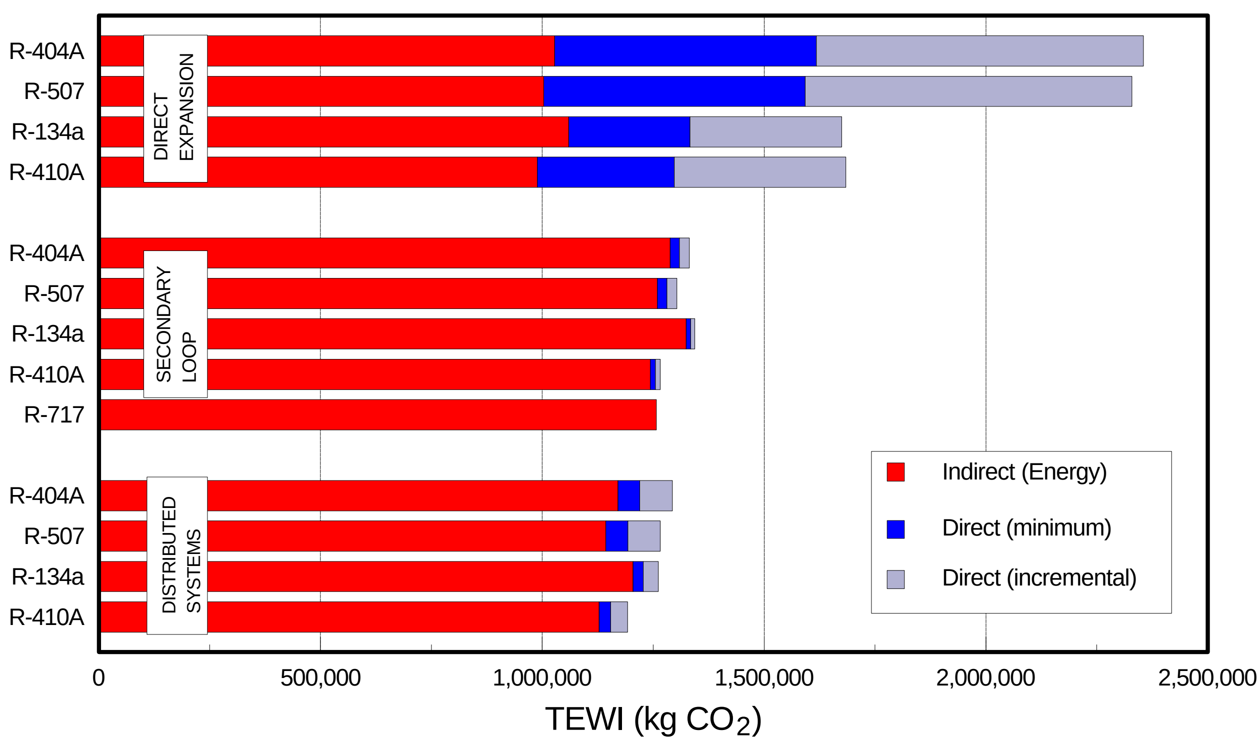

Figure 45. Medium temperature refrigeration in Europe.

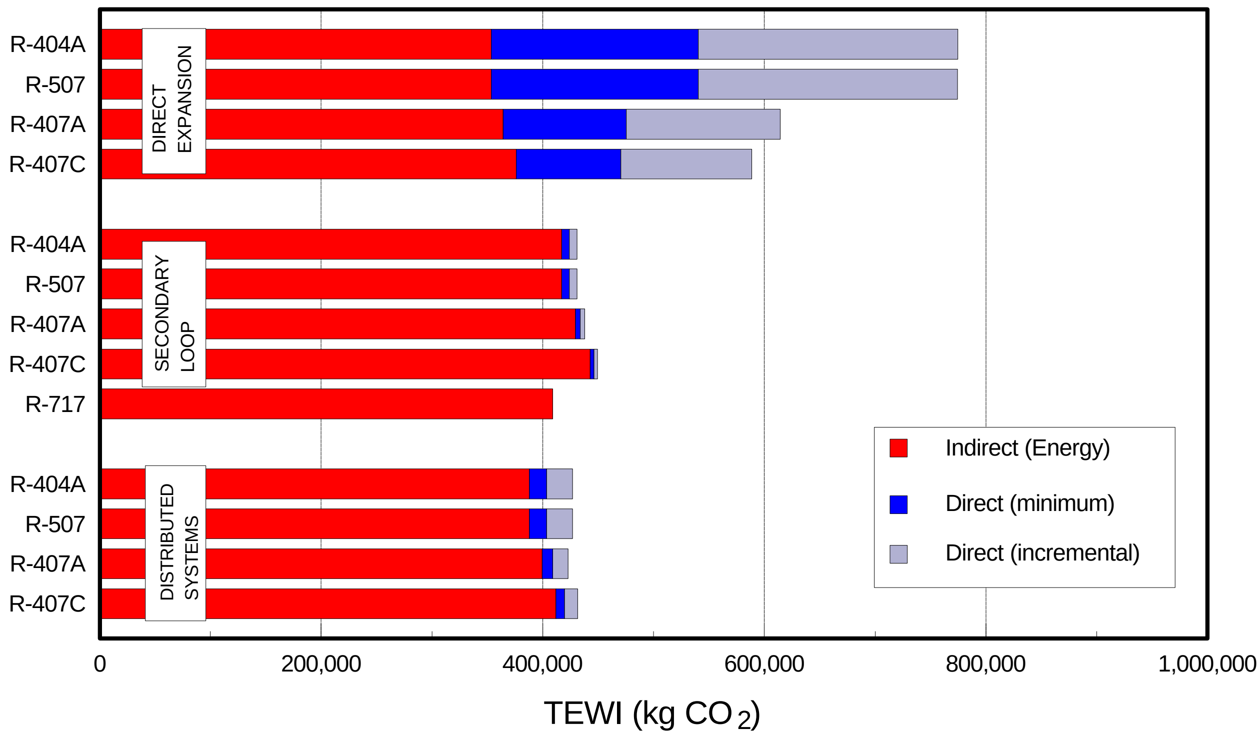

Figure 46. Medium temperature refrigeration in Japan.

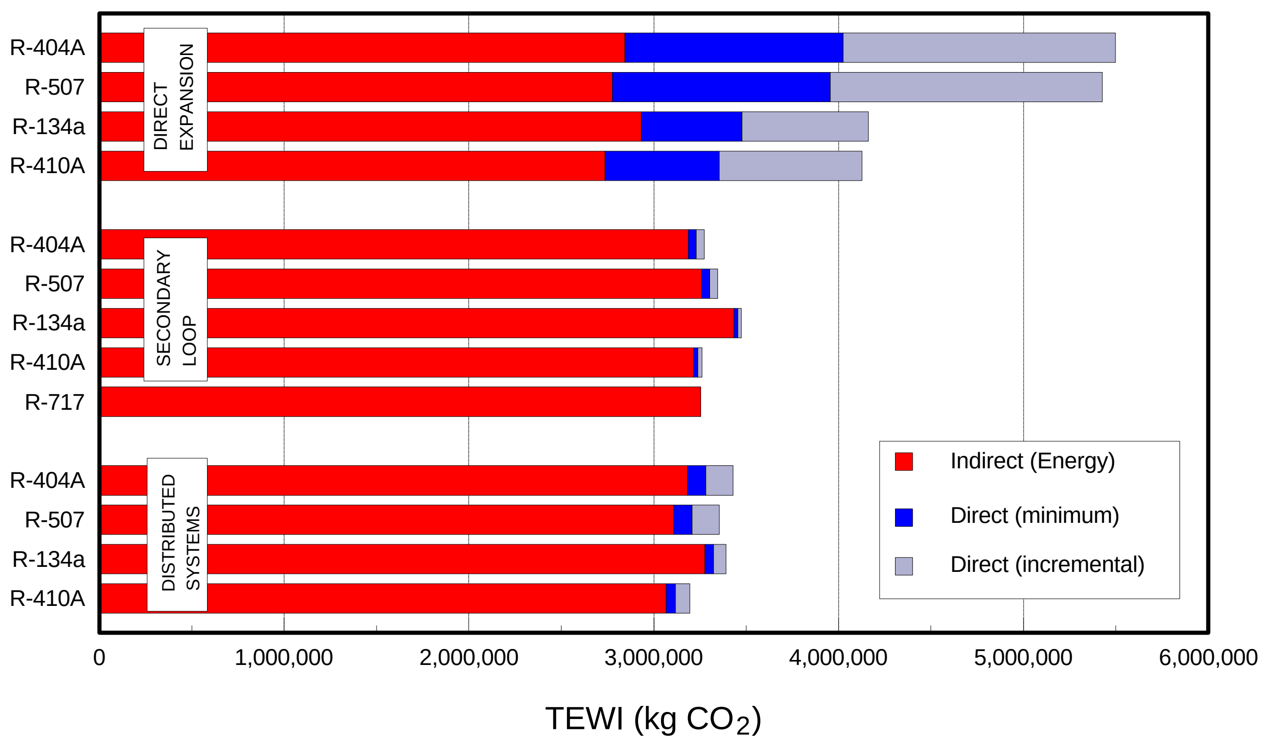

Figure 47. Medium temperature refrigeration in North America.

