IEA Annex 26: Advanced Supermarket Refrigeration / Heat Recovery Systems – Canada Report

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Executive Summary

CANMET Energy Technology Centre– Varennes (CETC-Varennes) contribution to IEA Annex 26 included firstly, a supermarket showcase with a fully integrated HVAC/Refrigeration system using secondary fluid for the condenser and evaporator sides, secondly, the organization of an advisory committee, thirdly, a workshop for the Canadian refrigeration industry, and finally, two research projects directly related to refrigeration in supermarkets.

Canada's refrigeration sector is facing several major challenges such as: legislation for the replacement of synthetic refrigerants that have negative environmental impacts; increasing cost of refrigerants; poor efficiency of equipment that is not adapted to Canadian climatic conditions; new regulations for refrigerated product storage and transportation; and a lack of innovation and adequate expertise in Canada to address these issues.

There are approximately 31,000 retail food stores in Canada of which 7,150 supermarkets are considered large – with a floor cover area of over 1,000 m^2. Supermarkets are among the most energy intensive buildings in the commercial sector with a total energy consumption of more than 800 kWh-equivalent/m^2/year (increasing with time). Refrigeration represents about 50% of the electrical energy consumption of a supermarket, costing on average $150,000 CND per year for a typical large size supermarket (2000 m^2). The typical refrigeration system contains approximately 750 kg of synthetic refrigerant equivalent to 2,000 tonnes of CO_2 in terms of greenhouse gas effect. Refrigerant losses are on average, 25% per year of the total charge.

CETC-Varennes signed an agreement with a major Canadian supermarket chain to develop a supermarket technology showcase in Montreal area in the end of 2003. The project's objective is to develop innovative integrated HVAC&R technologies and practices suitable for Canadian climatic conditions, using compact and hermetic refrigeration systems with secondary fluid for the evaporator and condenser sides, and to demonstrate them in partnership with a major supermarket chain in Canada. These demonstrations will allow a better understanding of these technologies technical requirements such as design, installation, commissioning, and performance as well as their non-technical barriers (knowledge transfer to energy consultants and technical operators, costs, procurements, etc.)

The selected measures should result in reductions of operating and maintenance costs, refrigerant leakage, and indirectly, food spoilage. Secondary fluid systems will contain less refrigerant and fewer potential refrigerant leak locations, resulting in a substantial reduction in greenhouse gas emissions. This project takes part in a technology transfer and know-how dissemination strategy.

CETC-Varennes organised the meetings of the Industry Advisory Panel, consisting in supermarket and refrigeration sectors representatives. The objective of this committee is to provide information on the Canadian supermarket sector needs in order to develop case studies, laboratory experiments or demonstration projects. The committee gathers representatives from retailer companies, compressor rack manufacturers, display case manufacturers, supermarket refrigeration control manufacturers, consulting engineers and government agencies. The Industry Advisory Panel participants are aware and recognise IEA Annex 26 strategy, which aims at increasing the energy efficiency/heat recovery in refrigeration systems for supermarkets, while reducing the overall refrigerant charge and potential leaks.

More than 85 people attended the first Canadian Workshop on Refrigeration Systems for Supermarkets on March 11, 2003 in Montreal. The event was sponsored by Natural Resources Canada's CANMET Energy Technology Centre – Varennes (CETC-Varennes) and the Office of Energy Efficiency (OEE), in collaboration with Agence de l'efficacité énergétique (AEE), Hydro-Quebec, Manitoba Hydro and BC Hydro. With a possible reduction of 1.4 Mt equivalent CO_2/year by 2010, refrigeration in supermarkets has a great potential of energy efficiency and greenhouse gas emission reductions. The refrigeration industry (equipment manufacturers, food retailers, design engineers, mechanical contractors, and utilities) sees this as an opportunity. The event allowed the identification of proven and emerging refrigeration technologies and the validation of the supermarkets' interests. In addition, six discussion groups shared ideas on how to lift barriers for adoption of the energy efficient refrigeration technologies. The information collected will provide CETC-Varennes and OEE with new ways to facilitate the adoption of the technologies developed.

A laboratory test bench was developed to test novel methods that provide refrigeration to the food retail industry. In the first series of tests, CETC-Varennes is expected to develop the knowledge to operate refrigeration systems at minimal head pressure and maintain controllability of refrigeration system using an electronic expansion valve. Results from this work will validate or further improve the accuracy of refrigeration equipment models and will help to transfer the know-how to refrigeration experts. Some technical papers will be published soon.

CETC-Varennes also started a new strategic project. The main objective is to construct and perform experiments on a liquid CO_2 secondary fluid loop test bench. The secondary objective is the development of basic know-how (technical documentation, simulation models, experimental data) and demonstration of the technologies feasibility to potential partners. CETC-Varennes has a team of researchers working on the modeling of complex problems related to refrigeration technologies, using tools such as a CFD (Computational Fluid Dynamic) software. A major partner from the industrial gas industry has joined the research team to carry out this project.

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1 INTRODUCTION

Canada's refrigeration sector is facing several major challenges such as: legislation for the replacement of synthetic refrigerants that have negative environmental impacts; increasing cost of refrigerants; poor efficiency of equipment that is not adapted to Canadian climatic conditions; new regulations for refrigerated product storage and transportation; and a lack of innovation and adequate expertise in Canada to address these issues.

There are approximately 31,000 retail food stores in Canada of which 7,150[1] supermarkets are considered large – with a floor cover area of over 1,000 m^2. There are approximately 100 to 125 of these supermarkets that are built every year by large companies. Supermarkets are among the most energy intensive buildings in the commercial sector with an energy consumption of more than 800 kWh/m^2/year (increasing with time). The refrigeration system is the crucial part of the supermarket, ensuring the conservation of frozen and perishable food. Refrigeration represents about 50% of the electrical energy consumption of a supermarket, costing on average $150,000 CND per year for a typical large size supermarket (2000 m^2). The typical refrigeration system contains approximately 750 kg of synthetic refrigerant equivalent to 2,000[2] tons of CO_2 in terms of greenhouse gas effect. Refrigerant losses are on average, 25%[3] per year of the total charge.

A study[4] carried out by CANMET Energy Technology Centre–Varennes (CETC-Varennes) revealed that the adoption of environmentally sound HVAC&R strategies in a supermarket will result in significant reductions for the building: 88% of the total refrigerant charge, 18% of the overall energy consumption and 76% of CO_2 equivalent emissions.

Therefore, there is an opportunity to provide Canadian supermarkets with advanced HVAC&R integrated technologies and practices to enable them to reduce their greenhouse gas emissions associated with energy consumption and synthetic refrigerant leaks. The supermarkets will benefit mainly from energy savings and synthetic refrigerant cost reduction. Other benefits from these technologies are increased equipment reliability and life expectancy, and reduced maintenance costs. The potential impact of a concerted effort to deploy these new refrigeration technologies in Canadian supermarkets has been evaluated by CETC-Varennes as reducing greenhouse gas emissions by 1.4 Mt eq. CO_2 and energy consumption by 2,900 GWh per year by 2010.

CETC-Varennes' contribution to IEA Annex 26 included firstly, a supermarket showcase with a fully integrated HVAC/Refrigeration system using secondary fluid for the condenser and evaporator sides, secondly, the organization of an advisory committee, thirdly, a workshop for the Canadian refrigeration industry, and finally, two research projects directly related to refrigeration in supermarkets. This report presents these activities.

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[1] Refrigerated Display Cabinets, Market Assessment – standardization – Energy Use Impact, Epsilon Technologies Ltd to Canadian Standard Association, March 1994

[2] Canada's greenhouse gas inventory: 1997 emissions and removals with trends, Environment Canada, April 1999 (1kg of the synthetic refrigerant R404a is equivalent to 3,300 kg of CO_2 in terms of greenhouse gas emissions)

[3] Van D. Baxter - Oak Ridge National Laboratory –Communication at IEA Heat Pump Program -Annex 26 - Advanced Supermarket Refrigeration/Heat Recovery Systems

[4] Système avancé de réfrigération pour les supermarchés, LRDEC 2001-101(CF), 2 octobre 2001

2 ADVANCED REFRIGERATION TECHNOLOGIES SHOWCASE

This demonstration project will advance the development and the demonstration of innovative, efficient and environmentally friendly integrated heating, ventilation, air-conditioning, and refrigeration (HVAC&R) technologies in Canadian supermarkets. The project is championed by CETC-Varennes and brings together private and public sector stakeholders including a major Canadian supermarket chain, the provincial utility, specialists in energy efficiency consulting, equipment and controls suppliers, federal and provincial organizations.

Over the next three years, the public funds will leverage private investments to:

- Develop and adapt innovative and environmentally sound HVAC&R systems for supermarkets in Canada
- Install and commission these novel HVAC&R systems in a Canadian supermarket and monitor their performance
- Develop training material and provide training on innovative HVAC&R systems for supermarkets
- Provide an in-depth understanding of technical and non-technical barriers in implementing these technologies
- Build the capacity of Canadian industry to implement these new technologies

This project aims at reducing the total energy consumption of a typical supermarket by approximately 25%. The greenhouse gas emissions resulting from these energy savings and from the reduction of refrigerant leaks will be reduced by more than 50%.

2.1 Innovation Strategy

The overall goal of the project is to develop innovative integrated HVAC&R technologies and practices suitable for Canadian climatic conditions, using compact and hermetic refrigeration systems, and to demonstrate them in partnership with a major supermarket chain in Canada. These demonstrations will allow a better understanding of these technologies technical requirements such as design, installation, commissioning, and performance as well as their non-technical barriers (knowledge transfer to energy consultants and technical operators, costs, procurements, etc.)

The technical and environmental performance of HVAC&R systems in Canadian supermarkets is hindered by the lack of expertise and technology innovation in Canada. For instance:

- The current HVAC&R equipment is designed for the southern U.S. climatic conditions. Therefore, it is not designed to benefit from the cold conditions of the Canadian climate.
- Supermarkets are positive energy balance buildings. For a typical supermarket, the refrigeration

systems reject 5,000,000 kWh/year [1] while the domestic hot water and space heating load is 1,500,000 kWh/year. Therefore, the annual heat released by the refrigeration system is three times higher than the heat required for the building heating and the hot water consumption of a typical supermarket. Even in very cold climatic conditions, the refrigeration system releases enough heat to meet the heating load of the supermarket. The major part of this reject heat is presently released to the environment – without recovery measures.

- Supermarkets heating and cooling are distributed by the circulation of synthetic refrigerant throughout the whole supermarket building to the display cases and heat exchangers. This design requires huge synthetic refrigerant charges and an extensive piping network which release an important refrigerant quantity to the environment through leakages.

Based on these current HVAC&R technology practices, this project will develop and demonstrate strategies to reduce energy consumption and GHG emissions in Canadian supermarkets by:

- Adapting current HVAC&R systems to derive maximum benefits from Canadian climatic conditions
- Integrating HVAC&R systems more effectively
- Reducing synthetic refrigerant charge and leaks

2.2 Technical Barriers:

- Develop innovative integrated HVAC&R systems and practices that meet supermarkets refrigeration and space heating needs while ensuring low energy consumption, low synthetic refrigerant charges and leaks, low energy costs and indoor comfort conditions
- Develop a control system to ensure good performance and reliability of HVAC&R systems
- Monitor the HVAC&R system to optimize the overall energy efficiency
- Work with supermarket staff to evaluate indoor comfort conditions and equipment maintenance issues

2.3 Non-technical Barriers:

- Increase confidence in the proposed innovative solutions by showcasing them and providing tools to facilitate replication
- By demonstrating the initial prototype system and by providing stakeholders with design tools and know-how, this project will enable replication for other Canadian supermarket chains[2].

[1] Système avancé de réfrigération pour les supermarchés, LRDEC 2001-101(CF), 2 octobre 2001

[2] One of the major supermarket chain owns more than 600 supermarkets in Canada, with several more in various stages of development. These HVAC&R systems have the potential of being adapted in future retrofit activities at these and other stores.

2.4 Major Deliverables

- A feasibility study for the implementation of innovative HVACV&R technologies and practices in a supermarket
- A supermarket (technology showcase) equipped with innovative technologies and practices
- Training material and training for innovative HVAC&R system operators and managers to ensure good operation and maintenance of the systems
- A stakeholder awareness information package with fact sheets that will:
    - Familiarize refrigeration stakeholders with the proposed technologies and practices
    - Demonstrate the technical and economic viability of the technologies and practices implemented
    - Design tools for consultants that will facilitate technology replication in other supermarkets

3 INCENTIVE PROGRAM FOR RETAIL FOOD STORES

Natural Resources Canada's Commercial Building Incentive Program (CBIP) offers a financial incentive for the incorporation of energy efficiency features in new commercial/institutional building designs. The objective of this incentive is to encourage energy-efficient design practices and to bring about lasting changes in the Canadian building design and construction industry. A financial incentive of up to $60,000 will be awarded to building owners whose designs meet CBIP requirements. The program requirements are based on two documents: the Model National Energy Code for Buildings (MNECB) and CBIP Technical Guide. An eligible building design must demonstrate a reduction in energy use by at least 25% when compared to the requirements of the MNECB.

Retail food stores are now included in the list of building types that are eligible for a CBIP incentive. The interaction between the building's environment and the refrigeration equipment cannot be ignored in retail food stores. Therefore, CBIP will take into account the impact of the refrigeration system on the building's performance when evaluating these two building types.

A new tool has been developed: the EE Wizard will be used to show compliance to CBIP for retail food. The Technical Guide for Retail Food Stores provides information on how to use the EE Wizard.

http://oee.nrcan.gc.ca/newbuildings/cbip.cfm

3.1 Supermarket Charette

During 2001, Canadian supermarket stakeholders have discussed measures to reduce energy consumption. The two meetings held in Quebec City and Toronto have demonstrated the supermarket sector’s strong interest to improve their installations energy efficiency. These meetings were organised under the Commercial Building Incentive Program (CBIP).

Measures discussed during the supermarket charette:

HVAC & Mechanical

- Humidification control - refrigeration & desiccant systems
- Heat pumps versus roof top units
- Renewable systems - photovoltaics on roof, solar wall heating
- Heat recovery opportunities - grey water, exhaust air
- Building pressurization control to reduce infiltration

Refrigeration System

- Centralized versus distributed compressors - improving compressor COP
- Natural and mechanical subcooling
- Refrigeration heat recovery - sources & sinks discussion
- Head pressure control - effect of reducing head pressure, floating head pressure

Control

- Reducing defrost - improving defrost control
- Effect of covering display case
- Scheduling auxiliary equipment (lighting, anti-sweat heaters)
- Display case efficiencies

4 INDUSTRY ADVISORY PANEL

CETC-Varennes has organised an Industry Advisory Panel composed of supermarket and refrigeration sectors representatives. The objective of this Panel is to advise on case studies, laboratory experiments and proposed demonstration projects based on Canadian supermarket sector needs.

Representatives from various areas of the Canadian supermarket sector participated in the two meetings at CETC-Varennes in 2000. During these meetings, the Panel discussed: refrigerant safety and policy issues, short term industry needs for refrigeration, new refrigeration technologies in America and Europe, various problems and issues for the supermarket sector.

Some subjects arose during the meeting were:

- Monitoring of refrigeration installations to confirm the estimated benefits
- The actual status of international R&D for refrigeration equipment and systems
- Life Cycle Cost concept as opposed to the lower first cost approach
- The benefits of detailed commissioning refrigeration installations
- Using secondary fluid with refrigeration equipment

5 WORKSHOP ON SUPERMARKETS REFRIGERATION[3]

A workshop on refrigeration systems for supermarkets was held in Montreal on March 11, 2003, under two Natural Resources Canada's organizations stewardship: the Office of Energy Efficiency (OEE) and the CANMET Energy Technology Centre – Varennes (CETC-Varennes). The objective was to define the means and the actions to develop in order to ensure a greater diffusion of technologies available, which are characterized by a notable improvement of energy efficiency and a considerable reduction of the environmental impacts. The participants in this workshop focused their discussions on the comparison between:

- Conventional multiplex central refrigeration systems with a refrigerant distribution network to display cases, and
- Central refrigeration systems with a secondary refrigerant (glycol, potassium formate, ice slurries, CO_2…) distribution network to display cases with a heat recovery use for space heating, from the refrigeration system. The whole as an integrated system approach to satisfy the energy demand of the building. The optimization of the operating pressures of the refrigerating systems was also addressed.

5.1 Objective

OEE and CETC-Varennes had laid down the following objectives:

- To make the supermarket sector aware of the environmental impacts of traditional refrigeration technologies
- To present the federal government objectives to reduce the environmental impacts of greenhouse gas emissions

[3] *Based on the Workshop Report* written by Yves Blanc and Andre Chalifour March 28, 2003

- To present available alternative technologies, their profitability and the results of their application in specific cases
- To give the opportunity to food distributors, experts and professionals of the refrigeration sector to discuss on the barriers related to the diffusion of these alternative technologies and on the ways to remove them

5.2 Participation

More than 85 people took part in this event. The number and the quality of the attendance show the relevance of the subject and the interest. The participants represented the scientific, technical and commercial sectors (supermarkets chains, engineering consultant, refrigeration contractor, mechanics, technicians, plumbers, manufacturers, scientific laboratories, electric utilities, universities). They came from all over Canada.

Distribution of the participants according to industry's sectors

Food Distributors	**10%**
Equipment Manufacturers	**24%**
Contractors	**11%**
Consultants	**17%**
Utilities	**13%**
Universities	**2%**
Governments Representatives	**9%**
Sponsor Representatives	**14%**

5.3 Conferences

OEE and CETC-Varennes recalled the objectives of the federal government in regarding reduction of greenhouse gas (GHG) emissions. CETC-Varennes stressed that the synthetic refrigerant leakage of the Canadian supermarkets corresponded to the emission of 3.6 megatons of GHG, which is 60% of total GHG emissions of the supermarket sector. A market penetration of 40% of secondary fluid loop technologies in new supermarkets and in major retrofits represents a reduction of GHG emissions of 1.1 MT.