Table 50. TEWI for low and medium temperature direct expansion refrigeration systems in North America.

	Low Temperature Refrigeration				Medium Temperature Refrigeration			
Component of TEWI	R-404A	R-507	R-407A	R-407C	R-404A	R-507	R-134a	R-410A
1. Compressor Calculations								
a. design load (kW)	87.9	87.9	87.9	87.9	263.7	263.7	263.7	263.7
b. fraction on-time	40%	40%	40%	40%	40%	40%	40%	40%
c. cooling delivered (kWh/y)	307,999	307,999	307,999	307,999	923,966	923,966	923,966	923,966
d. pumping power (kWh/y)	0	0	0	0	0	0	0	0
e. total cooling required (kWh/y)	307,999	307,999	307,999	307,999	923,996	923,996	923,996	923,996
f. COP	1.762	1.762	1.708	1.654	3.425	3.515	3.317	3.569
g. compressor input (kWh/y)	174.797	174,797	180,317	186,197	269,800	262,882	278,598	258,899
2. Condenser Calculations								
a. refrigeration load (kWh/y)	307,999	307,999	307,999	307,999	923,996	923,996	923,996	923,996
b. pumping power (kWh/y)	0	0	0	0	0	0	0	0
c. compressor power (kWh/y)	174,797	174,797	180,317	186,197	269,800	262,882	278,598	258,899
d. total heat rejected (kWh/y)	482,796	482,796	488,315	494,195	1,193,796	1,186,878	1,202,594	1,182,895
e. required fan power (kWh/y)	8,835	8,835	8,936	9,044	21,846	21,720	22,007	21,647
3. Total Energy Consumption								
a. compressor (kWh/y)	174,797	174,797	180,317	186,197	269,800	262,882	278,598	258,899
b. secondary fluid pump (kWh/y)	0	0	0	0	0	0	0	0
c. condenser fan (kWh/y)	8,835	8,835	8,936	9,044	21,846	21,720	22,007	21,6647
d. total annual energy (kWh/y)	183,632	183,632	189,253	195,240	291,646	284,602	300,605	280,546
e. lifetime energy (kWh)	2,754,478	2,754,478	2,838,792	2,928,604	4,374,692	4,269,024	4,509,075	4,208,185
f. lifetime CO_2 emissions (kg)	1,790,411	1,790,411	1,845,215	1,903,593	2,843,550	2,774,866	2,930,899	2,735,320
4. Refrigerant Emissions								
a. refrigerant GWP	3260	3300	1770	1530	3260	3300	1300	1730
b. refrigerant charge (kg)	323	319	353	347	402	397	467	397
c. current practice (13.5% loss/year)								
(1) lifetime emissions (kg)	654	646	716	704	814	804	947	804
(2) direct effect	2,133,177	2,132,317	1,266,949	1,076,358	2,653,710	2,652,640	1,230,459	1,390,626
d. near term practice (6% loss/year)								
(1) lifetime emissions (kg)	291	287	318	313	362	357	421	357
(2) direct effect	948,079	947,697	563,088	478,381	1,179,427	1,178,951	546,871	618,056
TEWI								
a. current practice								
(1) indirect effect (energy use)	1,790,411	1,790,411	1,845,200	1,903,600	2,843,600	2,774,900	2,930,900	2,735,300
(2) direct effect (emissions)	2,133,200	2,132,300	1,266,900	1,076,400	2,653,700	2,652,600	1,230,500	1,390,600
(3) TEWI	3,923,600	3,922,711	3,112,100	2,980,000	5,497,300	5,427,500	4,161,400	4,125,900
b. near term practice								
(1) indirect effect (energy use)	1,790,411	1,790,411	1,845,200	1,903,600	2,843,600	2,774,900	2,930,900	2,735,300
(2) direct effect (emissions)	948,100	947,700	563,100	478,400	1,179,400	1,179,000	546,900	618,100
(3) TEWI	2,738,500	2,738,100	2,408,300	2,382,000	4,023,000	3,953,900	3,477,800	3,353,400

Table 51. TEWI for low and medium temperature distributed refrigeration systems in North America.

	Low Temperature Refrigeration				Medium Temperature Refrigeration			
Component of TEWI	R-404A	R-507	R-407A	R-407C	R-404A	R-507	R-134a	R-410A
1. Compressor Calculations								
a. design load (kW)	87.9	87.9	87.9	87.9	263.7	263.7	263.7	263.7
b. fraction on-time	40%	40%	40%	40%	40%	40%	40%	40%
c. cooling delivered (kWh/y)	307,999	307,999	307,999	307,999	923,996	923,996	923,996	923,996
d. pumping power (kWh/y)	4,380	4,380	4,380	4,380	12,895	12,895	12,895	12,895
e. total cooling required (kWh/y)	312,379	312,379	312,379	312,379	936,891	936,891	936,891	936,891
f. COP	1.692	1.692	1.640	1.588	3.219	3.304	3.118	3.355
g. compressor input (kWh/y)	184,669	184,669	190,501	196,713	291,026	283,564	300,516	279,268
2. Condenser Calculations								
a. refrigeration load (kWh/y)	312,379	312,379	312,379	312,379	936,891	936,891	936,891	936,891
b. pumping power (kWh/y)	4,380	4,380	4,380	4,380	12,895	12,895	12,895	12,895
c. compressor power (kWh/y)	184,669	184,669	190,501	196,713	291,026	283,564	300,516	279,268
d. total heat rejected (kWh/y)	501,428	501,428	507,260	513,472	1,240,812	1,233,350	1,250,302	1,229,054
e. required fan power (kWh/y)	9,176	9,176	9,283	9,397	22,707	22,570	22,881	22,492
3. Total Energy Consumption								
a. compressor (kWh/y)	184,669	184,669	190,501	196,713	291,026	283,564	300,516	279,268
b. secondary fluid pump (kWh/y)	4,380	4,380	4,380	4,380	12,895	12,895	12,895	12,895
c. condenser fan (kWh/y)	9,176	9,176	9,283	9,397	22,707	22,570	22,881	22,492
d. total annual energy (kWh/y)	198,225	198,225	204,164	210,489	326,628	319,029	336,292	314,654
e. lifetime energy (kWh)	2,973,381	2,973,381	3,062,457	3,157,342	4,899,420	4,785,439	5,044,375	4,719,813
f. lifetime CO_2 emissions (kg)	1,932,698	1,932,698	1,990,597	2,052,272	3,184,623	3,110,535	3,278,844	3,067,878
4. Refrigerant Emissions								
a. refrigerant GWP	3260	3300	1770	1530	3260	3300	1300	1730
b. refrigerant charge (kg)	81	80	88	87	100	99	117	99
c. current practice (5% loss/year)								
(1) lifetime emissions (kg)	61	60	66	65	75	74	88	74
(2) direct effect	197,516	197,437	117,310	99,663	245,714	245,615	113,931	128,762
d. near term practice (2% loss/year)								
(1) lifetime emissions (kg)	24	24	27	26	30	30	35	30
(2) direct effect	79,007	78,975	46,924	39,865	98,286	98,246	45,573	51,505
TEWI								
a. current practice								
(1) indirect effect (energy use)	1,932,700	1,932,700	1,990,600	2,052,300	3,184,600	3,110,500	3,278,800	3,067,900
(2) direct effect (emissions)	197,500	197,400	117,300	99,700	245,700	245,600	113,900	128,800
(3) TEWI	2,130,200	2,130,100	2,107,900	2,152,000	3,430,300	3,356,100	3,392,700	3,196,700
b. near term practice								
(1) indirect effect (energy use)	1,932,700	1,932,700	1,990,600	2,052,300	3,184,600	3,110,500	3,278,800	3,067,900
(2) direct effect (emissions)	79,000	79,000	46,900	39,900	98,300	98,200	45,600	51,500
(3) TEWI	2,011,700	2,011,700	2,037,500	2,092,200	3,282,900	3,208,700	3,324,400	3,119,400

Table 52. TEWI for low and medium temperature direct expansion refrigeration systems in Europe.

	Low Temperature Refrigeration				Medium Temperature Refrigeration			
Component of TEWI	R-404A	R-507	R-407A	R-407C	R-404A	R-507	R-134a	R-410A
1. Compressor Calculations								
a. design load (kW)	43.9	43.9	43.9	43.9	131.8	131.8	131.8	131.8
b. fraction on-time	40%	40%	40%	40%	40%	40%	40%	40%
c. cooling delivered (kWh/y)	153,999	153,999	153,999	153,999	461,998	461,998	461,998	461,998
d. pumping power (kWh/y)	0	0	0	0	0	0	0	0
e. total cooling required (kWh/y)	153,999	153,999	153,999	153,999	461,998	461,998	461,998	461,998
f. COP	1.762	1.762	1.708	1.654	3.425	3.515	3.317	3.569
g. compressor input (kWh/y)	87,398	87,398	90,158	93,098	134,900	131,441	139,299	129,449
2. Condenser Calculations								
a. refrigeration load (kWh/y)	153,999	153,999	153,999	153,099	461,998	461,998	461,998	461,998
b. pumping power (kWh/y)	0	0	0	0	0	0	0	0
c. compressor power (kWh/y)	87,398	87,398	90,158	93,098	134,900	131,441	139,299	129,449
d. total heat rejected (kWh/y)	241,398	241,398	244,158	247,098	596,898	593,439	601,297	591,448
e. required fan power (kWh/y)	4,418	4,418	4,468	4,522	10,923	10,860	11,004	10,823
3. Total Energy Consumption								
a. compressor (kWh/y)	87,398	87,398	90,158	93,098	134,900	131,441	139,299	129,449
b. secondary fluid pump (kWh/y)	0	0	0	0	0	0	0	0
c. condenser fan (kWh/y)	4,418	4,418	4,468	4,522	10,923	10,860	11,004	10,823
d. total annual energy (kWh/y)	91,816	91,816	94,626	97,6620	145,823	142,301	150,302	140,273
e. lifetime energy (kWh)	1,377,239	1,377,239	1,419,396	1,464,302	2,187,346	2,134,512	2,254,537	2,104,093
f. lifetime CO_2 emissions (kg)	647,302	647,302	667,116	688,222	1,028,053	1,003,221	1,059,633	988,923
4. Refrigerant Emissions								
a. refrigerant GWP	3260	3300	1770	1530	3260	3300	1300	1730
b. refrigerant charge (kg)	162	160	177	174	201	198	234	198
c. current practice (13.5% loss/year)								
(1) lifetime emissions (kg)	327	323	358	352	407	402	473	402
(2) direct effect	1,066,589	1,066,159	633,474	538,179	1,326,855	1,326,320	615,230	695,313
d. near term practice (6% loss/year)								
(1) lifetime emissions (kg)	145	144	159	156	181	179	210	179
(2) direct effect	474,039	473,848	281,544	239,191	589,713	589,476	273,435	309,028
TEWI								
a. current practice								
(1) indirect effect (energy use)	647,300	647,300	667,100	688,200	1,028,100	1,003,200	1,059,600	988,900
(2) direct effect (emissions)	1,066,600	1,066,200	633,500	538,200	1,326,900	1,326,300	615,200	695,300
(3) TEWI	1,713,900	1,713,500	1,300,600	1,226,400	2,355,000	2,329,500	1,674,800	1,684,200
b. near term practice								
(1) indirect effect (energy use)	647,300	647,300	667,100	688,200	1,028,100	1,003,200	1,059,600	988,900
(2) direct effect (emissions)	474,000	473,800	281,500	239,200	589,700	589,500	273,400	309,000
(3) TEWI	1,121,300	1,121,100	948,600	927,400	1,617,800	1,592,700	1,333,000	1,297,900

Table 53. TEWI for low and medium temperature secondary loop refrigeration systems in North America.

	Low Temperature Refrigeration					Medium Temperature Refrigeration				
Component of TEWI	R-404A	R-507	R-407A	R-407C	R-717	R-404A	R-507	R-134a	R-410A	R-717
1. Compressor Calculations										
a. design load (kW)	87.9	87.9	87.9	87.9	87.9	263.7	263.7	263.7	263.7	263.7
b. fraction on-time	40%	40%	40%	40%	40%	40%	40%	40%	40%	40%
c. cooling delivered (kWh/y)	307,999	307,999	307,999	307,999	307,999	923,996	923,996	923,996	923,996	923,996
d. pumping power (kWh/y)	10,775	10,775	10,775	10,775	10,775	17,345	17,345	17,345	17,345	17,345
e. total cooling required (kWh/y)	318,774	318,774	318,774	318,774	318,774	941,341	941,341	941,341	941,341	941,341
f. COP	1.603	1.603	1.554	1.505	1.638	3.117	3.199	3.018	3.248	3.205
g. compressor input (kWh/y)	198,804	198,804	205,082	211,770	194,633	302,049	294,304	311,898	289,845	293,701
2. Condenser Calculations										
a. refrigeration load (kWh/y)	318,774	318,774	318,774	318,774	318,774	923,996	923,996	923,996	923,996	923,996
b. pumping power (kWh/y)	10,775	10,775	10,775	10,775	10,775	17,345	17,345	17,345	17,345	17,345
c. compressor power (kWh/y)	198,804	198,804	205,082	211,770	194,633	302,049	294,304	311,898	289,845	293,701
d. total heat rejected (kWh/y)	528,353	528,352	534,631	541,318	524,182	1,234,390	1,235,645	1,253,239	1,231,186	1,235,043
e. required fan power (kWh/y)	9,669	9,669	9,784	9,906	9,592	22,754	22,612	22,934	22,531	22,601
3. Total Energy Consumption										
a. compressor (kWh/y)	198,804	198,804	205,082	211,700	194,633	302,049	294,304	311,898	289,845	293,701
b. secondary fluid pump (kWh/y)	10,775	10,775	10,775	10,775	10,775	17,345	17,345	17,645	17,345	17,345
c. condenser fan (kWh/y)	9,669	9,669	9,784	9,906	9,592	22,754	22,612	22,934	22,531	22,601
d. total annual energy (kWh/y)	219,248	219,248	225,641	232,450	215,000	342,147	334,261	352,177	329,720	333,647
e. lifetime energy (kWh)	3,288,720	3,288,716	3,384,609	3,486,757	3,225,003	5,132,212	5,013,913	5,282,656	4,945,802	5,004,711
f. lifetime CO_2 emissions (kg)	2,137,688	2,137,665	2,199,669	2,266,392	2,096,252	3,335,938	3,259,044	3,433,727	3,214,771	3,253,062
4. Refrigerant Emissions										
a. refrigerant GWP	3260	3300	1770	1530	0	3260	3300	1300	1730	0
b. refrigerant charge (kg)	36	35	39	38	18	44	44	51	44	23
c. current practice (4% loss/year)										
(1) lifetime emissions (kg)	21	21	23	23	11	27	26	31	26	14
(2) direct effect	69,526	69,498	41,293	35,081	0	86,491	86,456	40,104	45,324	0
d. near term practice (2% loss/year)										
(1) lifetime emissions (kg)	11	11	12	11	5	13	13	15	13	7
(2) direct effect	34,763	34,749	20,647	17,541	0	43,246	43,228	20,052	22,662	0
TEWI										
a. current practice										
(1) indirect effect (energy use)	2,137,700	2,137,700	2,200,000	2,266,400	2,096,252	3,335,900	3,259,000	3,433,700	3,214,800	3,253,100
(2) direct effect (emissions)	69,500	69,500	41,300	35,100	0	86,500	86,500	40,100	45,300	0
(3) TEWI	2,207,200	2,207,200	2,241,300	2,301,500	2,096,252	3,422,400	3,345,500	3,473,800	3,260,100	3,253,100
b. near term practice										
(1) indirect effect (energy use)	2,137,700	2,137,700	2,200,000	2,266,400	2,096,252	3,335,900	3,259,000	3,433,700	3,214,800	3,253,100
(2) direct effect (emissions)	34,800	34,700	20,600	17,500	0	43,200	43,200	20,100	22,700	0
(3) TEWI	2,172,500	2,172,400	2,220,600	2,283,900	2,096,252	3,379,100	3,302,200	3,453,800	3,327,500	3,253,100

Table 54. TEWI for low and medium temperature secondary loop refrigeration systems in Europe.