CETC-Varennes described the refrigeration technologies used in several foreign countries; and explained why the CETC-Varennes privileged the systems with secondary loop integrating space heating that use heat recovery from refrigeration systems heat rejection.

Hydro-Quebec (electric Utility) exposed the technological concepts of heat recovery applied to refrigeration developed by refrigeration equipment manufacturers RSD and LMP. Energy savings were monitored and certified by Hydro-Quebec's Laboratory of technologies of the energy (LTE). The tested heat recovery systems completely eliminated the need for auxiliary heating. An innovating defrost process reduced by 85% the average duration of display case defrost cycles and decreased the losses of perishable products (meats).

Loblaw, a major supermarket chain, exposed its orientations regarding energy efficiency and presented its strategy for secondary refrigerant systems. It briefly described the showcase project under development in Quebec that will integrate these technologies.

Hill Phoenix, major manufacturer of refrigeration equipment, exposed the benefits obtained with secondary refrigerant systems. It insisted, specially, on reduction of the operating costs (elimination of possibilities of synthetic refrigerant leaks, significant energy savings, less moving parts and thus considerable reduction of the maintenance costs). He also underlined the great temperature stability (considerable reduction in fresh product losses and better appearance of the meat) and the greater comfort for the customers in the store (reduction of the diffusion of the cold over the floors in the alleys of the store). Another advantage of the secondary refrigerant systems is the easier operation of display cases defrosting and the reduction of the defrost cycle time for this operation.

5.4 Debates

5.4.1 Synopsis of the discussions

The debates made possible to establish the degree of interest of the sector for these technologies as well as the barriers to a broader diffusion of them.

The participants identified the following barriers:

- Ignorance of the technologies and their advantages. Recommendations: conferences, project controls in various areas and under various climates, tools for decision-making)
- Training needs for engineering consultants and fitters. Recommendations: software for economic analysis and feasibility studies, design guidelines
- The fragmentation of the market (refrigeration mechanics, plumbers, air handling systems installers, consultants…). Recommendation: Project controls showing ways of effective interactions of trade associations
- Dubious profitability. Recommendation: financial incentives to reduce the initial investment as long as the scales of production will not be sufficient to make it possible for the manufacturers to offer competitive prices compared to conventional systems.

Three significant remarks

1. The importance of continuity in the governmental policies regarding energy efficiency and environment was stressed. Customers adopted innovating technologies in the past considering that they were adequately prepared for the future. However, these recent technologies seem to be replaced by new concepts. They fear having wasted time and money in concepts without future. It is necessary to avoid this kind of situation, because the sector will not buy any more to new ideas for fear of beginning in a dead end.

2. A working group stressed that the refrigerant regulations were and will continue to be one of the most powerful incentive for the adoption of effective and nonaggressive technologies for the environment. In addition, the sector has difficulties to follow regulation changes. It should be

remembered that the choice of a refrigerant is a commitment for 20 to 25 years.

3. Some participants put forth the assumption that the customer is not aware that some stores can be "green" and that others have potentially harmful equipment for the environment. Supermarket chains do not feel pressure on behalf of their customers. Sensitizing the final customers and defining a "green" labeling for supermarkets could be ways to support the diffusion of the secondary refrigerant systems.

Method of discussion

To facilitate the debates, a guide was prepared. This guide was based on a report/ratio entitled "Market Transformation" developed by Natural Resources Canada with the assistance of the firm Navigant Consulting Inc. It comprises the analysis of the barriers to the penetration of new technologies in a market according to the method from the "five A". This approach highlights the following stages of diffusion of a new technology: availability, awareness, accessibility, affordability and acceptance.

The conclusions of this discussion should not be regarded as the result of a statistical survey. For as much, one should not minimize the importance of the opinions presented, because they are to some extent " the enlightened opinions" of the sector's representatives.

While recognizing their values in terms of energy and ecological effectiveness, they neglected, for lack of time, to pay a more attention to recover the energy from the refrigeration system heat rejections. This subject should be the subject of a workshop.

The comments summarized below thus treat mainly "application of the secondary refrigerant systems to supermarkets". The participants concentrated their discussions on the comparison between the secondary refrigerant systems and the conventional multiplex systems with direct expansion or "DX" [4].

Availability

- Technology is well known in Europe and there are more than 150 facilities in the USA. The equipment is available on the Canadian market, because American manufacturers normally supply them
- The customers have a preference for known and tested solutions, using less capital expenditure. They rather make their decision on the analysis of the "simple pay back " than on the "life cycle cost" approach
- It will thus be necessary to pay more attention to awareness and to accessibility to transform the market

Awareness

- The sector is not aware of the existence of the secondary refrigerant systems technology, in spite of the fact that they know it exists in the market
- According to participants, information is insufficient or non-existent. The number of demonstration projects is too small. It would be important to carry out other pilot projects through Canada and in particular under a large variety of climatic conditions.

[4] "DX" = abbreviation of "Direct eXpansion"

- All the participants insisted on the need for having a communication plan to raise awareness of these technologies. This plan must address all sectors and all levels, form the top management of supermarket chains to fitters, while passing by the supermarket departments of projects.
- Noting that the leaders of supermarket chains forsake the life cycle cost economic analyses of the projects on behalf of a simple evaluation of the initial capital costs. Several participants proposed to put in place a communication plan dealing with the economic value of technologies targeting the Presidents and Vice-presidents of finances of the large supermarket chains.

Accessibility

- Barriers at the fitters level slow down accessibility to secondary refrigerant systems technology. The principal problem is the fragmentation of this industry between "refrigeration technicians", air handling people and "plumbers". The second problem is the lack of experience and training of the contractors regarding this technology. It is not a matter of lack of scientific or technical training, but of practical experience. The fitters underlined also the absence of tools in the market (standards, guides of installation and exploitation).
- The engineering consultants, for their part, do not have yet enough experience nor design guidelines to design these systems efficiently. The time allocated for the preparation and the establishment of a project (new constructions or major retrofits) was shortened considerably. The completion date of a project was calculated formerly in terms of months; now, it is measured in terms of weeks. The customers thus do not give enough time to the engineering consultants to compare technologies or to seek innovative solutions. To meet the needs, it is necessary to be able to conceive a project with computation charts, design software and drawings preconceived, which make this process fast and reliable.

Affordability

- Engineering consultants and contractors do not propose the concept at competitive prices because they are not sufficiently familiar with it. However, those, which already offer the concept, recognize that the prices will drop with the increase in the number of installations. This technology thus represents currently, for the customer, a higher capital cost than that of the conventional multiplex systems. This is a significant barrier, even if, taking into account the significant economies in the exploitation costs, the secondary refrigerant systems technology offers a "net present value" (NPV) positive on the system's " life cycle cost". The demonstration projects and the economic tools for analysis are essential to overcome this barrier. Information and training are also essential means of diffusing these tools.

Acceptance

- These technologies will be accepted on a great scale when they meet the following criteria:
  - Their profitability is shown in real cases
  - They become the major trend in refrigeration
  - Their reliability is proven
  - Manufacturers, engineering consultants and contractors have adapted the concepts and are able to effectively put them in work and in a harmony of thought and execution
  - One is able to maintain and exploit normally these new systems
  - The time of construction is not affected

- The customers, the project leaders of new supermarkets or major retrofits consider that the technology of the secondary refrigerant systems with heat recovery has all the attributes of a competitive and credible solution.

6 R&D ACTIVITIES

The CETC-Varennes project team consists in five research scientists, two refrigeration engineers, a senior refrigeration technician, a commercial advisor along with technical and project management support staff. The Canadian Program of Energy Research and Development (PERD) will be a primary funding source for IEA Annex 26 project realization.

CETC-Varennes has started a new research project, named *New Practices in Refrigeration*. The first activity is to set up a laboratory test bench in order to test novel methods of providing refrigeration for the food marketing industry. In the first series of tests, CETC-Varennes expects to develop the knowledge to operate a refrigeration system at minimal condensing pressure and maintain controllability of refrigeration system using an electronic expansion valve. Results from this work will validate or further improve the accuracy of refrigeration equipment models and will help transfer the know-how to refrigeration experts.

A second strategic project is undertaken to design and build a test bench to develop the know-how for carbon dioxide use as a secondary refrigerant in a low temperature commercial refrigeration system. The main objective is to perform experiments on a liquid CO_2 secondary fluid loop test bench. The secondary objectives are the development of basic know-how (technical documentation, simulation models, and experimental data) and the demonstration of the technology feasibility to partners.

7 CONCLUSION

Canadian GHG emissions represent more than 6 Mt eq. CO_2 . Refrigerant leak to atmosphere represents 60% of the total GHG emission. The strategy to reduce GHG emissions is twofold: reduction of refrigerant charge and leaks and integration of space heating and refrigeration systems in order to recover heat from refrigeration systems. The potential impact of a concerted effort to deploy these new refrigeration technologies in Canadian supermarkets has been evaluated by CETC-Varennes as the reduction of greenhouse gas emissions by 1.4 Mt eq. CO_2 and the reduction of energy consumption by 2,900 GWh per year by 2010.

The participation of CETC-Varennes in IEA Annex 26 allowed the development of multiple activities to inform the supermarket refrigeration sector in Canada. The activities are: establishment of a strategy to minimize environmental impacts of the commercial refrigeration, the set up of a supermarket showcase, a workshop, an advisory committee, the participation in the development CBIP financial incentive program and R&D projects in commercial refrigeration. These research projects will allow knowledge development in order implement new refrigeration technologies that will be adapted to Canadian climate and needs, while reducing the environmental impacts of the commercial refrigeration.

IEA Heat Pump Program Annex 26

Advanced Supermarket Refrigeration/Heat recovery Systems

Country Report from Canada (Hydro Quebec - LTE)

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1. Introduction

Supermarkets are one of the most energy-intensive types of commercial buildings, because significant energy is used to maintain chilled and frozen food in display cases and storage rooms [**1**]. The major contributor to electric energy use and demand is the refrigeration system (around 50 %), for which compressors and condensers account for 60-70 %. The remainder is consumed by the display and freezing room fans, display case lighting, and for anti-sweat heaters. The majority of conventional supermarkets in Canada take partial advantage of the large amount of heat rejected by using this heat for space or water heating. Generally, a conventional heat reclaim coil, located in the central air handler, allows the recovery of between approximately 30 % and 50 % of total heat rejection, notably by de-superheating discharge hot gases. However, this amount of heat reclaim is not sufficient to completely eliminate the use of fossil fuels (natural gas, propane) during the coldest periods of the year. The first objective of the earlier developments in Canada was to increase the capability of conventional, multiplexed systems to satisfy the store's entire heating load, eliminate fossil fuels as energy sources, and thereby save energy and money. The "*System-1 - phase1*" (known as *RSD*) improved system includes an integral heat reclaim method [**2**], and introduces an original high velocity defrosting method [**3**] in order to improve food quality and lower perishable food losses. The "*System-2 - phase1*" (also known as *LMP*) advanced system [**4**] includes other original heat reclaim methods involving multifunctional cascade heat pumps capable of satisfying all store heating requirements during the winter. The following specific developmental steps aim to further reduce refrigerant quantities and premium costs compared with conventional systems. Effectively, today's world-wide priority for the low-TEWI supermarket is to meet environmental challenges, while concerns about heat reclaim and energy savings, although always important in cold climates, are somewhat secondary [**5**]. Several advanced supermarket refrigeration/heat reclaim systems have been developed, installed and tested, notably in Europe. They include new low-charge multiplex systems, secondary-loop refrigeration and distributed compressor systems [**6**]. Analyses show that the secondary fluid loops can reduce refrigerant charge in multiplex systems by approximately 35 %, and integrated water-source heat pumps could potentially reduce combined operating costs for refrigeration and HVAC by over 10 %. If properly designed and implemented, these advanced systems could reduce annual total equivalent warming impact (TEWI) by as much as 60 % [**7**]**.** The main objective of the LTE contribution to the IEA Annex 26 project was to analyse and experimentally document the benefits of the two above-mentioned improved supermarket multiplexed refrigeration and heat reclaim systems developed in Canada since 1999. The more specific scope of the project was to measure the energy usage for low temperature (frozen food), medium-temperature (chilled food) and total store refrigeration, the main operating parameters of the systems and their annual energy performances. An eventual goal was to further develop these initially improved options in order to reduce the total equivalent warming impact (TEWI) of supermarkets in cold climates by reducing the total refrigeration charge and leakage risks. LTE Laboratory worked with three retailers and two system manufacturers to field test these systems. Three new supermarkets were extensively instrumented and monitored over a 12-month period in order to compare a number of operating parameters, high-efficiency features such as heat reclaim, defrosting, sub-cooling and liquid pumping, and to quantify energy performance (compressors and store energy consumption and power demand, etc.). Two of the stores monitored, having approximately the same size and energy usage (basically, electric refrigeration and natural gas cooking) were each equipped with the improved refrigeration systems, and the third was a conventional store.

2. IMPROVED MULTIPLEX SYSTEMS

2.1 "System1-phase1"

The new supermarket chosen as the "test store" (*METRO – Messier*), located in the north area of Montreal, consists of 38,000 ft^2 total area (28,302 ft^2 sales area), and an average traffic of about 10,000 people per week. The store was equipped in 1999 with a "*System 1-phase 1*" improved multiplex refrigeration system, with a premium cost (equipment, installation, refrigerant charge, controls) of about $55,000 CD (1999). The main objective of this system was to recover enough energy to fully satisfy the store's heating requirements in a typical cold climate, and also to improve quality and reduce excessive losses of foodstuffs. **Figure 2.1** represents a schematic layout of the experimental store having 196,750 Btu/h as a low temperature (- 15 °F to –35 °F) refrigeration load, and 901,514 Btu/h as a medium temperature (15 °F to 45 °F) refrigeration requirement (**Table 2.1**). The display cases (tub, multi-deck and glass type) provide temporary storage for food prior to sale (**Appendix**). Five semi-hermetic reciprocating and three screw compressors with HCFC-22 respectively are used on low and medium temperature racks (about 104 kW of total electrical power). Located in a machine room on the mezzanine in the rear of the store, these compressors operate under common suction conditions, with refrigeration efficiency ranging from 2.3 and 1.3 at their saturated suction temperatures. Multiplexed parallel compressors have the advantage of continuously matching capacity with refrigeration loads, can operate at higher suction pressures and, at the same time, at lower head pressures. When no integral heat reclaim is called for, heat rejection is accomplished via remote air-cooled condensers located on the roof. The HVAC system for the store consists of a central air handler equipped with a heat reclaim coil, a natural gas-fired heater and a direct expansion coil located in the air handler for space cooling and dehumidification. An energy management system ("*Micro-Thermo*") controlled the refrigeration strategy, mainly by monitoring the compressor suction pressure and selecting display case and walk-in box return air temperatures.

Heat Reclaim Method

The patented heat reclaim method proposed by the "*System 1-phase 1*" [2] has been integrated into a baseline multiplexed refrigeration system consisting of parallel, semi-open compressors, remote condensers, evaporators in display cases, liquid receivers, manifolds and controls (**Figure 2.2**). As in a conventional heat reclaim system, there are bi-directional network valves (NV) that connect the superheated vapour to the remote condensers, or to the heat reclaim coils (HR), for extracting heat from the refrigeration system for space and domestic hot water heating, when required,. The heat reclaim coils are installed in series, and each of them contains 3-way solenoid and check valves for air temperature control.

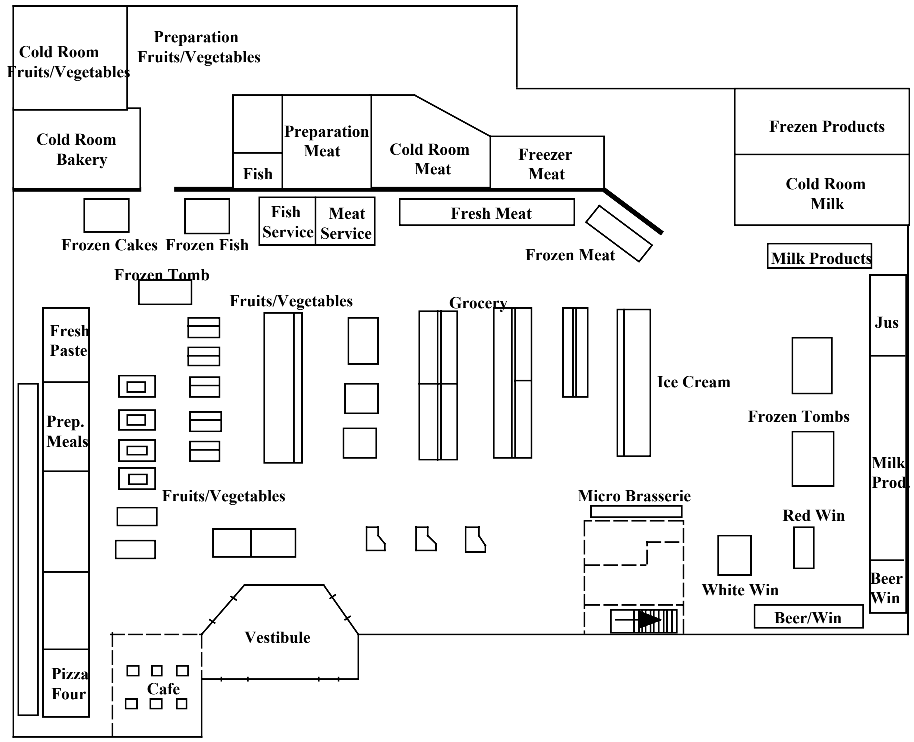

Figure 2.1 Layout of the Supermarket Equipped with "*System 1-phase1*"

Table 2.1 *System1-phase1*": Capacities as a Function of SST (SST - Saturated Suction Temperatures)

Total Area	Sale Area	Low/Dual Temperatures				Medium Temperatures				
-	-	-35 °F	-25 °F	-20 °F	-15 °F	10 °F	15 °F	20 °F	30 °F	45 °F
ft^2	ft^2	Btu/h	Btu/h	Btu/h	Btu/h	Btu/h	Btu/h	Btu/h	Btu/h	Btu/h
38 003	28 302	24 400	96 700	29 150	46 500	-	282 400	458 710	39 520	120 884

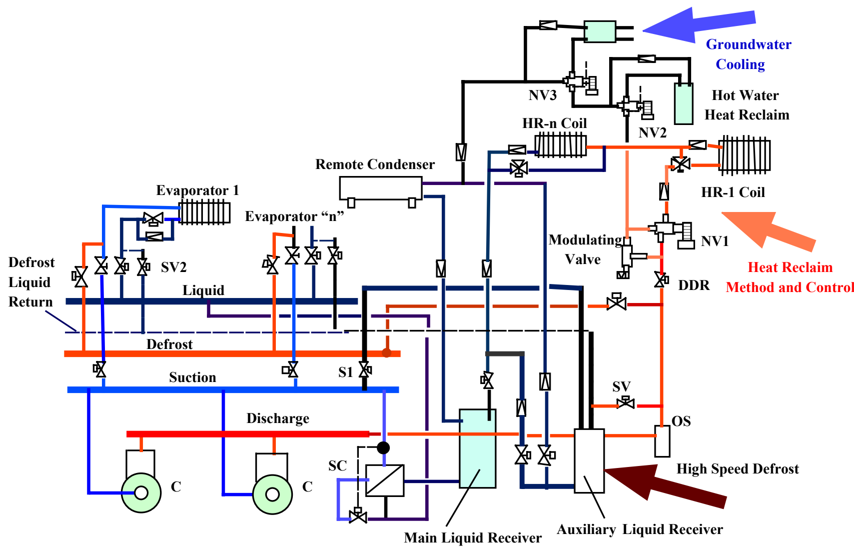

Figure 2.2 "*System 1-phase 1*": Heat Reclaim, High-speed Defrost and Groundwater Cooling
C – compressor; D – discharge; S – suction; DD – defrost discharge; DLR – defrost liquid return; LR – Liquid Receiver; ALR – auxiliary liquid receiver; SC – sub-cooler; S – solenoid valve

A modulating valve installed in parallel with the NV1 valve, automatically adjusts the discharge pressure of the compressors to different ambient temperatures, and re-directs a sufficient quantity of superheated vapour to the heat reclaim coils. This motorized valve, provided with an outside temperature sensor, modulates the refrigerant flow by a simple, economical and efficient method. Depending on outside temperature and the requirements of the refrigeration system, it can bypass some or all of the superheated vapour to the condensers, or during the winter, can direct part or all of the hot gas to the heat reclaim coils in order to achieve up to 100 % heat reclaim. The last heat reclaim coil and the remote condensers are separately connected to the main liquid receiver where the two liquid streams flow at approximately the same pressure. This represents the second system innovation allowing the heat reclaim coils to work at the same pressure as the condensers, and thus enabling partial or total condensing in order to maximize the heat recovery capability. Finally, prior being fed to the evaporators, the liquid is further sub-cooled by an internal heat exchange coil (SC) connected from the outlet of the main liquid receiver to the liquid manifold.

High-speed Defrost

The major disadvantage of conventional defrosting methods is their fairly lengthy cycles caused by a low-pressure differential across the evaporators. That results in adverse effects on foodstuff (losses and lower quality of perishable products) and compressors (increased energy costs, overheating, reduced technical life). The "*System 1-phase 1*" has developed a high-speed defrost concept capable of performing defrost cycles in a short period of time [**3**]. When a display case goes into a defrost cycle, superheated vapour is fed to the outlet of the evaporator coil. However, the return line is not connected to the remote condensers as in a conventional system, but to an auxiliary liquid receiver (**Figure 2.2**) The auxiliary receiver is linked up to the low-pressure header (S) by way of a solenoid valve (S1). When a defrost cycle is needed, this valve opens and a high pressure differential is created across the display case evaporator within a range of about 30 psi to up to 100 psi, thus achieving a very quick defrost cycle. In order to keep the main receiver supplied with sufficient quantities of liquid, it is necessary to periodically flush the auxiliary receiver. When the refrigerant temperature is below a predetermined value - normally 30 °F - a pressure differential of about 30 psi is created, and the liquid is directed to the main receiver. If the liquid is above 30 °F, it is sent to the remote condensers to finish the condensing process, and then back into the main receiver. The auxiliary receiver is provided with a level detector with an acoustic signal, so that when the liquid reaches a predetermined level, the "flushing mode" will operate during the next refrigeration cycle.

Geothermal Cooling

The "*System1-phase1*" also employed a geothermal cooling approach on both the low (LT) and medium (MT) temperature refrigeration lines. It is well known that in conventional systems, during the hottest periods of the year, the condensing heat is rejected to the outdoor air at high head pressure and temperature, which involves a high energy consumption. However, this excess heat can be rejected to the ground instead of to the ambient air in order to reduce energy consumption and thus optimize the overall efficiency of the refrigeration system. In fact, this concept allows a reduction in the compressor's high head pressures and condensing temperatures, because the cooling fluid is the groundwater at an average temperature of about 8°C (46.4 °F) instead of the ambient air at 35°C (95 °F) during the summer. Energy savings are possible by shutting down one or even two compressors on each refrigeration line. The test system contains groundwater supply and discharge wells, refrigerant-to-water heat exchangers on both low and medium temperature discharge lines, an automatic degasifying device and controls for cleaning (**Figure 2.3**). The main disadvantage of such a cooling method is the risk of groundwater heat exchangers clogging, which could involve additional costs for periodic cleanings. However, the energy savings may justify the additional investment and maintenance costs.

Figure 2.3 - "*System 1-phase 1*": View of the Groundwater Cooling Section

Test Approach

The first commercial prototype of the "System 1-phase 1" was thoroughly instrumented in order to asses its operating and energy performances. Thirty-six parameters were monitored (temperatures, pressures, electrical power, operating modes). The temperatures were measured with type-T thermocouples, electric power – by using OSI watts transducers, and pressures – with calibrated SENSOTEC amplified transducers. Scanned four times per minute and then saved every two minutes, these parameters permitted the calculation of several thermodynamic, energy and instantaneous and average performance parameters (daily, monthly and annual) (**Appendix**). A software system, equipped with a HP-75000 acquisition station and modem (**Figure 2.4**) permitted the continuous collection and daily transfer of data to the LTE mainframe computer where they were processed using special computer programs.

Figure 2.4 "*System 1-phase 1*": LTE's Data Acquisition and Transmission Panel

2.2 “System2-phase1”

A second ”test supermarket” (*IGA – Crevier*), built in 1999 and having a total area of 40,844 ft^2 (of which 30,838 ft^2 was for sales), also located in north area of Montreal (Figure 2.5) was equipped with a second improved refrigeration system (“*System2 – phase1*”). The central refrigeration room consisted of a low/dual temperature line (from -15 °F to –35 °F) with nine unequal semi-hermetic reciprocating compressors (49 kW) and a medium temperature rack (from 10 °F to 30 °F) with four semi-hermetic reciprocating compressors (25 kW). The store refrigeration load includes 238,714 Btu/h of low temperature cases and walk-in boxes, and 977,080 Btu/h of medium temperature loads (Table 2.2). Four refrigerant-to-refrigerant heat pumps (49 kW of nominal installed power) provided heat reclaim, air-conditioning, dehumidifying and sub-cooling capability. The HVAC system is a distributed type, the cooling and heating coils being located inside each heat pump cabinet, close to the conditioned spaces. The supplementary cost for the new equipment, including installation and controls for this first commercial prototype amounted 45,000 CD $ (1999). The patented “*System2 – phase1*”, contains four new elements compared with a conventional multiplex system (**Figure 2.6**) [**4**]:

(i) Four intermediate plate heat exchangers (plate HEXi=1,2,3 and plate HEX-4) on both dual (DT) and medium temperature (MT) discharge lines. These heat exchangers are for the de-superheating and partial or complete condensing of discharge hot gases.

(ii) Three refrigerant-to-refrigerant heat pumps for heat reclaim/air conditioning/dehumidifying purposes (HP-A, B and C).

(iii) A fourth heat pump (HP-S) for heat reclaim and liquid sub-cooling.

(iv) One liquid pumping station on each dual (low/medium - DT) and medium temperature (MT) rack (LP unit).

It should be noted that these components can just as easily be integrated into a new refrigeration system as they can into a retrofitted supermarket, because the assembly is completely independent, and the baseline refrigerating piping of the conventional system hasn’t basically been modified at all.

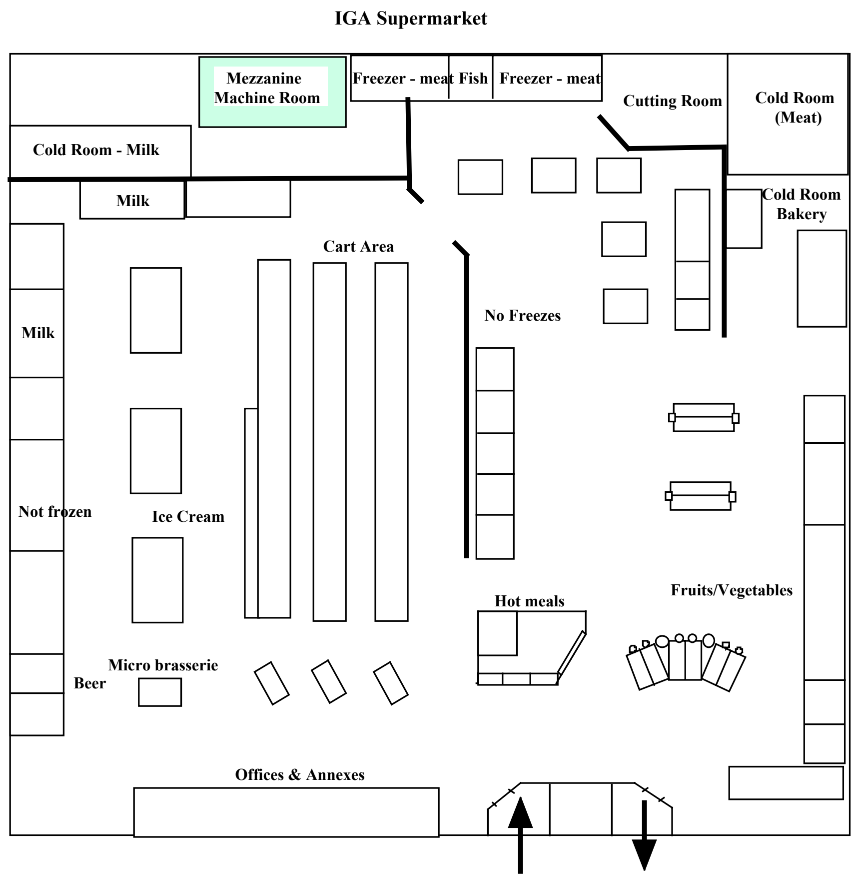

Figure 2.5 "*System2 – phase1*": Schematic Layout of the Test Supermarket

Table 2.2 "*System2 – phase1*": Refrigeration Capacities as a Function of the SST (SST - Saturated Suction Temperature)

Total Area	**Sale Area**	**Low/Dual Rack**		**Temperatures**		**Medium Temperatures Rack**				
-	-	-35 °F	-25 °F	-20 °F	-15 °F	10 °F	15 °F	20 °F	30 °F	45 °F
ft^2	ft^2	Btu/h	Btu/h	Btu/h	Btu/h	Btu/h	Btu/h	Btu/h	Btu/h	Btu/h
40 844	30 838	15 720	30 780	192 214	-	12 320	402 610	529 750	32 400	-

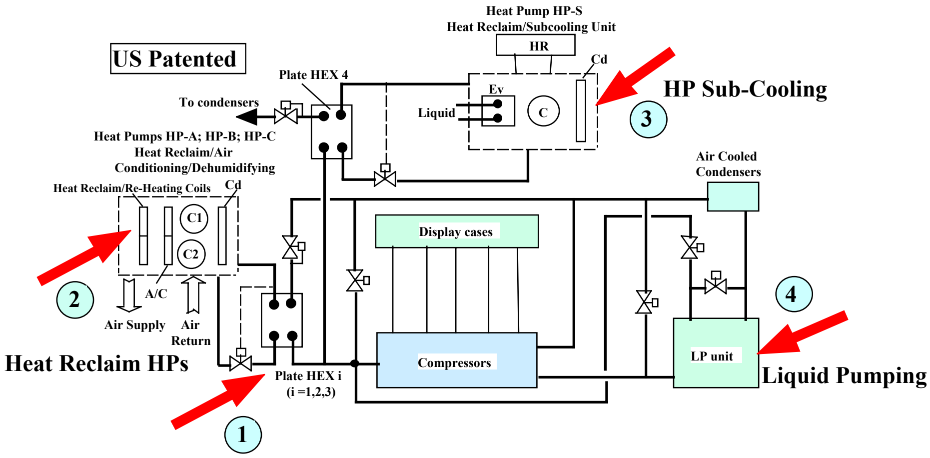

Figure 2.6 "*System2 – phase1*": Plate Heat Exchangers (1), Heat Reclaim/Sub-Cooling Heat Pumps (2; 3), and Liquid Pumping (4)

C – compressor ; HEX – intermediate heat exchanger; LD – liquid delivery unit; Cd – condenser; SC – sub-cooling heat exchanger; A/C – air conditioning coil; HR – heat reclaim coil.

Heat Reclaim Method

In conventional supermarket refrigeration systems, the condensing pressures and temperatures are determined by the outdoor air temperature. However, during the cold periods of the year, even if there is a possibility of reducing the condensing pressure, a "too high" head pressure is artificially maintained in order to permit proper operation of the expansion valves and of the hot gas defrost. As previously noted, the "*System2–phase1*" has integrated four new components into a multiplexed system, in order to provide an original method of maximizing the extraction of the condensing heat. This concept uses cascade refrigerant-to-air heat pumps so as to not raise the compressor's head pressures excessively during the cold periods of the year. At the same time, this system uses the same heat pumps for air conditioning/dehumidifying and sub-cooling purposes, and profits from the well-known efficiency of liquid pumping technology [**4**].

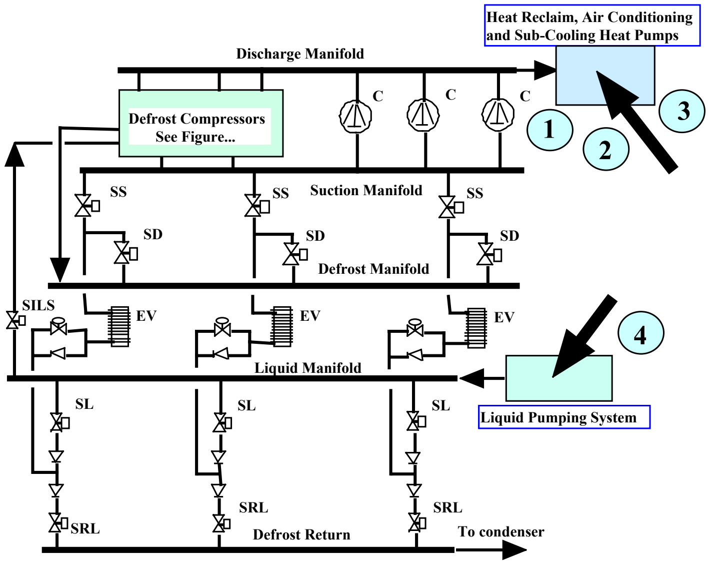

Figure 2.7 "*System2 – phase1*": Layout of the Improved Refrigeration System
1 – Plate Heat Exchanger; 2 – Heat Reclaim Heat Pumps; 3 – Sub-Cooling Heat Pump; 4 – Liquid Pumping Unit.

Operation

When heat reclaim is required, the plate heat exchangers HEX (A, B, C or S) and their respective refrigerant-to-refrigerant heat pumps are activated (**Figure 2.8 and 2.10**). Two in-tandem heat exchangers are connected to the medium temperature (MT) discharge line, while another two are installed on the dual temperature (DT) discharge line. Each of these heat exchangers contains three internal circuits, two for each of the heat pump compressors, and another for the system discharge superheated vapour. Sensible and, if necessary, latent heat are removed from this superheated vapour, which is then fed to the remote condensers. Entering pressure regulators (EPR) located after each group of intermediate heat exchangers, keep the discharge pressure at a minimum 125 psig (70 °F condensing temperature) during the winter. Heat extraction from the superheated vapour is done without excessively increasing the discharge pressure, thus allowing a reduction of the compressor's energy consumption. It is also possible to extract all the condensing heat without affecting the normal operation of the main refrigeration system. The liquid from the remote condensers returns to the liquid delivery station where its pressure is further increased. All the known benefits of the liquid delivery system are present during the heat reclaim mode.