	Low Temperature Refrigeration					Medium Temperature Refrigeration				
Component of TEWI	R-404A	R-507	R-407A	R-407C	R-717	R-404A	R-507	R-134a	R-410A	R-717
1. Compressor Calculations										
a. design load (kW)	43.9	43.9	43.9	43.9	43.9	131.8	131.8	131.8	131.8	131.8
b. fraction on-time	40%	40%	40%	40%	40%	40%	40%	40%	40%	40%
c. cooling delivered (kWh/y)	153,999	153,999	153,999	153,999	153,999	461,998	461,998	461,998	461,998	461,998
d. pumping power (kWh/y)	6,044	6,044	6,044	6,044	6,044	17,345	17,345	17,345	17,345	17,345
e. total cooling required (kWh/y)	160,044	160,044	160,044	160,044	160,044	479,343	479,343	479,343	479,343	479,343
f. COP	1.603	1.603	1.554	1.505	1.638	3.117	3.199	3.018	3.248	3.205
g. compressor input (kWh/y)	99,812	99,812	102,964	106,321	97,718	153,807	149,863	158,822	147,593	149,556
2. Condenser Calculations										
a. refrigeration load (kWh/y)	160,044	160,044	160,044	160,044	160,044	479,343	479,343	479,343	479,343	479,343
b. pumping power (kWh/y)	6,044	6,044	6,044	6,044	6,044	17,345	17,345	17,345	17,345	17,345
c. compressor power (kWh/y)	99,812	99,812	102,964	106,321	97,718	153,807	149,863	158,822	147,593	149,556
d. total heat rejected (kWh/y)	265,900	265,900	269,052	272,409	263,806	650,495	646,551	655,510	644,280	646,244
e. required fan power (kWh/y)	4,866	4,866	4,924	4,985	4,828	11,904	11,832	11,996	11,790	11,826
3. Total Energy Consumption										
a. compressor (kWh/y)	99,812	99,812	102,964	106,321	97,718	153,807	149,863	158,822	147,593	149,556
b. secondary fluid pump (kWh/y)	6,044	6,044	6,044	6,044	6,044	17,345	17,345	17,345	17,345	17,345
c. condenser fan (kWh/y)	4,866	4,866	4,924	4,985	4,828	11,904	11,832	11,996	11,790	11,826
d. total annual energy (kWh/y)	110,722	110,722	113,932	117,351	108,590	183,056	179,040	188,163	176,728	178,728
e. lifetime energy (kWh)	1,660,832	1,660,832	1,708,976	1,760,261	1,628,848	2,745,838	2,685,599	2,822,447	2,650,916	2,680,913
f. lifetime CO_2 emissions (kg)	780,591	780,591	803,219	827,323	765,559	1,290,544	1,262,232	1,326,550	1,245,930	1,260,029
4. Refrigerant Emissions										
a. refrigerant GWP	3260	3300	1770	1530	0	3260	3300	1300	1730	0
b. refrigerant charge (kg)	18	18	19	19	9	22	22	26	22	11
c. current practice (4% loss/year)										
(1) lifetime emissions (kg)	11	11	12	11	5	13	13	15	13	7
(2) direct effect	34,763	34,749	20,647	17,541	0	43,246	43,228	20,052	22,662	0
d. near term practice (2% loss/year)										
(1) lifetime emissions (kg)	5	5	6	6	3	7	7	8	7	3
(2) direct effect	17,381	17,374	10,323	8,770	0	21,623	21,614	10,026	11,331	0
TEWI										
a. current practice										
(1) indirect effect (energy use)	780,600	780,600	803,200	827,300	765,600	1,290,500	1,262,200	1,326,500	1,245,900	1,260,000
(2) direct effect (emissions)	34,800	34,700	20,600	17,500	0	43,200	43,200	20,100	22,700	0
(3) TEWI	815,400	815,300	823,800	844,800	765,600	1,333,700	1,305,400	1,346,600	1,268,600	1,260,000
b. near term practice										
(1) indirect effect (energy use)	780,600	780,600	803,200	827,300	765,600	1,290,500	1,262,200	1,326,500	1,245,900	1,260,000
(2) direct effect (emissions)	17,400	17,400	10,300	8,800	0	21,600	21,600	10,000	11,300	0
(3) TEWI	798,000	798,000	813,500	836,100	765,600	1,312,100	1,283,800	1,336,500	1,257,200	1,260,000

Table 55. TEWI for low and medium temperature distributed refrigeration systems in Europe.

	Low Temperature Refrigeration				Medium Temperature Refrigeration			
Component of TEWI	R-404A	R-507	R-407A	R-407C	R-404A	R-507	R-134a	R-410A
1. Compressor Calculations								
a. design load (kW)	43.9	43.9	43.9	43.9	131.8	131.8	131.8	131.8
b. fraction on-time	40%	40%	40%	40%	40%	40%	40%	40%
c. cooling delivered (kWh/y)	153,999	153,999	153,999	153,999	461,998	461,998	461,998	461,998
d. pumping power (kWh/y)	2,873	2,873	2,873	2,873	8,585	8,585	8,585	8,585
e. total cooling required (kWh/y)	156,873	156,873	156,873	156,873	470,583	470,583	470,583	470,583
f. COP	1.692	1.692	1.640	1.588	3.219	3.304	3.118	3.355
g. compressor input (kWh/y)	92,739	92,739	95,667	98,787	146,177	142,429	150,944	140,271
2. Condenser Calculations								
a. refrigeration load (kWh/y)	156,873	156,873	156,873	156,873	470,583	470,583	470,583	470,583
b. pumping power (kWh/y)	2,873	2,873	2,873	2,873	8,585	8,585	8,585	8,585
c. compressor power (kWh/y)	92,739	92,739	95,667	98,787	146,177	142,429	150,944	140,271
d. total heat rejected (kWh/y)	252,485	252,485	255,413	258,533	625,345	621,597	630,112	619,439
e. required fan power (kWh/y)	4,620	4,620	4,674	4,731	11,444	11,375	11,531	11,336
3. Total Energy Consumption								
a. compressor (kWh/y)	92,739	92,739	95,667	98,787	146,177	142,429	150,944	140,271
b. secondary fluid pump (kWh/y)	2,873	2,873	2,873	2,873	8,585	8,585	8,585	8,585
c. condenser fan (kWh/y)	4,620	4,620	4,674	4,731	11,444	11,375	11,531	11,336
d. total annual energy (kWh/y)	100,232	100,232	103,214	106,391	166,206	162,389	171,060	160,192
e. lifetime energy (kWh)	1,503,485	1,503,485	1,548,217	1,595,867	2,493,087	2,435,836	2,565,895	2,402,873
f. lifetime CO_2 emissions (kg)	706,638	706,638	727,662	750,058	1,171,751	1,144,843	1,205,971	1,129,350
4. Refrigerant Emissions								
a. refrigerant GWP	3260	3300	1770	1530	3260	3300	1300	1730
b. refrigerant charge (kg)	40	40	44	43	50	50	58	50
c. current practice (5% loss/year)								
(1) lifetime emissions (kg)	30	30	33	33	38	37	44	37
(2) direct effect	98,758	98,718	58,655	49,831	122,857	122,807	56,966	64,381
d. near term practice (2% loss/year)								
(1) lifetime emissions (kg)	12	12	13	13	15	15	18	15
(2) direct effect	39,503	39,487	23,462	19,933	49,143	49,123	22,786	25,752
TEWI								
a. current practice								
(1) indirect effect (energy use)	706,600	706,600	727,700	750,100	1,171,800	1,144,800	1,206,000	1,129,400
(2) direct effect (emissions)	98,800	98,700	58,700	49,800	122,900	122,800	57,000	64,400
(3) TEWI	805,400	805,300	786,400	799,900	1,294,700	1,267,600	1,263,000	1,193,800
b. near term practice								
(1) indirect effect (energy use)	706,600	706,600	727,700	750,100	1,171,800	1,144,800	1,206,000	1,129,400
(2) direct effect (emissions)	39,500	39,500	23,500	19,900	49,100	49,100	22,800	25,800
(3) TEWI	746,100	746,100	751,200	770,000	1,220,900	1,93,900	1,228,800	1,155,200

Table 56. TEWI for low and medium temperature direct expansion refrigeration systems in Japan.