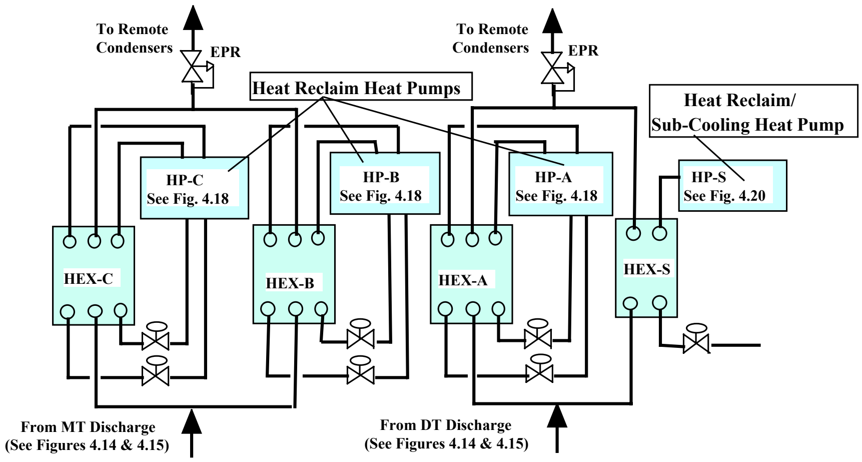

Figure 2.8 "*System2 – phase1*": Intermediate Heat Exchangers and Heat Pumps

Figure 2.9 "*System2 – phase1*": View of the Intermediate Heat Exchangers Section

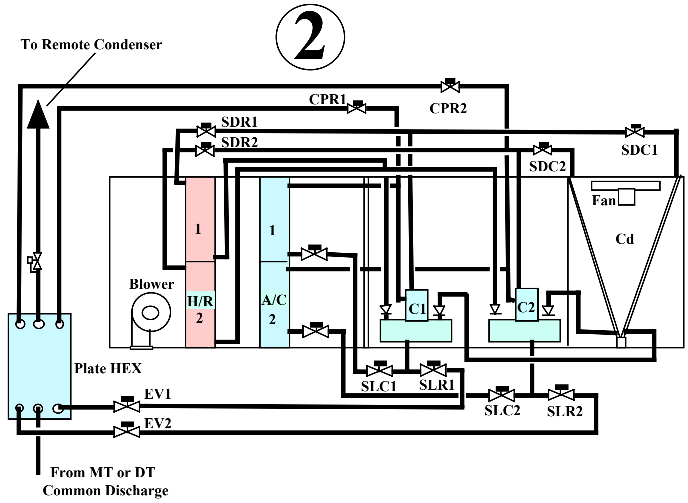

Figure 2.10 "*System2 – phase1*": Heat Reclaim & Air Conditioning Heat Pumps (HP-A, B, and C)
C – compressor; H/R – heat reclaim/dehumidifying coil; A/C – air conditioning coil; Cd – condenser; HEX – heat exchanger; DT – dual temperature; MT – medium temperature

Each of the heat pumps A, B and C (**Figure 2.11**) has two HCFC-22 compressors (C1 & C2), each with its own liquid receiver, refrigerant-to-air coils for heating/dehumidifying (H/R) and air conditioning (A/C), solenoid valves and pressure control devices, and a common air-cooled condenser with fan (Cd). When a heat pump calls for heat reclaim, the intermediate plate heat exchanger operates as an evaporator and supplies low pressure vapour to the compressors C1 and/or C2. The regulating valves CPR maintain the suction pressure of the compressors C1 and C2 at a constant value corresponding to an evaporating temperature of about 50 °F. The compressed refrigerant vapour is then fed to the heat reclaim coils (HR) through the solenoid valves SDR, and the condensed refrigerant returns to the liquid receivers, from which it comes back through the solenoid valves SLR and the expansion valves EV to the intermediate heat exchanger where it evaporates by absorbing heat from the superheated refrigerant vapour. In this mode, the condenser Cd and the air conditioning coils A/C are not operational, and are separated from the heat reclaim circuits by closing the solenoid valves SDC and SLC. In air conditioning mode, the low-pressure refrigerant vapour from the air conditioning evaporators A/C comes into the suction of the compressors, is compressed, and is then fed to the condenser Cd. The liquid returns to the receiver, then though the expansion valves, to the air conditioning coils. When in this mode, the heat reclaim coils could be operational if air dehumidifying is needed. In this case, the air-cooled condenser (Cd) is out of service. The intermediate heat exchanger HEX is not operational, because the solenoid valves SLR are closed, as are the pressure regulating valves CPR.

Figure 2.11 "System*2 – phase1*": View of a Heat Reclaim Heat Pump

The heat pump HP-S contains only one compressor (C) with a liquid receiver, a plate heat exchanger as evaporator (S/C) and an air-cooled condenser (Cd) with fan (**Figure 2.12**). In heat reclaim mode, solenoid valves SLS and SDS are closed, while the pressure-regulating valve CPR and solenoid valves SLR are open. Superheated vapour from the dual temperature (DT) common discharge manifold transfers sensible and latent heat to the plate intermediate heat exchanger (HEX-S) that acts as an evaporator for the heat pump HP-S. The heat recovered is then supplied to the heat reclaim coils (back of store and hall entrance) in order to locally heat the indoor air.

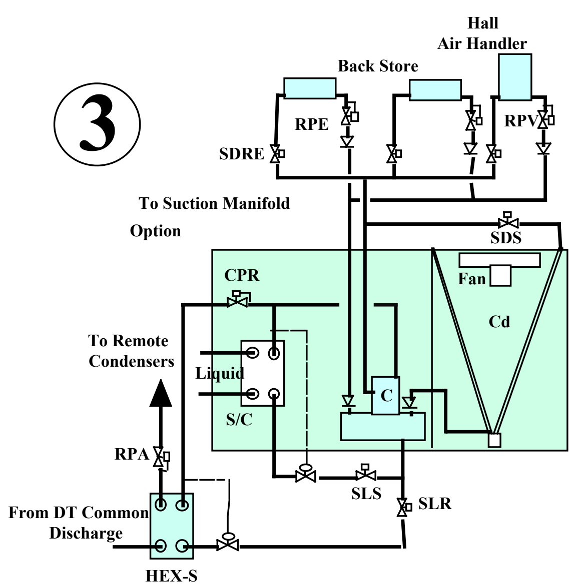

Figure 2.12 "*System2 – phase1*": Heat Reclaim /Sub-Cooling Heat Pump

For liquid sub-cooling purposes, the SLR valve closes, while SLS opens. The suction of the compressor C is connected to the outlet of the sub-cooling evaporator S/C. The superheated vapour from the compressor (C) is fed through the solenoid valve SDS to the air-cooled condenser (Cd), the liquid returns to the receiver, and is then fed through the solenoid valve SLS and the expansion valve to the S/C heat exchanger where the liquid coming from the main refrigeration system is cooled before returning to the main receiver.

Liquid Pumping

As noted above, the condensing pressure and temperature are mainly dependant on the outdoor temperature. Standard refrigeration systems are designed to compress the refrigerant vapour to approximately 200 psig, equivalent to a "saturated" condensing temperature of 95-100 °F in order to reduce the effects of vapour formed within the liquid line due to pressure losses (friction, fittings, filters, elevation, etc.). On the other hand, refrigeration compressor energy consumption and efficiency depend on the ratio between the condensing and the evaporating pressures. By lowering the condensing pressure, the compressor energy consumption and power diminish, and system refrigeration capacity increases. For example, by reducing condensing temperatures to as low as 10 °C (50 °F), savings of up to 1.5 % for each °F of condensing temperature reduction can be obtained, but it creates difficulties in heat reclaim because the air to be heated could be warmer then the refrigerant to be condensed. The defrost efficiency could also be affected. All these inconveniences are avoided by the developed "*System 2-phase1*" concept, because it uses a liquid delivery technology (**Figure 2.13**), a simple and reliable way of allowing the refrigeration systems to work better with floating condensing pressures [**9**]. The pumping station pressurises the liquid refrigerant enough to compensate for any pressure losses between the main receiver and the expansion valve. In this way, the formation of vapour ("flashing") can be avoided. The liquid pump is a sealed, magnetically driven unit, with no restriction when the shuts down, has no internal bearings, and is lubricated by the refrigerant (**Figure 2.14**). Power savings with this method could range from 1.0 % at low temperature to 1.5 % at high temperature applications for each °F of reduction in the condensing temperature. Moreover, because the pump pressure is usually higher than the compressor discharge, it is possible to inject a small percentage of the liquid (6 – 10 %) back into the discharge line. At high ambient conditions, the vapour temperature is reduced and thus the efficiency of the condenser increased, resulting in increased system capacity. In fact, liquid injection means lower velocity through the condenser, a lower pressure drop, and increasing area used for the condensing process. Liquid injection must only be operational for ambient temperatures above 75 °F, because it normally provides small benefits at lower ambient conditions.

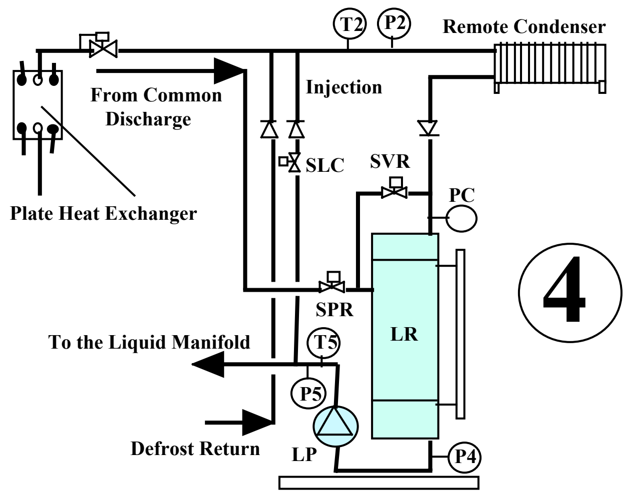

Figure 2.13 "*System2 – phase1*": Liquid Pumping Simplified Diagram
LR – liquid receiver; LP – liquid pump; PC – pressure control;
SPR – system pressure regulator; SLC, SVR – solenoid valves;
T – temperature; P - pressure

Figure 2.14 "*System2 – phase1*": View of the Dual temperature Liquid Pumping Station

Test Approach

As was the case with the previous system, the improved "*System2 – phase1*" was thoroughly instrumented in order to assess its operating and energy performance. About 50 measurement points (temperatures, pressures, electrical power, operating modes) were provided (**Appendix**). Temperatures were measured with type-T thermocouples, electric power – by using OSI watts transducers, and pressures – with calibrated amplified transducers. These parameters, scanned four times per minute and then saved every two minutes, permitted the calculation of instantaneous and average parameters. A software system, equipped with a HP-75000 acquisition station with integrated hard disks and modem, collected and sent data daily to the LTE laboratory central computer, where appropriate computer programs where used for data analysis.

Conventional System

A third "test supermarket", 18,000 ft^2 in total area (of which 12,200 ft^2 was for sales), built in 1999, and equipped with a conventional direct expansion-type refrigeration system (*METRO - Bordeleau*), ("*System-CON*"), was monitored (Figure 2.15). This system consists of a low temperature rack (-25 °F) with three semi-hermetic reciprocating compressors (19 kW), and a medium temperature rack (10 °F and 20 °F) with four compressors (23 kW). The store's refrigeration loads (Table 2.3) amount 121,380 Btu/h of low temperature cases and walk-in boxes (- 25 °F), 155,940 Btu/h of medium temperature loads at 15 °F, and 268,205 Btu/h at 20 °F (Appendix). This baseline store was operated on a similar schedule to that of the previously described test supermarkets. In this case, the heat reclaim was provided by a conventional method, and the back-up heating was done by a propane, direct-fired rooftop unit. In this system, when space heating was called for, a three-way valve was activated, directing superheated refrigerant vapour to a coil located inside a rooftop air handler. Sensible heat was transferred to the store and the refrigerant passed to the condenser where condensing occurred. Control of heat reclaim was handled through a two-stage store thermostat in which the first stage of heating was handled by condensation of the refrigerant. When heat reclaim was inadequate to meet the space-heating load, the second stage activated the auxiliary heater.

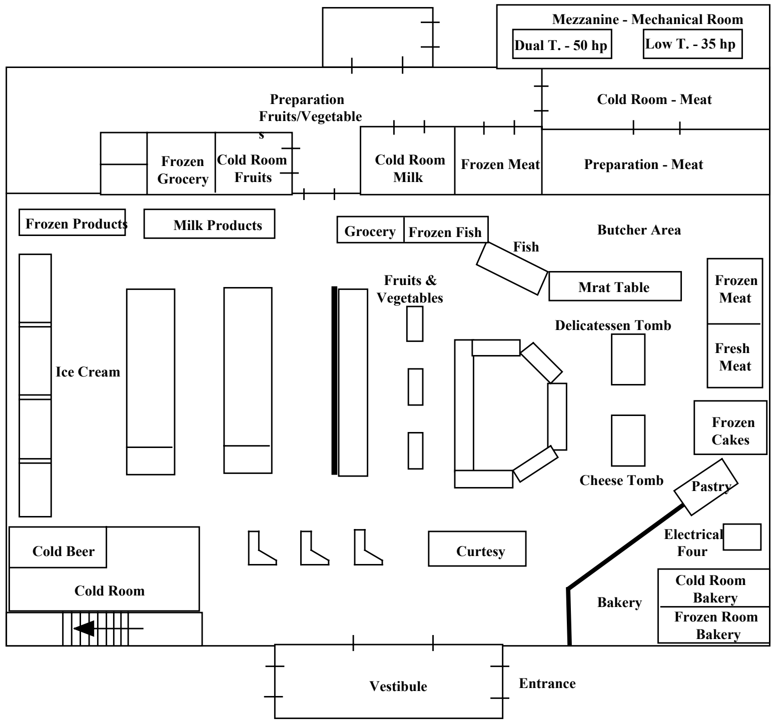

Figure 2.15 "*System-CON*": Schematic Layout of the Conventional Supermarket

Table 2.3 "*System-CON*": Refrigeration Capacities as a Function of SST (SST - Saturated Suction Temperatures)

Total Area	**Sale Area**	**Low/Dual SST**				**Medium SST**				
-	-	- 35 °F	- 25 °F	- 20 °F	- 15 °F	10 °F	15 °F	20 °F	30 °F	45 °F
ft^2	ft^2	Btu/h	Btu/h	Btu/h	Btu/h	Btu/h	Btu/h	Btu/h	Btu/h	Btu/h
18 000	12 200	-	121 380	-	-	-	155 940	268 205	-	-

Fourteen measurement points (temperatures, electrical power, etc.) enabled the comparison of some performance parameters with those of the other two improved systems.

3. FIELD MEASUREMENTS

Suction and Discharge Pressures

One of the objectives of this project was to survey the suction and discharge pressures of the refrigeration compressors in the context of using new heat reclaim, defrosting and liquid pumping technologies. As noted above, even if the variation of discharge pressures with changes in ambient conditions is currently used in conventional systems, there are still two important limitations in cold climates: during the winter, when heating demand is very significant, heat reclaim and defrost need hot gas at high head pressure and condensing temperature. These two obstacles were successfully by-passed by "*System2 – phase1*" because its operation with floating condensing pressure was facilitated by the liquid pumping system employed on both low and medium temperature refrigeration lines. When the head pressure was lowered during the cold periods of the year to a minimum of 125 psig, the liquid pump boosted the refrigerant prior to its flowing to the expansion valves. On the other hand, when no heat reclaim was called for during the warmer periods of the year, the discharge pressures "floated" on the outdoor temperature with the aid of a conventional condenser fan control **Figures 3.1 and 3.2**). At the same time, "*System1 – phase1*" operated at higher discharge pressures on both low and medium temperature refrigeration lines. During the cold weather period, primarily comprised of heat recovery periods, the average discharge pressure varied around 175 psig. However, during the first summer of the system's testing period (2001), the appropriate operation of groundwater cooling heat exchangers lowered the compressor head pressures, and thus saved energy, because one or even two compressors shut-down.

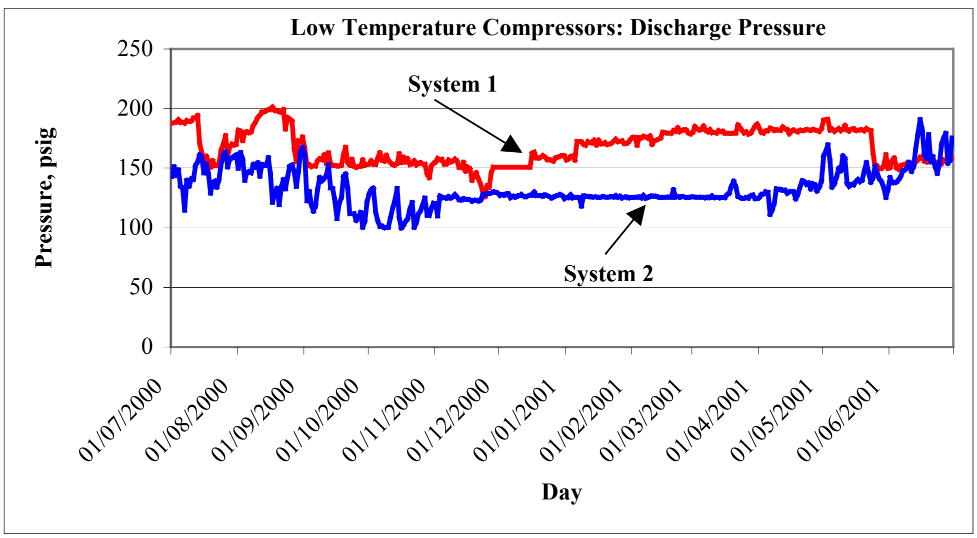

Figure 3.1 Daily Average Discharge Pressure Comparison – Low Temperature Compressors

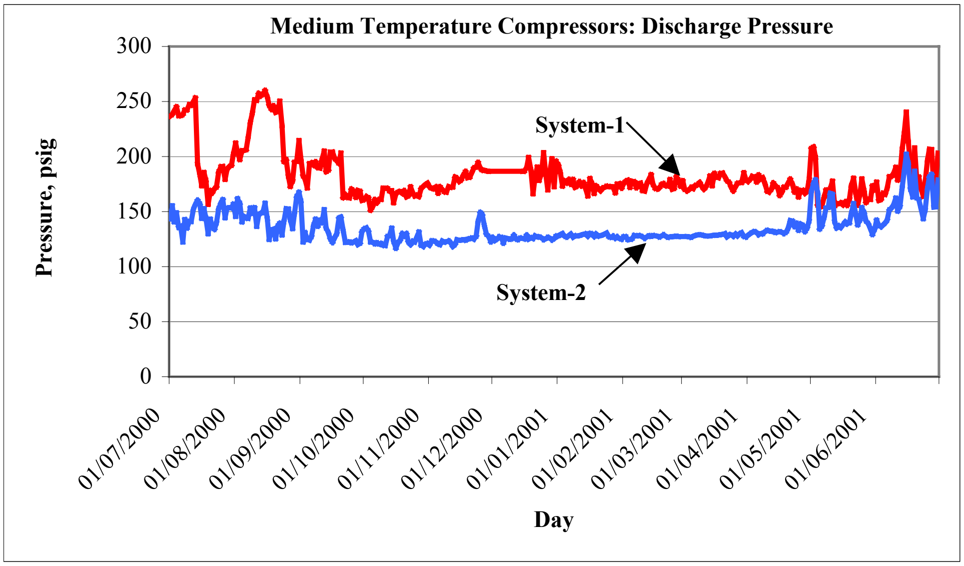

Figure 3.2 Daily Average Discharge Pressure Comparison – Medium Temperature Compressors

Evaporating and Condensing Temperatures. Sub-Cooling

A significant disparity between the two systems was observed with respect to the condensing temperatures as a consequence of the compressor head pressure strategy control (**Figures 3.3 and 3.4**). During the winter, "*System1-phase1*" worked at an average condensing temperatures of 32 to 35 °C (89.6 to 95 °F), while in "*System2-phase1*", the condensing temperatures varied around 22 °C (71.6 °F) on both dual and medium refrigeration racks. This situation was a direct consequence of the heat reclaim methods employed. Effectively, while the "*System2-phase1*" was able to recover latent heat at a condensing temperature as low as 22°C, because of use of intermediate heat exchangers coupled with refrigerant-to-air heat pumps as a cascade system, the "*System1-phase1*" had to keep a higher condensing temperature in order to heat and supply heated air at a high enough temperature (30°C or 86 °F). However, during the warm periods of the year, the condensing temperatures fluctuated with the ambient conditions on both systems. In the "*System2-phase1*", the condensing temperatures were also lowered further during the summer by using the liquid injection technique allowing an increase in overall system efficiency.