	Low Temperature Refrigeration				Medium Temperature Refrigeration			
Component of TEWI	R-404A	R-507	R-407A	R-407C	R-404A	R-507	R-134a	R-410A
1. Compressor Calculations								
a. design load (kW)	23.8	23.8	23.8	23.8	126.8	126.8	126.8	126.8
b. fraction on-time	40%	40%	40%	40%	40%	40%	40%	40%
c. cooling delivered (kWh/y)	83,519	83,519	83,519	83,519	444,442	444,442	444,442	444,442
d. pumping power (kWh/y)	0	0	0	0	0	0	0	0
e. total cooling required (kWh/y)	83,519	83,519	83,519	83,519	444,442	444,442	444,442	444,442
f. COP	1.762	1.762	1.708	1.654	3.425	3.515	3.317	3.569
g. compressor input (kWh/y)	47,399	47,399	48,896	50,490	129,774	126,446	134,005	124,530
2. Condenser Calculations								
a. refrigeration load (kWh/y)	83,519	83,519	83,519	83,519	444,442	444,442	444,442	444,442
b. pumping power (kWh/y)	0	0	0	0	0	0	0	0
c. compressor power (kWh/y)	47,399	47,399	48,896	50,490	129,774	126,446	134,005	124,530
d. total heat rejected (kWh/y)	130,918	130,918	132,415	134,009	574,216	570,888	578,448	568,973
e. required fan power (kWh/y)	2,396	2,396	2,423	2,452	10,508	10,447	10,586	10,412
3. Total Energy Consumption								
a. compressor (kWh/y)	47,399	47,399	48,896	50,490	129,774	126,446	134,005	124,530
b. secondary fluid pump (kWh/y)	0	0	0	0	0	0	0	0
c. condenser fan (kWh/y)	2,396	2,396	2,423	2,452	10,508	10,447	10,586	10,412
d. total annual energy (kWh/y)	49,795	49,795	51,319	52,943	140,282	136,893	144,591	134,942
e. lifetime energy (kWh)	746,923	746,923	769,786	794,140	2,104,227	2,053,401	2,168,865	2,024,137
f. lifetime CO_2 emissions (kg)	353,294	353,294	364,109	375,628	995,299	971,259	1,025,873	957,417
4. Refrigerant Emissions								
a. refrigerant GWP	3260	3300	1770	1530	3260	3300	1300	1730
b. refrigerant charge (kg)	64	63	70	69	240	237	279	237
c. current practice (13.5% loss/year)								
(1) lifetime emissions (kg)	129	128	141	139	486	480	565	480
(2) direct effect	421,303	421,133	250,222	212,581	1,585,462	1,584,822	735,139	830,831
d. near term practice (6% loss/year)								
(1) lifetime emissions (kg)	57	57	63	62	216	213	251	213
(2) direct effect	187,246	187,170	111,210	94,480	704,650	704,366	326,728	369,258
TEWI								
a. current practice								
(1) indirect effect (energy use)	353,300	353,300	364,100	375,600	995,300	971,300	1,025,900	957,400
(2) direct effect (emissions)	421,300	421,100	250,200	212,600	1,585,500	1,584,800	735,100	830,800
(3) TEWI	774,600	774,400	614,300	588,200	2,580,800	2,556,100	1,761,000	1,788,200
b. near term practice								
(1) indirect effect (energy use)	353,300	353,300	364,100	375,600	995,300	971,300	1,025,900	957,400
(2) direct effect (emissions)	187,200	187,200	111,200	94,500	704,600	704,400	326,700	369,300
(3) TEWI	540,500	540,500	475,300	470,100	1,699,900	1,675,700	1,352,600	1,326,700

Table 57. TEWI for low and medium temperature secondary loop refrigeration systems in Japan.

	Low Temperature Refrigeration					Medium Temperature Refrigeration				
Component of TEWI	R-404A	R-507	R-407A	R-407C	R-717	R-404A	R-507	R-134a	R-410A	R-717
1. Compressor Calculations										
a. design load (kW)	23.8	23.8	23.8	23.8	23.8	126.8	126.8	126.8	126.8	126.8
b. fraction on-time	40%	40%	40%	40%	40%	40%	40%	40%	40%	40%
c. cooling delivered (kWh/y)	83,519	83,519	83,519	83,519	83,519	444,442	444,442	444,442	444,442	444,442
d. pumping power (kWh/y)	2,540	2,540	2,540	2,540	2,540	4,205	4,205	4,205	4,205	4,205
e. total cooling required (kWh/y)	86,059	86,059	86,059	86,059	86,059	448,647	448,647	448,647	448,647	448,647
f. COP	1.603	1.603	1.554	1.505	1.638	3.117	3.199	3.018	3.248	3.205
g. compressor input (kWh/y)	53,671	53,671	55,366	57,171	52,545	143,958	140,266	148,652	138,141	139,979
2. Condenser Calculations										
a. refrigeration load (kWh/y)	86,059	86,059	86,059	86,059	86,059	448,647	448,647	448,647	448,647	448,647
b. pumping power (kWh/y)	2,540	2,540	2,540	2,540	2,540	4,205	4,205	4,205	4,205	4,205
c. compressor power (kWh/y)	53,671	53,671	55,366	57,171	52,545	143,958	140,266	148,652	138,141	139,979
d. total heat rejected (kWh/y)	142,271	142,271	143,966	145,771	141,145	596,810	593,118	601,504	590,993	592,831
e. required fan power (kWh/y)	2,604	2,604	2,635	2,668	2,583	10,922	10,854	11,008	10,815	10,849
3. Total Energy Consumption										
a. compressor (kWh/y)	53,671	53,671	55,366	57,171	52,545	143,958	140,266	148,652	138,141	139,979
b. secondary fluid pump (kWh/y)	2,540	2,540	2,540	2,540	2,540	4,205	4,205	4,205	4,205	4,205
c. condenser fan (kWh/y)	2,604	2,604	2,635	2,668	2,583	10,922	10,854	11,008	153,161	10,849
d. total annual energy (kWh/y)	58,815	58,815	60,541	62,380	57,669	159,084	155,325	163,864	2,297,417	155,033
e. lifetime energy (kWh)	882,227	882,227	908,116	935,693	865,029	2,386,260	2,329,879	2,457,963	1,086,678	2,325,493
f. lifetime CO_2 emissions (kg)	417,293	417,293	429,539	442,583	409,159	1,128,701	1,102,033	1,162,616		1,099,958
4. Refrigerant Emissions										
a. refrigerant GWP	3260	3300	1770	1530	0	3260	3300	1300	1730	0
b. refrigerant charge (kg)	7	7	8	8	4	26	26	31	26	14
c. current practice (4% loss/year)										
(1) lifetime emissions (kg)	4	4	5	5	2	16	16	18	16	8
(2) direct effect	13,731	13,726	8,155	6,929	0	51,674	51,653	23,960	27,079	0
d. near term practice										
(1) lifetime emissions (2% loss/year)	2	2	2	2	1	8	8	9	8	4
(2) direct effect	6,866	6,863	4,078	3,464	0	25,837	25,827	11,980	13,539	0
TEWI										
a. current practice										
(1) indirect effect (energy use)	417,300	417,300	429,500	442,600	409,200	1,128,700	1,102,000	1,162,600	1,086,700	1,100,000
(2) direct effect (emissions)	13,700	13,700	8,200	6,900	0	51,700	51,700	24,000	27,100	0
(3) TEWI	431,000	431,000	437,700	449,500	409,200	1,180,400	1,153,700	1,186,600	1,113,800	1,100,000
b. near term practice										
(1) indirect effect (energy use)	417,300	417,300	429,500	442,600	409,200	1,128,700	1,102,000	1,162,600	1,086,700	1,100,000
(2) direct effect (emissions)	6,900	6,900	4,100	3,500	0	25,800	25,800	12,000	13,500	0
(3) TEWI	424,200	424,200	433,600	446,100	409,200	1,154,500	1,127,800	1,174,600	1,100,200	1,100,000