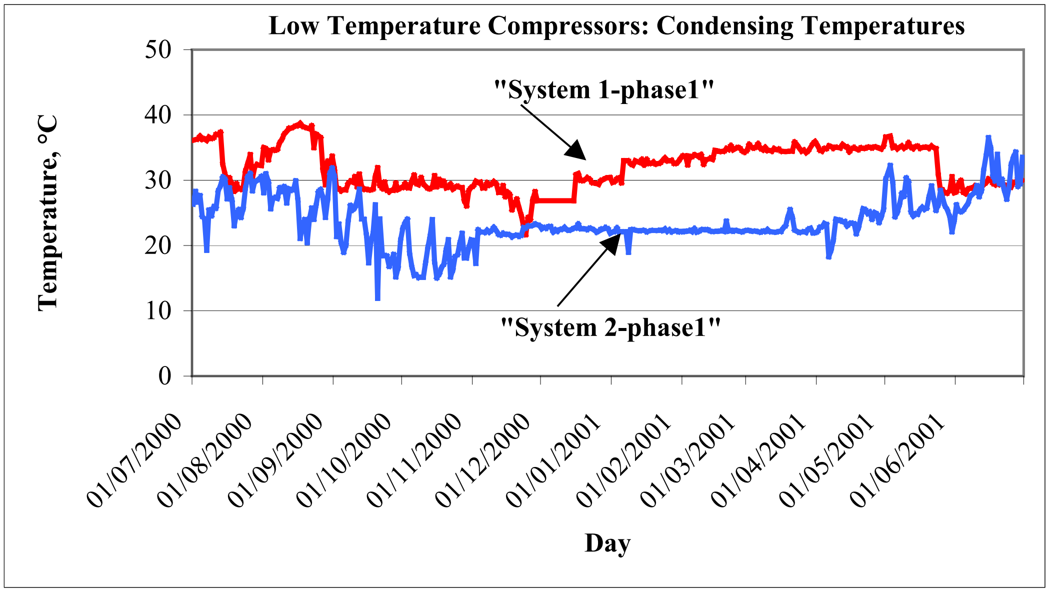

Figure 3.3 Condensing Temperature Comparison – Low Temperature Compressors

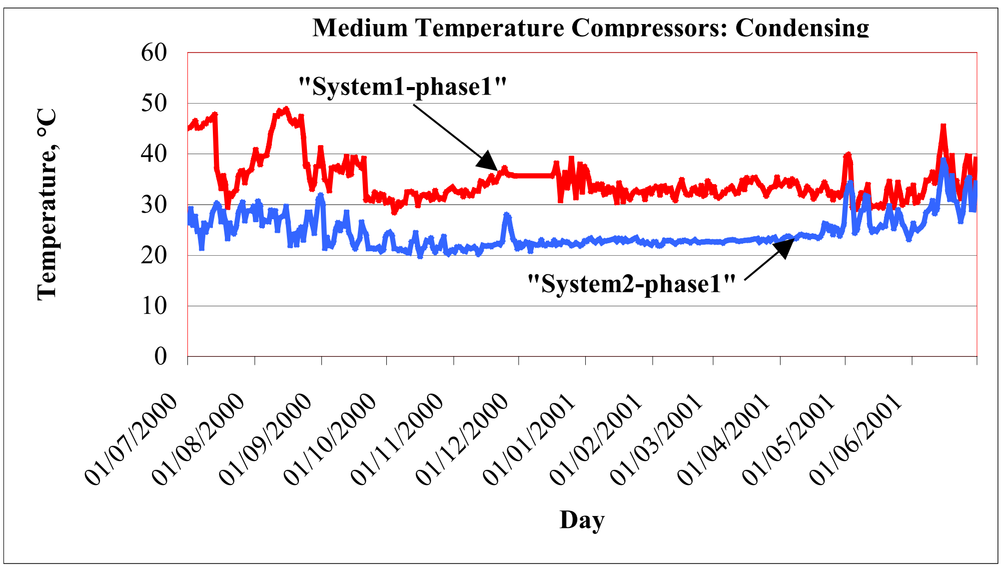

Figure 3.4 Condensing Temperature Comparison – Medium Temperature Compressors

For liquid sub-cooling, the "*System1-phase1*" used a conventional method involving slightly oversized condensers and additional passive heat exchangers and thermostatic expansion valves. Insufficient energy savings were generated by this process, because the amount of measured sub-cooling (5°C or 9 °F) was lower than the design value (20°C or 36 °F) (**Figure 3.5**). On the other hand, the "*System2-phase1*" employed a new mechanical sub-cooling method provided by a dedicated heat pump (HP-S) having its own HCFC-22 refrigerant circuits. This heat pump operated independently of the refrigeration system about 2,698 hour/year in sub-cooling mode during the 12-month monitoring period. Excess heat was directly rejected to the ambient air. However, subcooling with the heat pump requires an efficient control of refrigerating capacity, since the flow rate of the liquid to be cooled is variable. A strong sub-cooling effect of more than 20°C (or 36 °F) was in fact measured during the first winter of operation (**Figure 3.6**).

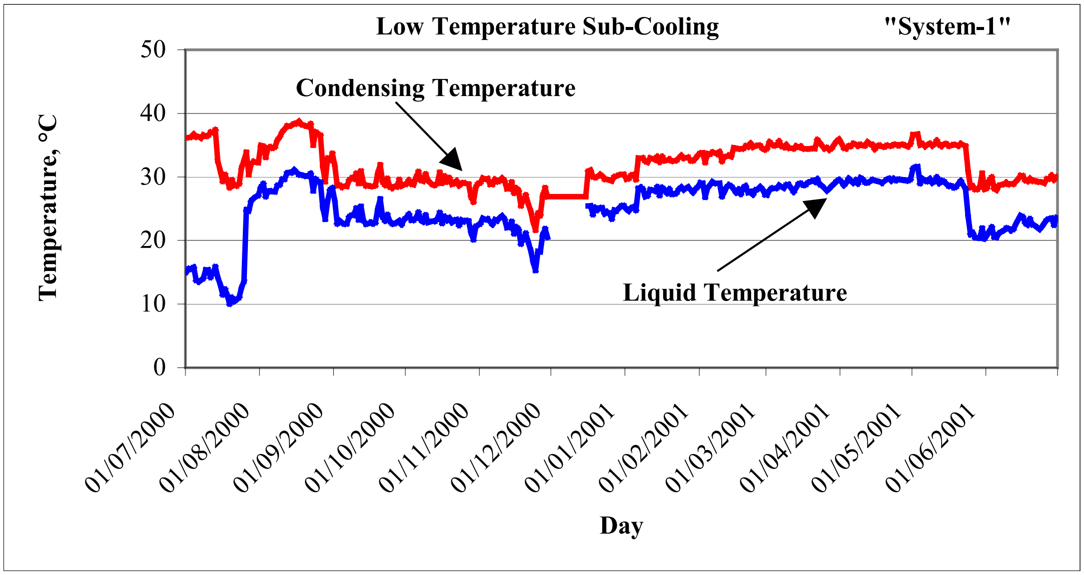

Figure 3.5 "*System1-phase1*": Condensing and Sub-Cooled Liquid Temperatures – Low Temperature Compressors

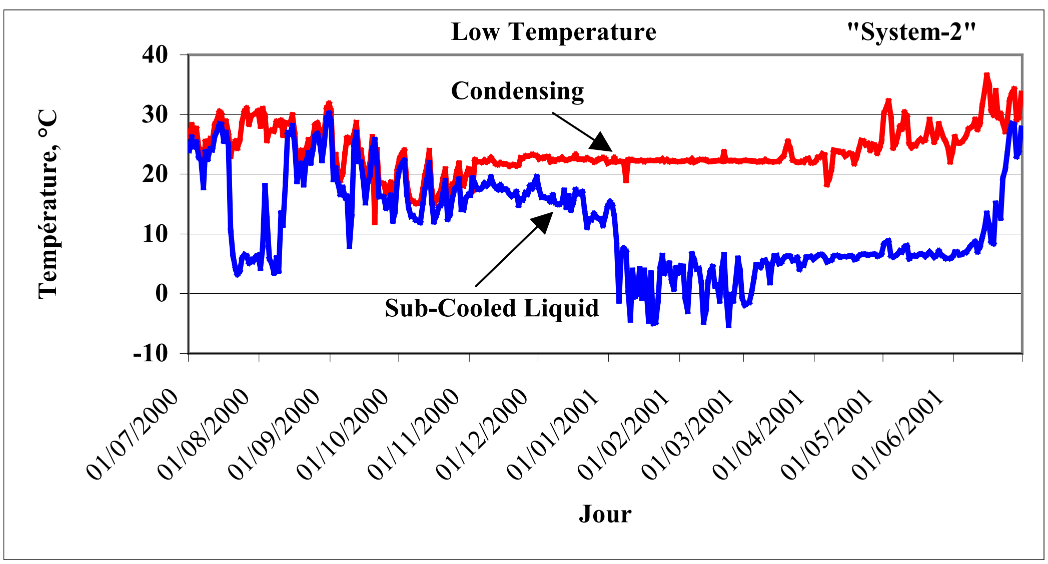

Figure 3.6 "System*2-phase1*": Condensing and Sub-Cooled Liquid Temperatures – Low Temperature Compressors

Total Heat Reclaim

A part of the maximum thermal power rejection available in "*System1-phase1*" ($1.6*10^6$ Btu/h or 469 kW) was successfully used to satisfy up to 100 % of the store's heating load, and also for domestic water heating requirements. Domestic water was preheated to 40°C (104 °F) (**Figure 3.7**)

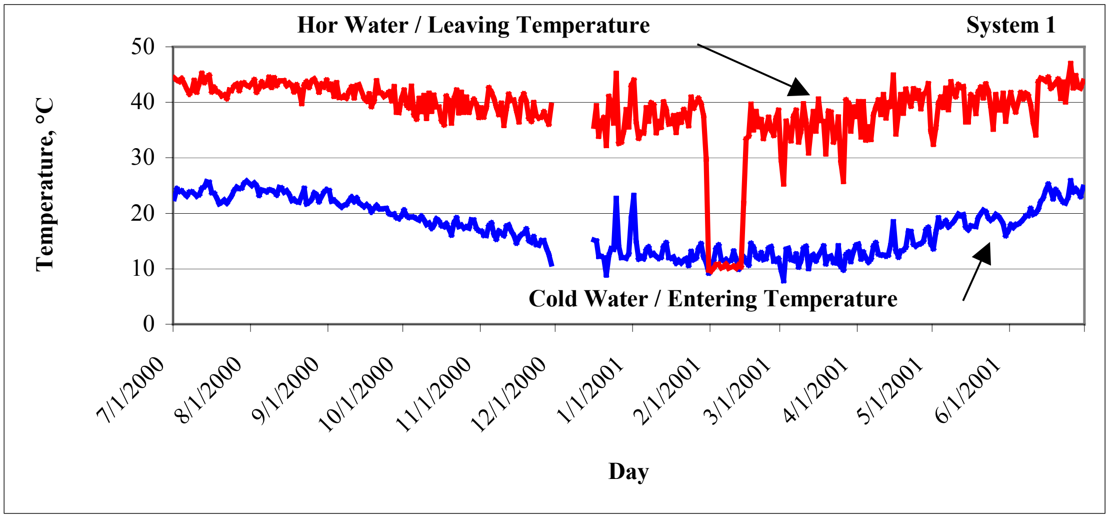

Figure 3.7 "*System1-phase1*" Hot Water Temperature Profiles

As previously noted, the "*System2-phase1*" prototype, the operation of which was completely independent of the heat reclaim section, provided heat recovery without excessively increasing the discharge pressures. During the winter, heat reclaim was made possible through intermediate plate heat exchangers, and four cascade heat pumps operated with coefficients of performance of minimum 3. This cascade arrangement allowed raising of the thermal potential of the delivered heat to 29°C or more during the dominant heat reclaim-heating mode. **Figure 3.8** and **Figure 3.9** represent two daily operating profiles in heat reclaim mode for the heat pumps HP-B and HP-S respectively. It is observed that the first compressor of the heat pump HP-B operated about 50 % of the total annual time in heat reclaim mode, while the second compressor was only used about 32 % of the time. During the summer, these compressors operated in dehumidifying mode.

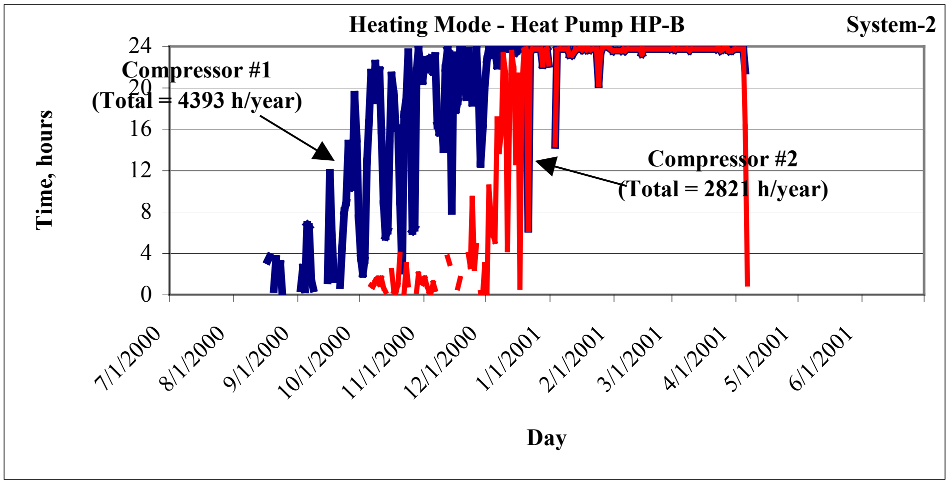

Figure 3.8 "*System2-phase1*": Daily Operating Time in Heat Reclaim Mode (Heat Pump *HP-B*)

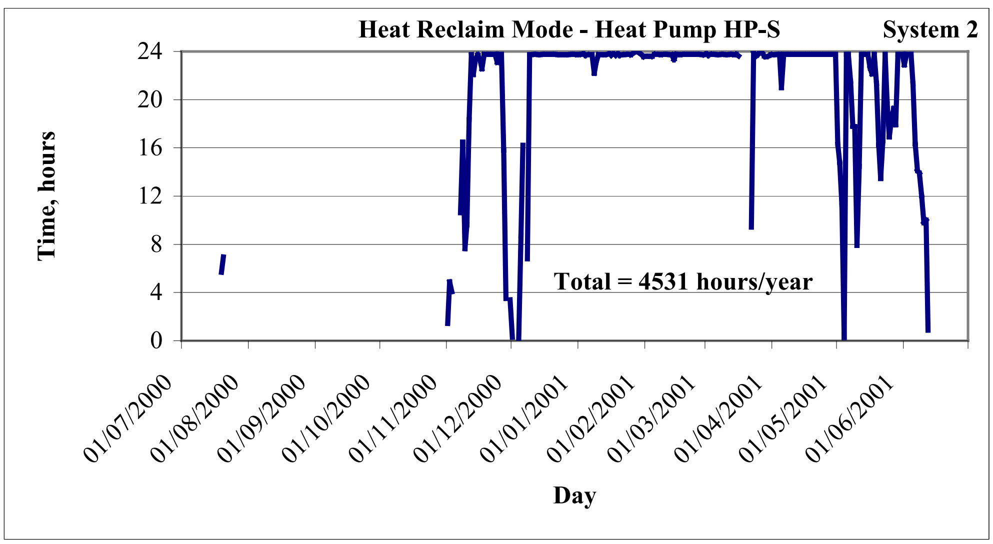

Figure 3.9 "*System2-phase1*": Daily Operating Time in Heat Reclaim Mode (Heat Pump *HP-S*)

Different areas of the test stores were heated to proper temperature levels using only recovered condensing heat. For example, the heat reclaim coil located in the central HVAC air handler of the "*System1-phase1*" supplied air to the store at about 29°C (84 °F) average temperature during the winter. In the second store, the temperature in the entrance hall varied between 20°C and 25°C in February (**Figure 3.10**), and between 20°C and 30°C in March (**Figure 3.11**).