Table 58. TEWI for low and medium temperature distributed refrigeration systems in Japan

Component of TEWI	Low Temperature Refrigeration				Medium Temperature Refrigeration			
	R-404A	R-507	R-407A	R-407C	R-404A	R-507	R-134a	R-410A
1. Compressor Calculations								
a. design load (kW)	23.8	23.8	23.8	23.8	126.8	126.8	126.8	126.8
b. fraction on-time	40%	40%	40%	40%	40%	40%	40%	40%
c. cooling delivered (kWh/y)	83,519	83,519	83,519	83,519	444,442	444,442	444,442	444,442
d. pumping power (kWh/y)	1,752	1,752	1,752	1,752	8,760	8,760	8,760	8,760
e. total cooling required (kWh/y)	85,271	85,271	85,271	85,271	453,202	453,202	453,202	453,202
f. COP	1.692	1.692	1.640	1.588	3.219	3.304	3.118	3.355
g. compressor input (kWh/y)	50,410	50,410	52,002	53,697	140,778	137,168	145,369	135,090
2. Condenser Calculations								
a. refrigeration load (kWh/y)	85,271	85,271	85,271	85,271	453,202	453,202	453,202	453,202
b. pumping power (kWh/y)	1,752	1,752	1,752	1,752	8,760	8,760	8,760	8,760
c. compressor power (kWh/y)	50,410	50,410	52,002	53,697	140,778	137,168	145,369	135,090
d. total heat rejected (kWh/y)	137,433	137,433	139,025	140,720	602,741	599,131	6607,331	597,052
e. required fan power (kWh/y)	2,515	2,515	2,544	2,575	11,030	10,964	11,114	10,926
3. Total Energy Consumption								
a. compressor (kWh/y)	50,410	50,410	52,002	53,697	140,778	137,168	145,369	135,090
b. secondary fluid pump (kWh/y)	1,752	1,752	1,752	1,752	8,760	8,760	8,760	8,760
c. condenser fan (kWh/y)	2,515	2,515	2,544	2,575	11,030	10,964	11,114	10,926
d. total annual energy (kWh/y)	54,677	54,677	56,298	58,025	160,568	156,893	165,243	154,776
e. lifetime energy (kWh)	820,151	820,151	844,467	870,368	2,408,525	2,353,389	2,478,644	2,321,644
f. lifetime CO_2 emissions (kg)	387,932	387,932	399,433	411,684	1,139,232	1,113,153	1,172,399	1,098,137
4. Refrigerant Emissions								
a. refrigerant GWP	3260	3300	1770	1530	3260	3300	1300	1730
b. refrigerant charge (kg)	16	16	17	17	60	59	70	59
c. current practice (5%/year loss)								
(1) lifetime emissions (kg)	12	12	13	13	45	44	52	44
(2) direct effect	39,009	38,994	23,169	19,683	146,802	146,743	68,068	76,929
d. near term practice (2%/year loss)								
(1) lifetime emissions (kg)	5	5	5	5	18	18	21	18
(2) direct effect	15,604	15,598	9,267	7,873	58,721	58,697	27,227	30,772
TEWI								
a. current practice								
(1) indirect effect (energy use)	387,900	387,900	399,400	411,700	1,139,200	1,113,200	1,172,400	1,098,100
(2) direct effect (emissions)	39,000	39,000	23,200	19,700	146,800	146,700	68,100	76,900
(3) TEWI	426,900	426,900	422,600	431,400	1,286,000	1,259,900	1,240,500	1,175,000
b. near term practice								
(1) indirect effect (energy use)	87,900	87,900	399,400	411,700	1,139,200	1,113,200	1,172,400	1,098,100
(2) direct effect (emissions)	15,6600	15,6600	9,300	7,900	58,700	58,700	27,200	30,800
(3) TEWI	403,500	403,500	408,700	419,600	1,197,900	1,171,900	1,199,600	1,128,900

APPENDIX J: AUTOMOBILE AIR-CONDITIONING REFRIGERANT EMISSION SCENARIOS

Three scenarios are considered for HFC-134a emissions from automobile air conditioners. The first uses an estimate for leakage from hoses, gaskets, and seals based on manufacturers data for the design and operation of new systems (Zietlow 1997, Hara 1996, Fernqvist 1997). This estimate of 35 g/y (1.23 oz) represents a lower limit on HFC-134a emissions and it is *very unlikely* that emissions would be lower than this. The second and third scenarios are based on 1½ and 2 service visits during the operating lifetime of the air conditioner requiring refrigerant additions of 40% of the original equipment charge (Baker 1996). These three scenarios are shown in Figs. 48 to 50. The horizontal axis in Fig. 48 has been extended to 22 years to show that three recharges would indeed be required under this scenario in twice the assumed system lifetime.

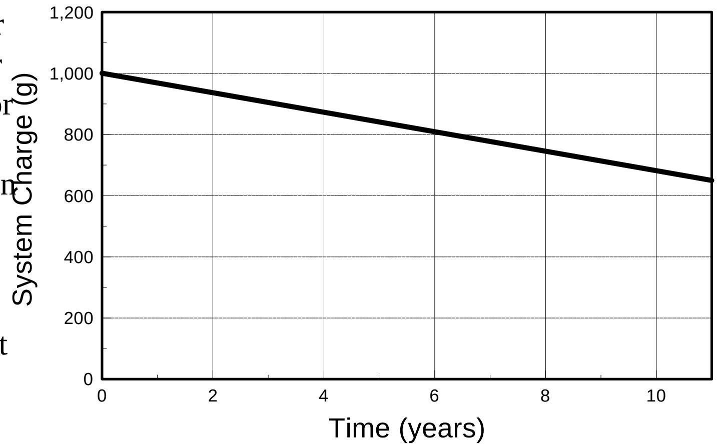

Figure 48. Air-conditioner charge with "as manufactured" emission rate of 35 g/year.

The lifetime system refrigerant emissions under these assumptions are summarized in Table 59. This shows lifetime losses of 415 g (14.6 oz) under the minimal emissions scenario, 860 g (30.3 oz) assuming 1½ recharges, and 1080 g (38.1 oz) assuming 2 recharges.

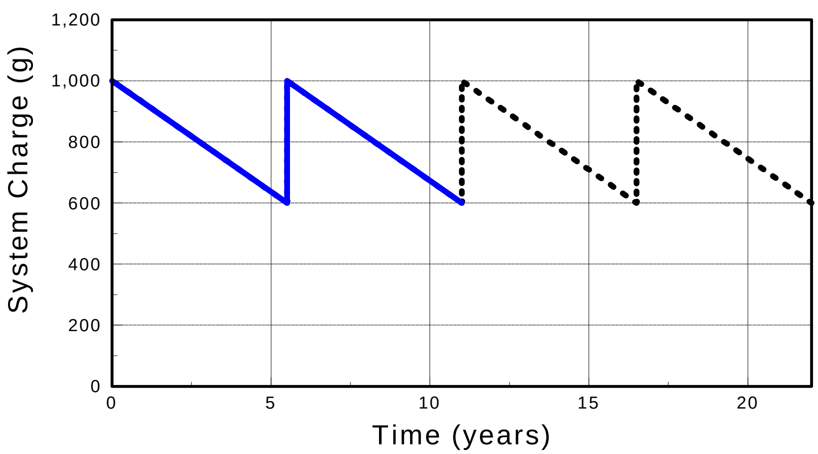

Figure 49. Air-conditioner charge with 1½ recharges during operational lifetime (3 recharges in 22 years).

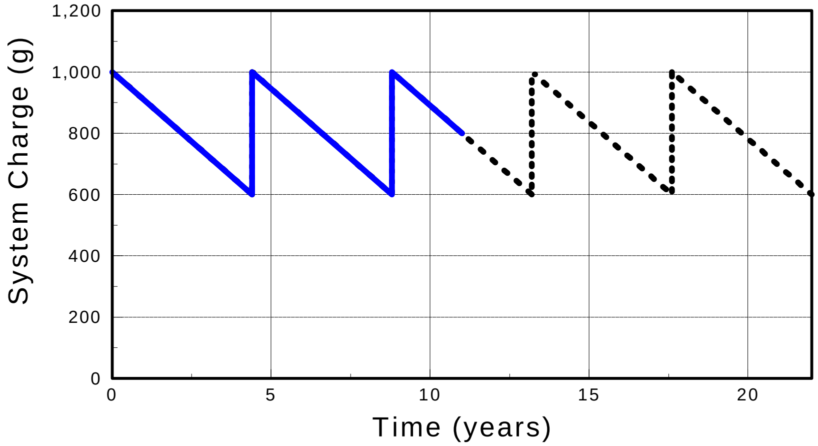

Figure 50. Air-conditioner charge with two recharges during operational lifetime (4 recharges in 22 years).

Table 59. Refrigerant emissions for automobile air conditioning (North America).

Refrigerant Requirement	"As Manufactured"	1½ Recharges	2 Recharges
Original Equipment Charge	1000 g	1000 g	1000 g
Service Additions	0 g	400 g	800 g
Total Refrigerant Usage	1000 g	1400 g	1800 g
End-of-Life Charge	650 g	600 g	800 g
End-of-Life Refrigerant Recovery	585 g	540 g	720 g
Net Lifetime Refrigerant Usage: Emissions (Total Usage - Recovery)	415 g	860 g	1080 g

APPENDIX K: AMBIENT TEMPERATURE DISTRIBUTIONS

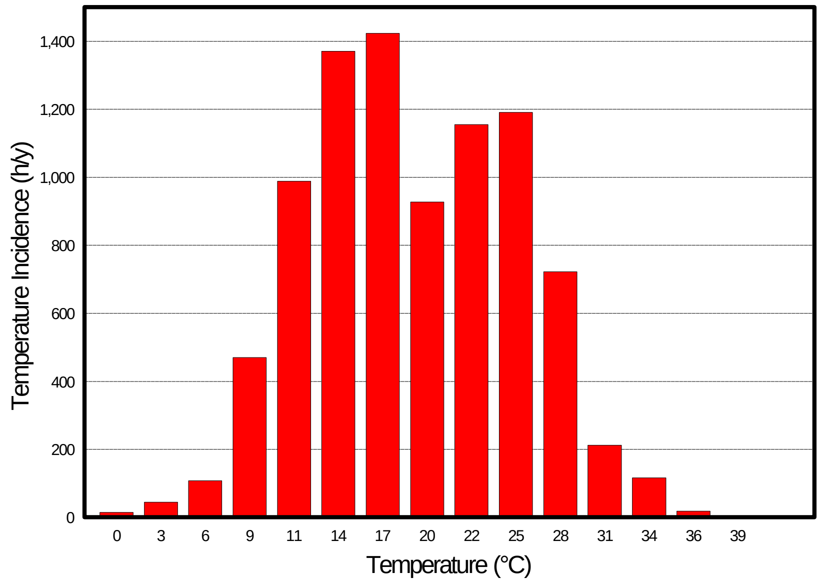

Figure 51. Outdoor ambient air temperature distribution for Greece (average).

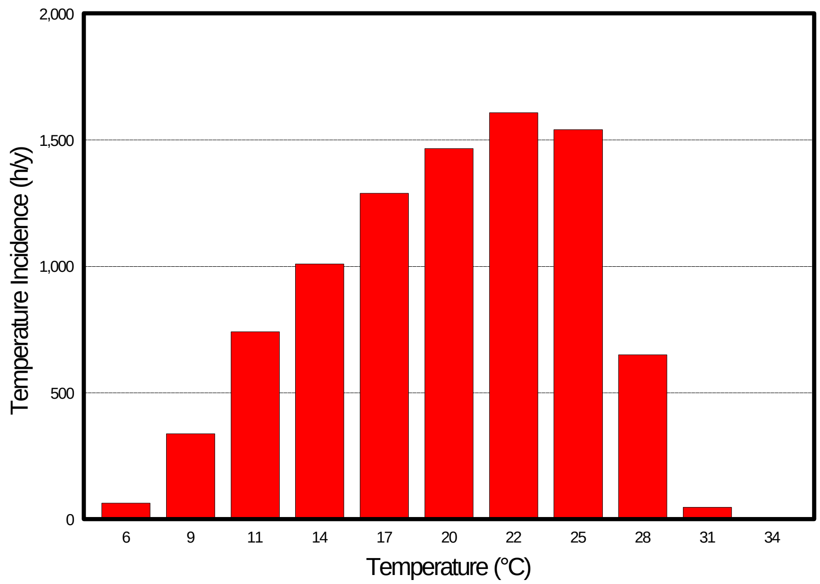

Figure 52. Outdoor ambient temperature distribution for Okinawa, Japan.

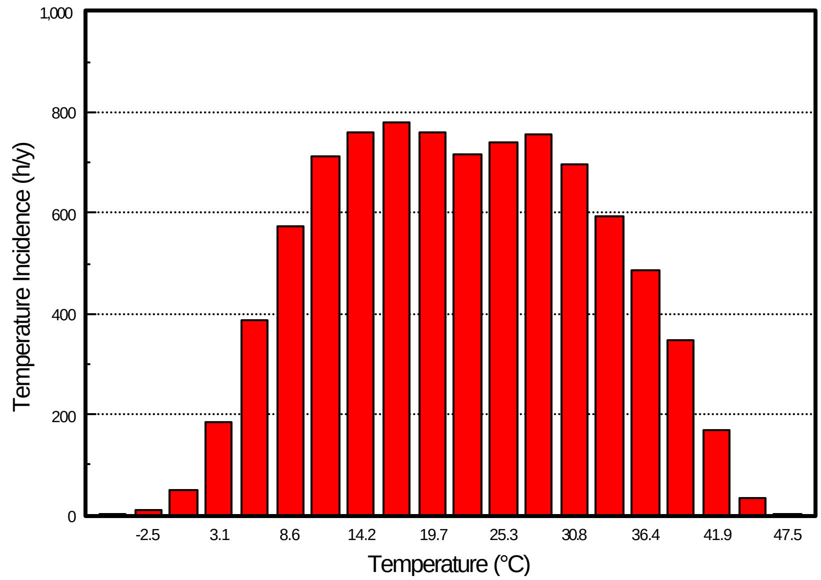

Figure 53. Outdoor ambient temperature distribution for the southwestern U.S.

APPENDIX L: TEWI FOR AUTOMOBILE AIR CONDITIONING

Table 60.. TEWI for automobile air conditioners in the U.S.

Air-Conditioning System	Midwest	Southeast	Northeast	Southwest
HFC-134a: as manufactured				
Direct Effect	540	540	540	540
Weight	363	363	363	363
Air Conditioner Energy	1088	3093	1405	2889
TEWI	1991	3996	2308	3792
HFC-134a: 1½ recharges				
Direct Effect	1118	1118	1118	1118
Weight	363	363	363	363
Air Conditioner Energy	1088	3093	1405	2889
TEWI	2569	4574	2886	4370
HFC-134a: 2 recharges				
Direct Effect	1404	1404	1404	1404
Weight	363	363	363	363
Air Conditioner Energy	1088	3093	1405	2889
TEWI	2855	4860	3172	4656
Hydrocarbon				
Direct Effect	4	4	4	4
Weight	491	491	491	491
Air Conditioner Energy	1598	4482	2062	4160
TEWI	2093	4977	2557	4655
CO_2: prototype				
Direct Effect	1	1	1	1
Weight	491	491	491	491
Air Conditioner Energy	1339	3798	1730	3759
TEWI	1831	4290	2221	4251
CO_2: equivalent air-side				
Direct Effect	1	1	1	1
Weight	491	491	491	491
Air Conditioner Energy	1773	5035	2291	4902
TEWI	2264	5526	2782	5393

Table 61. TEWI for automobile air conditioners in Europe.