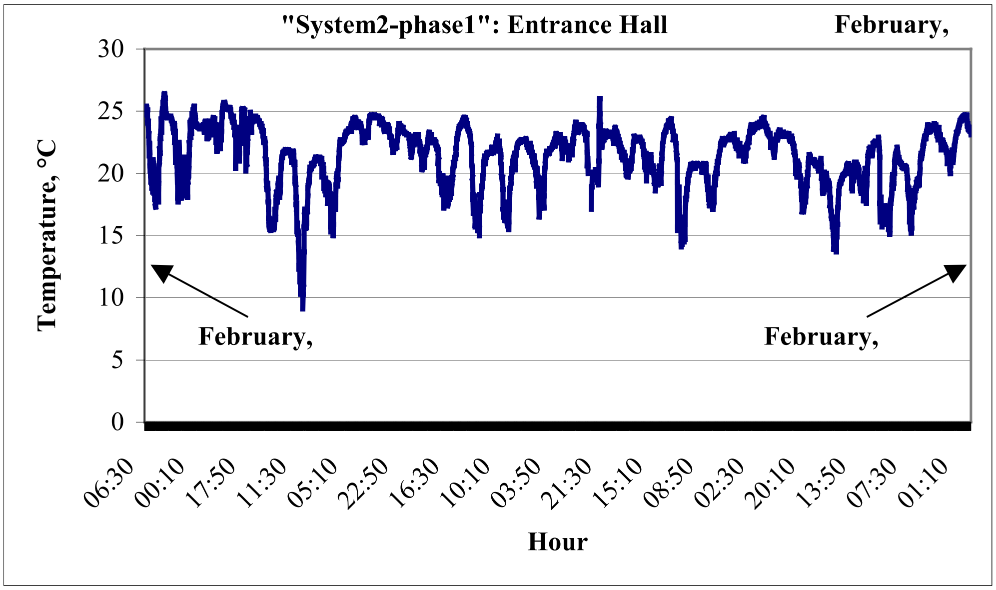

Figure 3.10 "*System2-phase1*" : Entrance Hall Air Temperature (February, 2001)

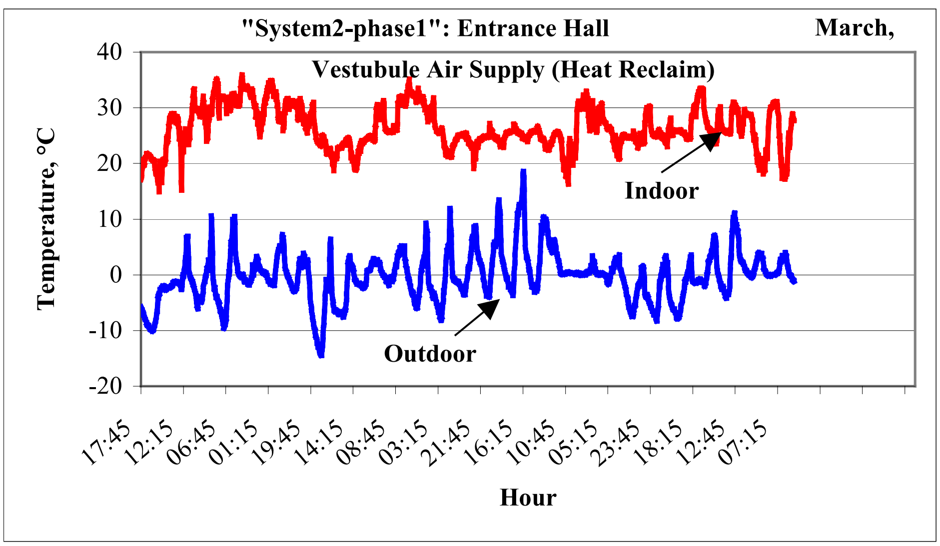

Figure 3.11 "*System2-phase1*" : Entrance Hall Air Temperature (March, 2001)

During the winter (for example, in January), the store's average indoor air temperature was kept constant at around 23°C (73.6 °F) (**Figure 3.12**), also by the exclusive use of heat recovered from the refrigeration system.

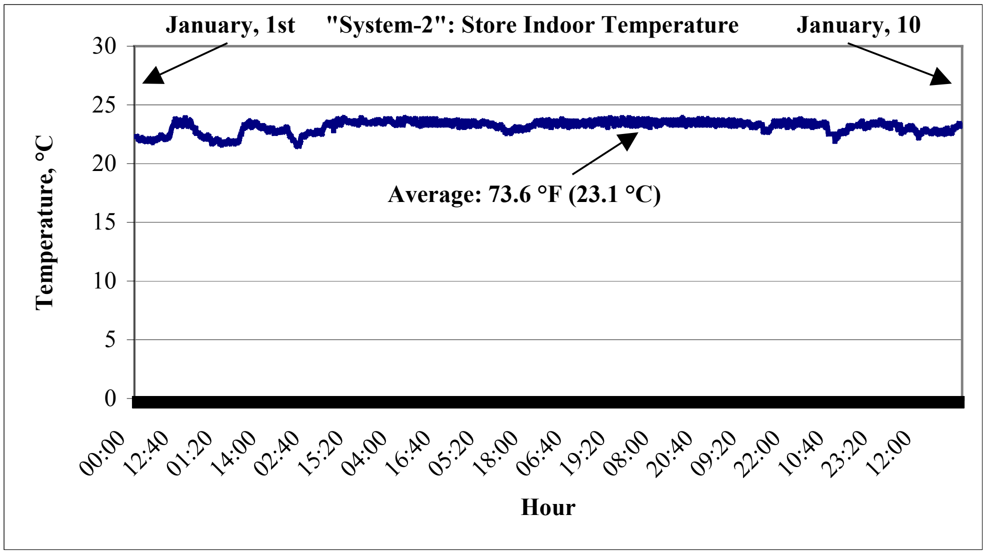

Figure 3.12 "*System2-phase1*" : Store's Indoor Air Temperature (January 1st to January 10, 2001)

Conclusions

- For heat recovery purposes, the "*System1–phase1*" raised the compressor head pressures during heating requirement periods, and also modified the conventional refrigerant flow and controls to eliminate the current limitation of heat reclaim capability and to give priority to the heat reclaim mode. A modulating valve controlled by the ambient temperature optimizes this process by varying head pressures as a function of the actual heating demand and the outdoor temperature variation.

- The "*System2-phase 1*" proved capable of reducing discharge pressures to an almost constant level (about 29 % lower than the other system), without compromising its own heat reclaim method. The discharge pressures were lowered to about 125 psig during the winter, but the thermodynamic level of the heat reclaim was increased by using heat pumps as a cascade stage. Savings in compressor energy consumption were obtained mainly by using four refrigerant-to-air heat pumps having average coefficients of performance of around 3.

- The 12-month field monitoring proved that *fossil fuel consumption* for space and domestic hot water heating *had been completely eliminated* by both systems tested. To achieve this, a supplementary quantity of electrical energy of about 300,000 kWh/year was consumed by each system, compared with a baseline de-superheating heat reclaim system. There wasn't substantial energy cost savings, because of the actual price ratio between electricity and natural gas, but the net emission of greenhouse gases was reduced by about 286,000 kg CO_2/year (assuming 6.64 kg CO_2/m^3 of natural gas consumed). (note: in Quebec, almost 100 % of electric energy comes from hydraulic sources).

High-Speed Defrost

The defrost initiation for both improved systems was done with timers, and termination of the cycles was performed on the basis of the actual coil temperatures. In the conventional system, the defrost cycle duration is set long enough to ensure complete defrosting for worst-case situations, and a minimum temperature of 21°C (70 °F) is needed for proper operation of the hot gas defrost cycles. The temperature-based termination of defrost cycles allows "*System1-phase1*" to take advantage of their newly developed defrost method that imposed a high pressure differential across the evaporators. The main result of this method was a reduction by about 85 % of the defrosting cycle length versus that of the conventional system (**Table 3.1**). For example, for a typical medium temperature display-case (fresh meat), the average duration of defrost cycles was 2.3 minutes/cycle, compared to 12 minutes/cycle for the conventional system (**Figure 3.13**), or 39.32 hours versus 265.48 hours between October 1st 2000 and June 30, 2001 (**Figure 3.14**). It was estimated by the store owner himself, that the average loss of fresh meat, which is normally about 4 % of the total annual store business in conventional supermarkets, was reduced to 2 % or less.

Table 3.1 "*System1-phase1*" vs. "*System CON*": Comparison of Two Defrosting Cycles (October 1st to June 30, 2001)

System	**Medium Temperature Circuit**	**Number of Defrosting Cycles**	**Average Cycle's Length**	**Total Defrosting Time**	
-	**-**	**-**	**Minutes**	**Hours**	**Minutes**
System1-phase1	15 °F – Fresh Meat (#21)	1 084	2.8	50.04	3 002
	20 °F – Fresh Meat Room (#30)	1 023	2.3	39.32	2,359.20
System CON	15 °F – Fresh Meat (#19)	1 332	12	265.48	15,928.80
	15 °F – Fresh Meat Room (#20)	1 069	12.9	230.78	13,846.80

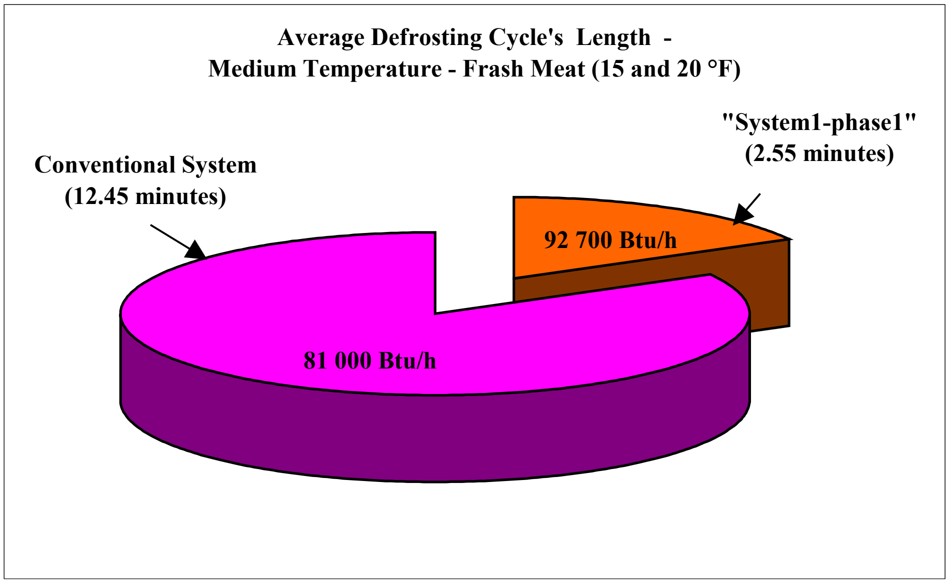

Figure 3.13 Comparison of the Average Defrosting Time– Medium Temperature (15 and 20 °F) (October 1st, 2000 to June 30, 2001)

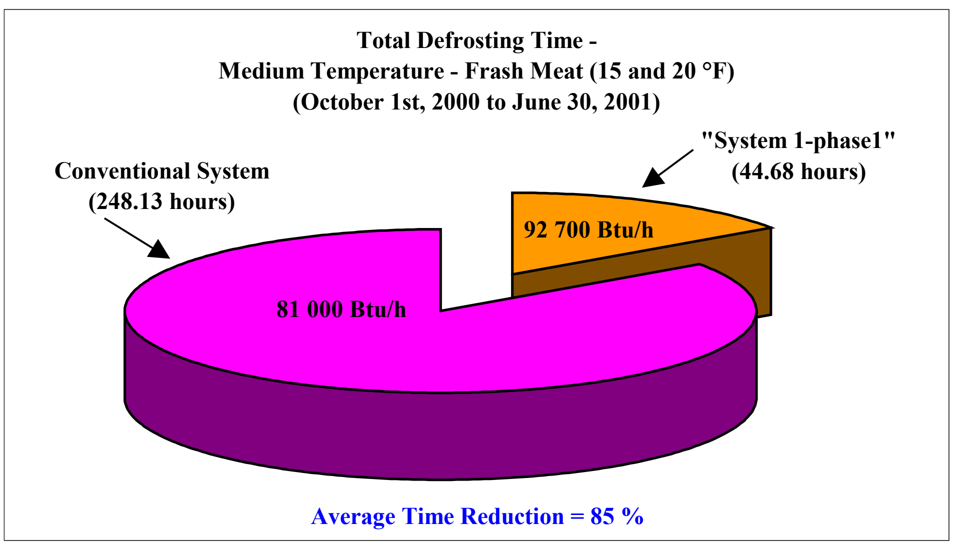

Figure 3.14 Comparison of the Total Defrosting Time – Medium Temperature (15 and 20 °F) (October 1st, 2000 to June 30, 2001)

The **Figure 3.15** and **Figure 3.16** compare the times for four defrost cycles measured on similar fresh meat display cases in the ”*System1-phase1*” and the conventional store respectively.

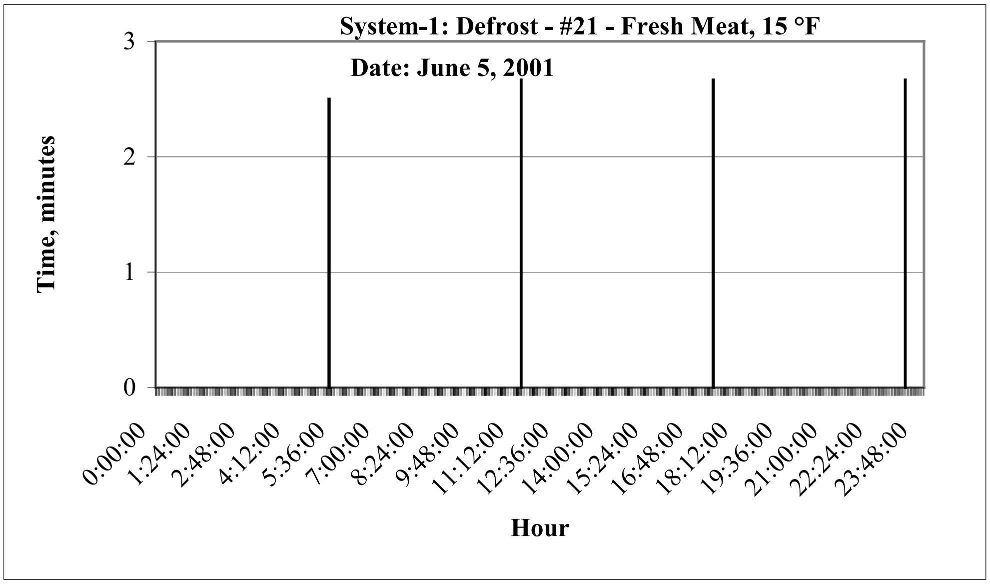

Figure 3.15 "System1-phase1": Length of Defrost Cycles (#21 - Fresh Meat) (Example : June 5, 2001)

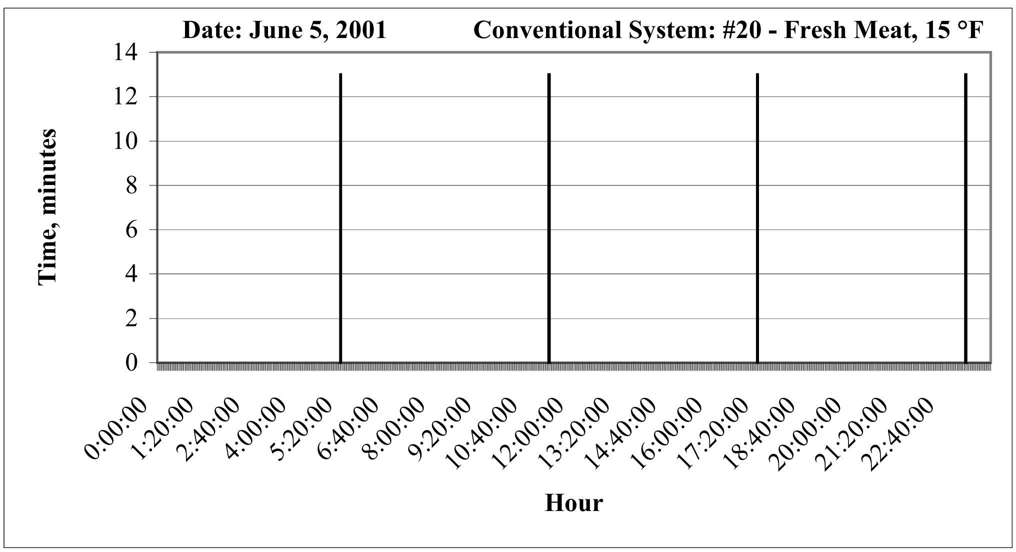

Figure 3.16 "System1-phase1": Conventional System: Length of Defrost Cycles (#20 - Fresh Meat) (Example : June 5, 2001)

An important issue for the "*System1-phase1*" high-speed defrost method was the influence of the high-speed circulation of superheated vapour inside the evaporators during the defrost cycle. A comparative study of piping expansion was performed *« in-situ »* [**13**] in order to determine if there was any danger of damaging the evaporator's L-type copper materials. The measurements of the material's constraints were done on two identical evaporators, the first one in a conventional system with long defrost cycles, and the second in the monitored store equipped with the high-speed defrost concept.

Maximum deformation of pipes obtained on both evaporators represented about 56 % of their respective elastic limits. The study concluded that there is no apparent danger of damaging the evaporators when fast defrost cycles are done with the "*System1-phase1*" method at this stage of the development.

Conclusion

The high-speed defrost method proposed by "*System1-phase1*" was able to reduce the defrost cycle length by about 85 % compared with the baseline hot gas defrost method. The store owner estimated that actual beneficial effects resulted in better quality and about 50 % less perishable food losses. The actual maximum deformation of a representative case evaporator during a fast defrost cycle represents about 56 % of the elastic limit of existing materials, but the long-term effects are not known. Finally, the high-speed defrost method could, in future installations, allow a reduction in the number of the refrigeration lines by increasing their total refrigerant transport capacity. Consequently, the total store refrigerant charge could be reduced by up to 20 %.

Liquid Pumping

By using cascade heat pumps for heat reclaim and liquid pumping technology, the "*System2-phase1*" was able to operate with floating head pressures, and thus to lower the compressor discharge pressures by up to 125 psig during the heating requirement period in winter. Moreover, the liquid delivery pumps have contributed to:

- Avoid the liquid "flashing" and increase the refrigeration capacity of about 1.4 % for every °F drop in condensing temperature.
- Reduce the discharge superheating by injecting liquid, particularly during the periods of high outdoor temperatures, resulting in less compressor work and improved condenser efficiency.

During the 12-month experimental testing period, the average pressure differential across the liquid delivery units varied around 26 psig (**Figure 3.17**). It was estimated that this technology could produce energy savings of up to 35 % [**9**].

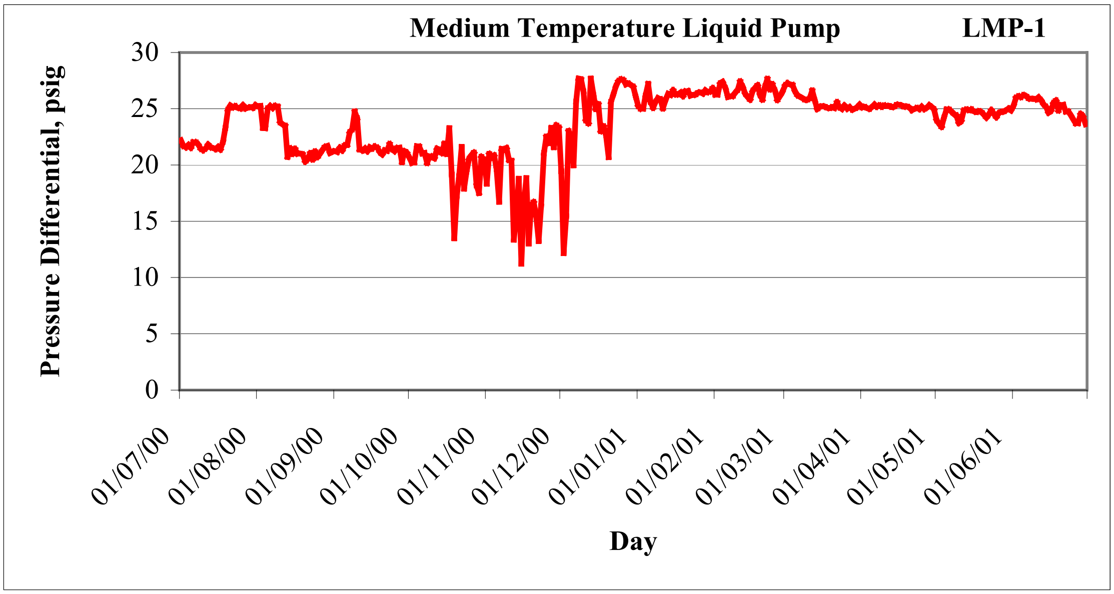

Figure 3.17 "System2-phase1" : Pressure Differential Across the Liquid Delivery Pump (Medium Temperature Line)

Energy Consumption
System monitoring enabled a determination of the daily average power demand for each store and for their low and medium temperature refrigeration compressors (**Figure 3.18** and **Figure 3.19**). The measured daily average power profiles were nearly alike, but any relevant comparison of these profiles was not possible because of a variety of refrigerant loads, the number and type of refrigeration compressors, and other non-measured electrical loads (lighting, HVAC, etc.).

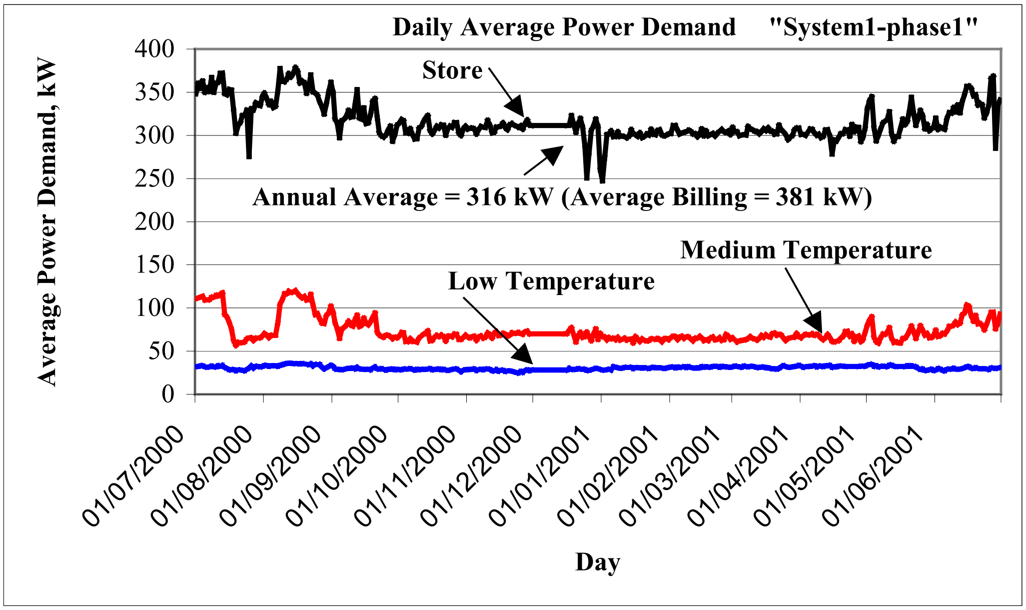

Figure 3.18 "*System1-phase1*": Measured Daily Average Power Demands

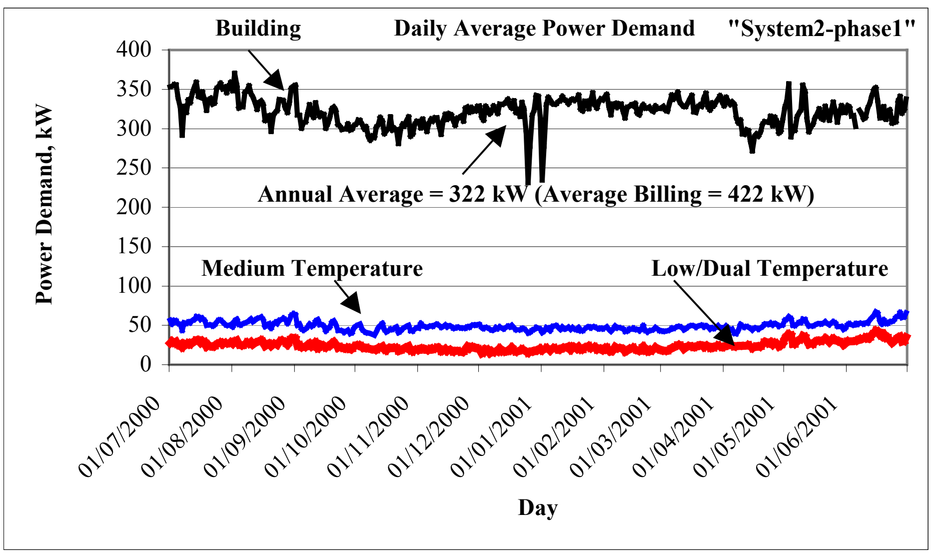

Figure 3.19 "*System2-phase1*": Measured Daily Average Power Demands
(excluding Heat Pumps)

The daily electrical energy consumption of the improved stores and their refrigeration compressors are shown in **Figure 3.20** and **Figure 3.21** respectively. The annual electrical energy consumption of the two improved supermarkets (including the refrigeration compressors) was almost identical (about 2,770,000 kWh/year) (**Table 3.2**), which represented 97.9 kWh/ ft^2 versus 89.8 kWh/ ft^2 of sales area respectively, while the total annual specific consumption of the conventional store (*CON1*) was 95.5 kWh/ ft^2, excluding the electric energy for cooking. These results show that the specific annual energy consumption of the "*System1-phase1*" store was 2.4 % higher than the conventional store "CON", while the specific annual consumption of the "*System2-phase1*" store was 5.7 % lower. This difference in energy consumption could in part be attributed to the use of cascade heat pumps for heat reclaim instead of increasing compressor head pressures. The higher specific energy consumption of the "*System1-phase1*" store could also be attributed to other elements (lighting, traffic, operating schedule, etc.). However, in terms of total energy cost (including power penalty), both stores had practically the same specific energy cost (5.6 vs. 5.3 CD$/ ft^2.year), but were respectively 12.5 % and 17.2 % lower that those of the conventional system (6.4 CD$/ ft^2.year), excluding the electrical energy consumption for cooking. These experimental data combine the actual utility power penalties and local energy costs (2001).

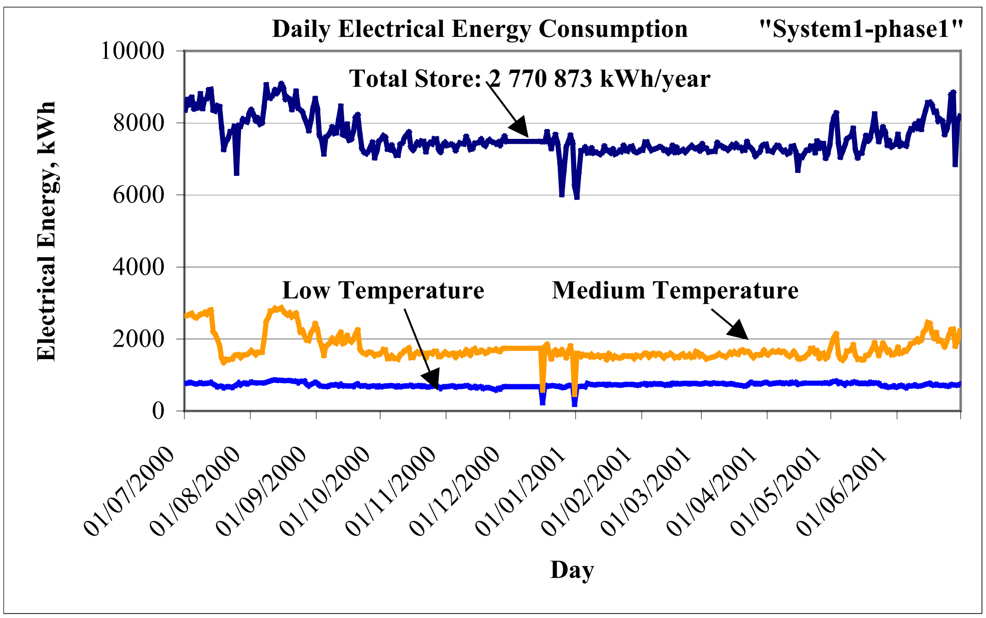

Figure 3.20 "*System1-phase1*": Store and Refrigeration Compressor Daily Energy Consumption

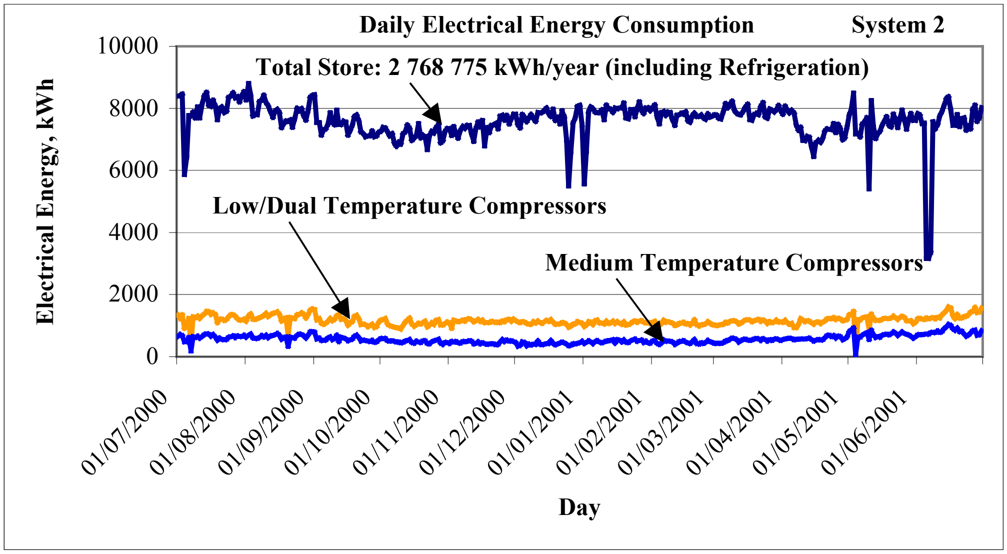

Figure 3.21 "System2-phase1": Daily Energy Consumption Profiles for Store and Refrigeration Compressors

Table 3.2 Total Store Annual Electrical Energy Consumption and Costs (Including Demand Power Cost) vs. Sale Area (CAN$)

System	**Sale Area**	**Total Store Electrical Energy Consumption (including compressors and heat pumps)**			**Total Store Electrical Energy Cost (Power : 11,97 $/kW)**		
	ft^2	**kWh/year**	**kWh/ ft^2.year**	**vs. CON.1 (%)**	**$/year**	**$/ ft^2.year**	**vs. CON.1 (%)**
Syst-1	28,302	2,770,873	97.9	+ 2.4	157,.704	5.6	- 12.5
Syst-2	30,838	2,768,775	89.8	- 5.7	163,411	5.3	- 17.2
CON. 1	12,200	1,095,462*	95.5*	-	78 440*	6.4*	-

* Without Electrical Cooking

Another goal of the monitoring was to establish the actual compressor energy consumption as a percentage of total store energy consumption. **Table 3.3** summarizes the total energy consumption and costs for each rack of refrigeration compressors, including the heat reclaim heat pumps existing in "*System2-phase1*". **Figure 3.22** represents the annual profile for heat pump total energy consumption (more than 313,000 kWh/year).

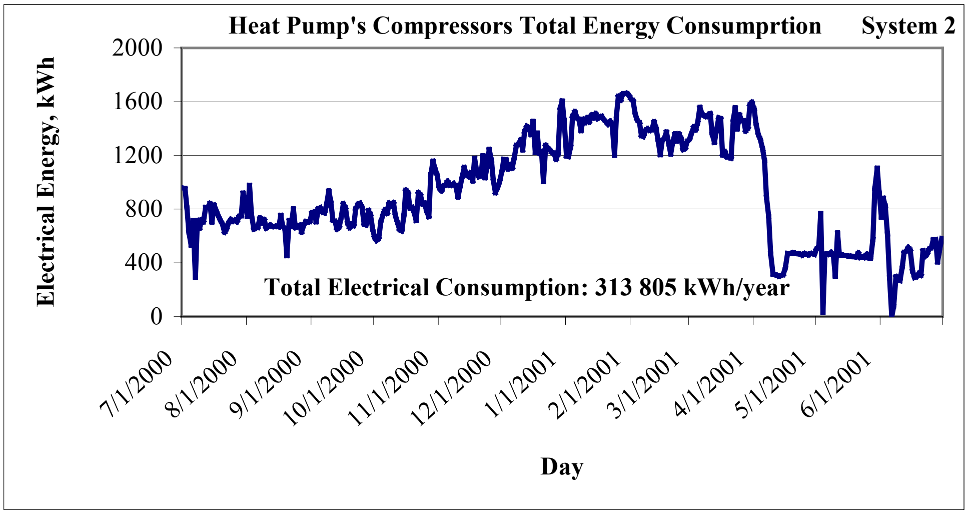

Figure 3.22 "*System2-phase1* ": Heat Pump Energy Consumption Profile

It can be seen that the total annual energy consumption for the refrigeration compressors as a percentage of total store energy consumption was 32.5 % (*System1-phase1*) (**Figure 3.23**) and 34 % (*System2-phase1*, including the four heat pumps) (**Figure 3.24**), being 5.8 % and 10 % higher respectively than those of the conventional systems. In terms of the annual specific energy costs (reported for the sales area), both refrigeration compressors had costs that were about 8.6 % less than those of the "*CON1*" system. It should be also noted that "System1-phase1" was penalized by using screw compressors on the medium temperature rack, and high discharge pressures during the winter for heat reclaim. It was estimated that the use of these compressors has increased total compressor annual energy costs by about 15 %. During the summer, fouling and lack of maintenance of geothermal cooling devices also caused higher head pressures than those predicted. By way of comparison, the annual energy consumption of the supermarket compressors typically accounts for up to 39 % of total store energy consumption.

Table 3.3 Annual Compressor Electrical Energy Consumption and Costs (Including Power Demand) – vs. Sales Area (CAN$)

System	Sale Area	Compressors	Total Energy Consumption (Compressors only)			Total Electrical Energy Cost (Compressors only)	
	ft^2	-	kWh/year	% of Total Store Cons.	kWh/ ft^2.year	$/year	$/ ft^2.year
Syst-1	28,302	Low Temp.	264,906	9.6	9.4	15,077	0.53
		Medium T.	637,299	23	22.5	36,262	1.28
		TOTAL compressors	902,205	32.5	31.9	51,339	1.81
Syst-2	30,838	Low/Dual T.	423,291	15.3	13.7	24,982	0.81
		Medium T.	206,298	7.5	6.69	12,172	0.39
		TOTAL compressors	629,589	22.7	20.4	37,154	1.2
		Heat Pumps	313,805	11.3	10.2	18,521	0.6
		TOTAL	943,394	34.0	30.6	55,675	1.8
CON.	12,200	TOTAL compressors	355,721	30.6*	29.2	23,995	1.97

* Excluding electrical cooking.