Air-Conditioning System	United Kingdom	Germany	Greece	Italy	Spain
HFC-134a: as manufactured					
Direct Effect	540	540	540	540	540
Weight	245	245	245	245	245
Air Conditioner Energy	454	480	1255	927	974
TEWI	1239	1265	2040	1712	1759
HFC-134a: 1½ recharges					
Direct Effect	1118	1118	1118	1118	1118
Weight	245	245	245	245	245
Air Conditioner Energy	454	480	1255	927	974
TEWI	1817	1843	2618	2290	2337
HFC-134a: 2 recharges					
Direct Effect	1404	1404	1404	1404	1404
Weight	245	245	245	245	245
Air Conditioner Energy	454	480	1255	927	974
TEWI	2103	2129	2904	2576	2623
Hydrocarbon					
Direct Effect	4	4	4	4	4
Weight	331	331	331	331	331
Air Conditioner Energy	681	715	1838	1362	1431
TEWI	1016	1050	2173	1697	1766
CO_2: prototype					
Direct Effect	1	1	1	1	1
Weight	331	331	331	331	331
Air Conditioner Energy	576	601	1547	1152	1210
TEWI	908	933	1879	1484	1542
CO_2:equivalent air-side					
Direct Effect	1	1	1	1	1
Weight	331	331	331	331	331
Air Conditioner Energy	757	792	2048	1521	1598
TEWI	1089	1124	2380	1853	1930

Table 62. TEWI for automobile air conditioners in Japan.

Air-Conditioning System	Itazuke	Okinawa	Misawa	Yokota
HFC-134a: as manufactured				
Direct Effect	507	507	507	507
Weight	154	154	154	154
Air Conditioner Energy	610	972	355	521
TEWI	1271	1633	1016	1182
HFC-134a: 1½ recharges				
Direct Effect	783	783	783	783
Weight	154	154	154	154
Air Conditioner Energy	610	972	355	521
TEWI	1547	1909	1292	1458
HFC-134a: 2 recharges				
Direct Effect	983	983	983	983
Weight	154	154	154	154
Air Conditioner Energy	610	972	355	521
TEWI	1747	2109	1492	1658
Hydrocarbon				
Direct Effect	3	3	3	3
Weight	167	167	167	167
Air Conditioner Energy	888	1408	521	760
TEWI	1058	1578	691	930
CO_2: prototype				
Direct Effect	0	0	0	0
Weight	167	167	167	167
Air Conditioner Energy	719	1147	412	608
TEWI	886	1304	579	775
CO_2: equivalent air-side				
Direct Effect	0	0	0	0
Weight	167	167	167	167
Air Conditioner Energy	1065	1694	612	902
TEWI	1232	1861	780	1069

APPENDIX M: AUTOMOBILE AIR CONDITIONING (EQUATIONS)

1. Compressor power:

$$P_{comp}(T_{amb}) = \frac{\Delta H(T_{amb})}{COP(T_{amb})}$$

2. Energy output (cooling rate):

$$\dot{Q}(T_{amb},rpm) = \eta_{vol}(rpm) \cdot \rho \cdot D \cdot rpm \cdot \Delta H(T_{amb})$$

3. System COP:

$$COP_{sys}(T_{amb},rpm) = \frac{\dot{Q}(T_{amb},rpm)}{P_{comp} + P_{blower} + P_{pump}}$$

4. Hours of air conditioner operation:

$$Time_{sys}(T_{amb}) = \frac{Hours(T_{amb},city)}{8760\ hours/year} \cdot \frac{1210\ vehicle\ hours}{50{,}000\ miles} \cdot (\text{miles per year for the region}) \cdot (\text{\% compressor on-time})$$

5. Energy input (system power consumption):

$$P(rpm) = \sum_{T_{amb}} \frac{\dot{Q}(T_{amb},rpm) \cdot Time_{sys}(T_{amb})}{COP_{sys}(T_{amb},rpm)}$$

APPENDIX N: RESIDENTIAL GAS HEATING/COOLING TEWI RESULTS

Table 63. TEWI calculations for unitary space conditioning equipment: Pittsburgh, Pennsylvania USA.

Equipment Lifetime	15	years			
Building Description	1800 74,700,000 16,100,000	ft^2 Btu/y heating Btu/y cooling	Carbon Dioxide Conversions	0.0559 0.65 0.19	kg CO_2 / 1000 Btu gas input kg CO_2/kWh electricity kg CO_2 / 1000 Btu I^2R

Equipment Description	Refrigerant	Seasonal Efficiency		Indirect Effect			Direct Effect	TEWI
		Cooling	Heating	Electricity	Gas	Parasitics	Refrigerant	
1996 Equipment	refrigerant emissions: make-up rate: 4% of charge per year, end-of-life loss: 15% of charge							
Central A/C and Gas Furnace	HCFC-22 HCFC-22	SEER - 10 SEER - 12	80% flue 92% flue	15,698 13,081	78,295 68,083	0 0	3570 3570	97,562 84,734
Premium Options High Efficiency A/C and Gas Furnace Gas Engine Driven	HCFC-22 HCFC-22	SEER - 14 gCOP-1.30	92% flue gCOP-1.26	11,213 0	68,083 60,080	0 8010	3570 3570	82,866 71,660
2005 Equipment	refrigerant emissions: make-up rate: 2% of charge per year, end-of-life loss: 15% of charge							
Premium Heat Pump Options Gas Engine Driven GAX Absorption Heat Pump	HCFC-22 R-717	gCOP-1.30 gCOP - 1.0	gCOP-1.26 gCOP- 1.5	0 0	60,080 55,274	8,010 15,578	2142 0	70,233 70,851

Table 64. TEWI calculations for unitary space conditioning equipment: Atlanta, Georgia USA.

Equipment Lifetime	15	years			
Building Description	1800	ft^2	Carbon Dioxide Conversions	0.0559	kg CO_2 / 1000 Btu gas input
	34,800,000	Btu/y heating		0.65	kg CO_2/kWh electricity
	33,800,000	Btu/y cooling		0.19	kg CO_2 / 1000 Btu I^2R

		Seasonal Efficiency		Indirect Effect			Direct Effect	
Equipment Description	Refrigerant	Cooling	Heating	Electricity	Gas	Parasitics	Refrigerant	TEWI
1996 Equipment	refrigerant emissions: make-up rate: 4% of charge per year, end-of-life loss: 15% of charge							
Central A/C and Gas Furnace	HCFC-22	SEER - 10	80% flue	32,955	36,475	0	3570	73,000
	HCFC-22	SEER - 12	92% flue	27,463	31,717	0	3570	62,750
Central A/C and Electric Furnace	HCFC-22	SEER - 12	100%	27,463	0	99,440	3570	130,476
Premium Options								
High Efficiency A/C and Gas Furnace	HCFC-22	SEER - 14	92% flue	23,539	31,717	0	3570	58,826
Gas Engine Driven	HCFC-22	gCOP - 1.30	gCOP-1.26	0	44,973	7,780	3570	56,323
2005 Equipment	refrigerant emissions: make-up rate: 2% of charge per year, end-of-life loss: 15% of charge							
Premium Options								
Gas Engine Driven	HCFC-22	gCOP - 1.30	gCOP-1.26	0	44,973	7,780	2142	54,895
GAX Absorption Heat Pump	R-717	gCOP - 1.0	gCOP- 1.5	0	47,809	13,668	0	61,476

Table 65. TEWI for European residential space heating (data from the *International Heat Pump Status and Policy Review Report*, PHC-AR3, IEA Heat Pump Centre, September 1994, Part 1.

Equipment Specification:			Refrigerant Emission Data:		CO_2 Conversion Factors:		
Lifetime	15	years	Make-up rate	5% of charge per year	Electricity Generation	0.469	kg/kWh
Heat pump capacity	10	kW	End-of-Life Loss	15% of charge	Natural Gas Combustion	0.190	kg/kWh
Annual heating load	16,400	kW/year					

Technology	Efficiency (SPF)	Electricity			Natural Gas		Refrigerant Emissions						TEWI (kg CO_2)
		Heating (kWh/y)	Auxiliary (kWh/y)	Lifetime CO_2 (kg)	Heating (kWh/y)	Lifetime CO_2 (kg)	Charge (kg)	Leakage (g/y)	End-of-Life (g)	Lifetime Loss (kg)	Refrigerant GWP (kg CO_2/kg)	Direct Effect (kg CO_2)	
Electric Resistance Heat	1.00	16,400	0	115,374	0	0	0	0	0	0	0	0	115,374
<u>Gas Boiler or Furnace</u>	0.80 0.92	0 0	0 0	0 0	4203 3654	58,548 50,911	0 0	0 0	0 0	0 0	0 0	0 0	58,548 50,911
Thermal Sorption	1.27	0	895	6299	2647	36881	0	0	0	0	0	0	43,180