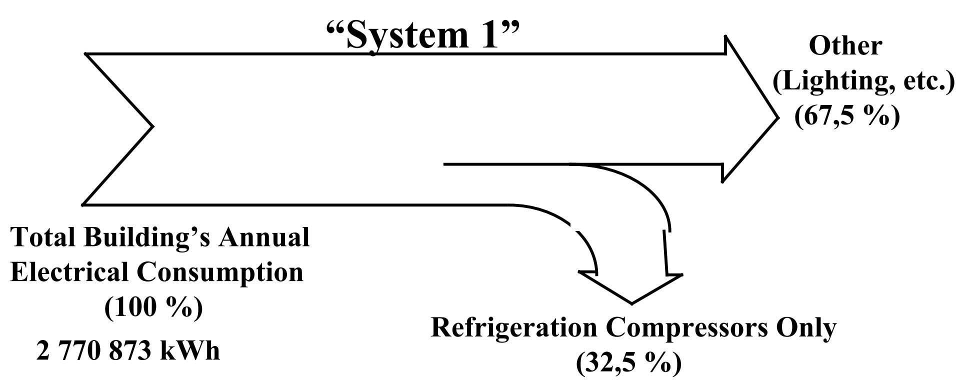

Figure 3.23 "*System1-phase1*" - Annual Balance of Store Energy Consumption

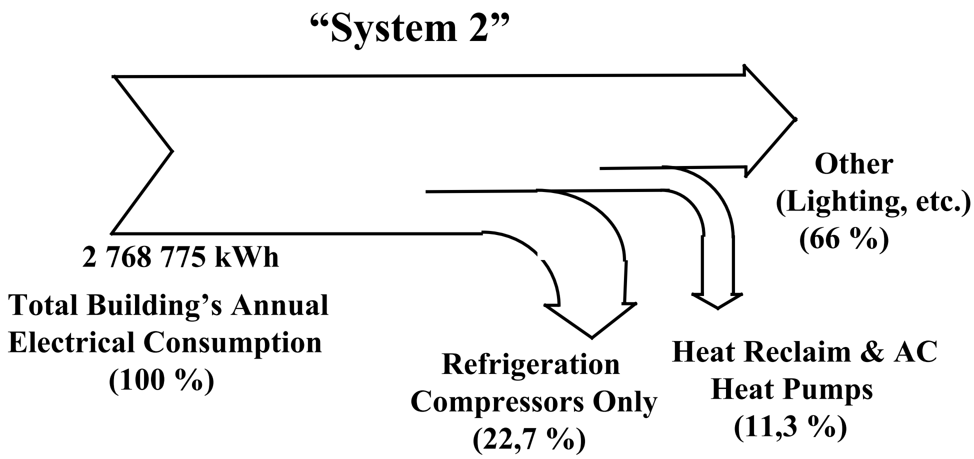

Figure 3.24 "*System2-phase1*" - Annual Balance of Store Electrical Energy Consumption

A comparison of the natural gas usage of the improved supermarkets showed that "*System2-phase1*" consumed 20.5 % less combustible fuel for cooking than "*System1-phase1*" (**Table 3.4**). This large difference could be explained by significant differences in installed capacities, usage operating schedules, etc. However, compared to other similar conventional stores, the improved stores have reduced natural gas consumption by 26 %, a performance mainly attributed to the complete elimination of fossil fuel for space and hot water heating.

Table 3.4 Store Annual Natural Gas Consumption vs. Sales Area

System	**Sale Area**	**Total Consumption**		**Equivalent Energy**		**Total Natural Gas Cost (CAN$)**	
	ft^2	m^3 /year	m^3 / ft^2	GJ/year (kWh/year)	GJ/ ft^2 /year (kWh/ ft^2 /year	$/year	$/ ft^2 /year
Syst-1	28,302	45,179	1.59	1.6 ($4.4*10^6$)	$5.6*10^{-5}$ (155)	22,534	0.80
Syst-2	30,838	35,625	1.15	1.27 ($3.5*10^6$)	$4.12*10^{-5}$ (113)	17,860	0.58

Conclusions

- The total annual specific energy consumption of the *refrigeration compressors* represented 32.5 % ("*System1–phase1*") and 34 % ("*System2–phase1*", including the four heat reclaim-heat pumps), compared to store total electrical energy usage, while the average consumption of traditional systems is currently 39 %.

- The total annual specific electrical energy consumption (vs. sales area) of improved *supermarkets* was reduced by up to 6 %, and the corresponding energy costs, including power demand cost, by 12.5 % and 17.2 % respectively, compared to the baseline monitored supermarket.4.

4. FURTHER IMPROVEMENTS

As noted above, in Canada, the majority of the supermarkets still use direct expansion refrigeration systems and operate in a typical cold climate where space-heating loads are an important issue. However, these systems still use large quantities of HCFC or HFC refrigerants, and environmental constraints are an impetus to reducing charges and the associated risks of leakage.

"System 1"

The first development step for "*System 1*", in effect since 1999, aimed to develop new heat reclaim and defrosting methods for existing multiplex systems. The first objectives were to improve the capability for total heat reclaim and the effectiveness of the defrost cycles. "*System 1-phase 1*" comprised a central refrigeration room with two compressor racks (low and medium temperature). Both sides of the system (direct expansion refrigeration and condensing heat rejection) used the same primary refrigerant (**Figure 4.1**), and thus the piping length, the total amount of refrigerant (3,500 lbs) and leakage rates (about 20 % annually) were still very significant.

No reduction in the number or length of the refrigeration lines, and no refrigerant substitution were proposed during this first stage of the "*System1*" development. Even if the newly developed high-speed defrost method could reduce the total refrigerant charge by up to 20 %, this amount is considered insufficient today. These were the main reasons why the two new improved systems ("*System 1–phase 2*" and "*System 1–phase 3*") were developed since 2001. The objectives were to reduce the total refrigerant charges and risks of leakage, and also to replace the common synthetic refrigerants by secondary fluids, and thus develop environmentally low-TEWI systems, while preserving previous innovations and technical expertise.

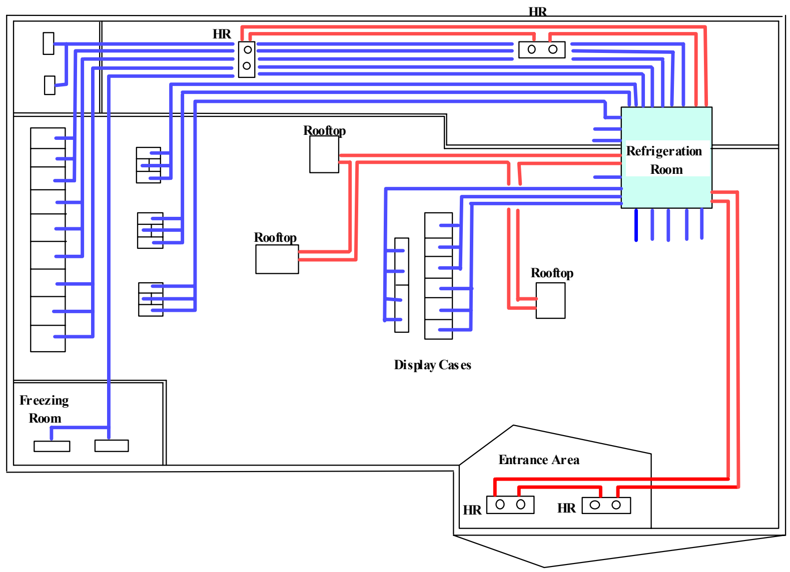

Figure 4.1 "*System 1 - phase1*": Improved Multiplex Supermarket Refrigeration System

The second step in the development ("*System 1–phase 2*") (**Figure 4.2**) retains both main prior innovations (integral heat reclaim and high-speed defrost), replaces the refrigerant HCFC-22 with R-404a, an azeotropic, non-ozone depletion fluid. The central refrigeration room is eliminated and the space saved may be used for commercial activities. The central multiplex system is divided into several smaller, multiplexed, ultra-compact units installed on the roof or inside the store. Each compact unit (**Figure 4.3**) contains its own air-cooled condenser, and is located very close to the display cases, and the heat reclaim coils (entrance hall, back of store, central air handler, etc.). Each cabinet is thermally insulated, ensuring an internal temperature of 21°C in winter and 32°C in summer, without any additional ventilation, make-up or air evacuation. During the winter, maintenance work is facilitated by a tight curtain surrounding the cabinet. The access paths on the roof are also cleared of snow by means of electrically heated carpets. This second phase of development was completed in 2001, and several new and retrofit supermarkets have already been equipped with this kind of cabinet. It was determined that this type of configuration reduce the total length of refrigeration piping and the number of fittings, thermal insulation and electric piping by up to 60 %, and the total refrigerant charge by 30 to 40 %, because of the proximity of display-cases and heat reclaim coils, and the smaller diameters and length of the refrigeration lines.

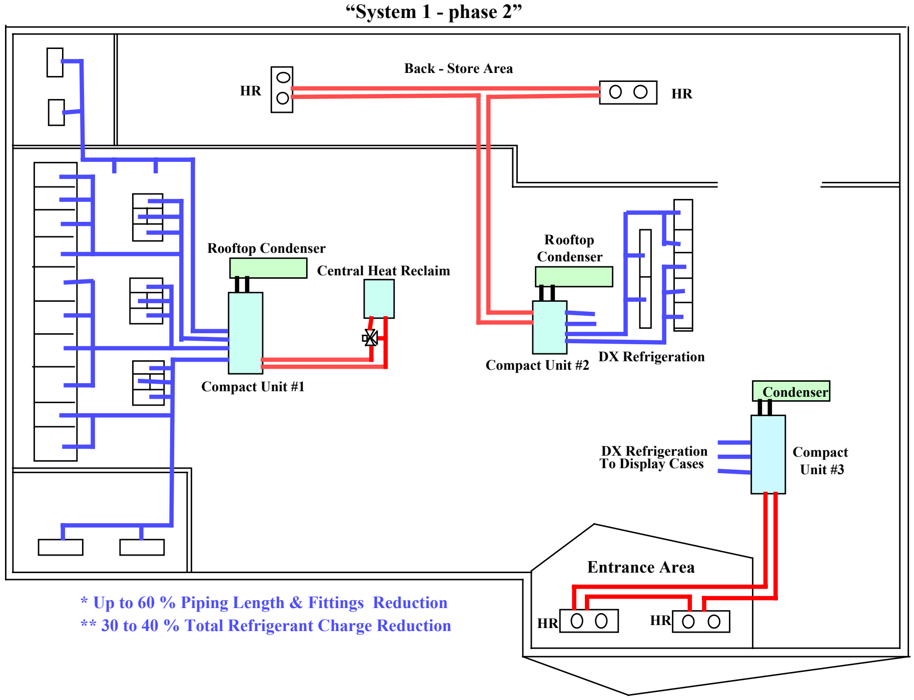

Figure 4.2 "*System 1 – phase2*": Second Step of Development

Figure 4.3 "*System 1 – phase2*": Compact Refrigeration Cabinet installed on Roof

The 3rd step of development ("*System 1–phase 3*") also aims to preserve the main prior innovations, but to reduce the refrigerant charge by 60 % or more, compared with the first concept. This new system contains similar compact, small cabinets, and two different types of plate heat exchangers (**Figure 4.4**):

- Heat exchanger (A) serves as a condenser during the summer and rejects heat into a secondary anti-freeze fluid loop.

- Heat exchanger (B) acts as a condenser during the winter, and rejects heat into a second secondary fluid loop that supplies heat reclaim coils for indoor air heating. Water-to-air or water-to-water heat pumps could be installed on the warm closed-loop. Each of the closed loops described contains circulation pumps, storage tanks and dedicated controls. However, direct expansion refrigeration is used in this case as the most economical means of refrigeration. At the same time, the previously developed high-speed defrosting method allows, with this configuration, an approximate 30% reduction in the number of refrigeration lines, and their associated fittings/valves, controls and quantities of refrigerant.

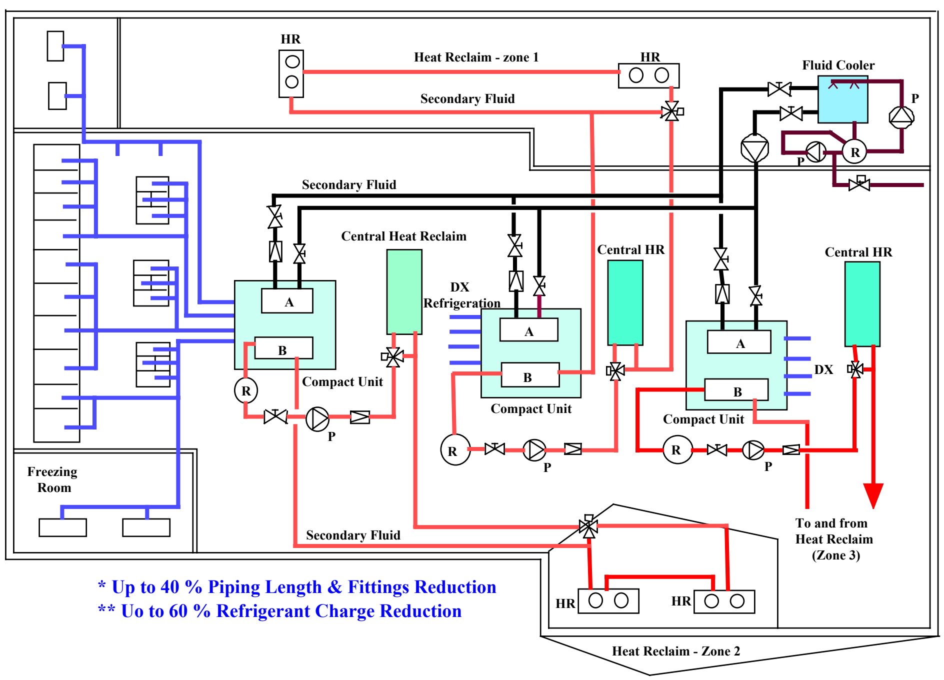

Figure 4.4 "*System 1 - phase 3*": Third Step of Development

The next step of development ("*System 1–phase 4*") aims to replace the direct expansion refrigeration technique by an appropriate closed-loop with cold secondary fluid. The objective is to reduce the total refrigerant charge by up to 90 % compared to the first "*System 1–phase 1*" system. In this case, several small compressor units will be installed on the roof, and both cold and warm circuits will use two separate secondary fluids: first, an anti-freeze solution (i.e. water-glycol) on the condenser side, and another one (i.e. potassium format) on the refrigeration side. Consequently, very small quantities of refrigerant (i.e. R-410a) will be confined in these de-centralized cabinets. The main disadvantages of this concept are in the defrosting techniques to be employed, and the low energy efficiency of the low temperature loops. However, neither heat reclaim, nor previously developed fast defrosting methods can be used in this case.

Table 4.1 summarizes the main features of the above-mentioned development steps in ”*System 1*”. The work already done aimed to maintain and further develop high-quality expertise in the province of Quebec, and also to create employment and move toward more environmentally friendly solutions, because they reduce the quantities of refrigerant and replace existing fossil fuel usage in heating (low TEWI systems).

Table 4.1 “*System-1*” – Summary of Development Steps

System Evolution	Main Particularities
“System-1” – phase 1 (1999)	• Central DX Multiplex (Mechanical Room) • Maximal Refrigerant Charge (4 000 à 5 000 lbs) • 100 % Heat Recovery (Central) • Fast Defrost • Geothermal Cooling • Additional Cost (38 000 square feet): CD$ 55 000
“System-1”– phase 2 (2001)	• Distributed DX Multiplex (2 to 4 Compressor Racks) • 100 % Heat Recovery (Decentralized) • Fast Defrost • Refrigerant Charge Reduction : 30 to 40 % • Piping, Fittings, Labour, etc. Reduction : up to 60 % • Additional Cost Reduction: up to 40 %
“System-1”– phase 3 (2003)	• Distributed DX Multiplex (2 to 4 Compressor Racks) • Secondary Fluid Condenser Side • 100 % Heat Recovery (with or without glycol-to air Heat Pumps) • Refrigerant Charge Reduction : up to 60 % • Piping, Fittings, Labour, etc. Reduction : 40 % • Additional Cost Reduction: up to 50 %
“System-1”– phase 4 (2005…)	• Distributed Indirect Refrigeration System (2 to 4 Compressor Groups) • Secondary Fluids both Condensers and Display Cases Sides • Refrigerant Charge Reduction : up to 90 % • Piping, Fittings, Labour, etc. Reduction : 60 % • Additional Cost Reduction: up to 60 %

”System 2”

The ”*System2-phase1*” has successfully met the main objective of eliminating natural gas usage for heating, but total refrigerant charge and the additional costs for the new heat recovery method are considered to be too high at the present time. These limitations were the main reasons for proceeding to the development of more advanced systems by using the previous innovations and expertise (**Table 4.2**). The new development of the ”*System2-phase2*”, begun in 2002, aims to perform total heat reclaim by a simplified, less expensive method, and also to reduce the refrigerant charge by up to 50 %, compared to phase 1 of the development, by using secondary fluid on the condensing side, and simplified refrigeration configurations.

Finally, the "*System2-phase3*" proposes the testing of more efficient secondary fluids for refrigeration (i.e., ice slurry) in order to reduce additional energy consumption for pumping, and to perform the defrosting cycle by employing a new, original method. It is expected that the amount of piping, valves, fittings and associated controls can be reduced by 40 %, and the total refrigerant charge by 30 % compared to phase 2. Moreover, the total heat reclaim method will work independently of the refrigeration system which will be able to operate with floating head pressures.

Table 4.2 "*System2 – phase1*": Development Summary

"System-2" Evolution	Main Particularities
"System-" – phase 1 (1999)	- Central DX Multiplex (Mechanical Room)- Large Refrigerant Charge (3 000 pounds) - 100 % Heat Recovery with Cascade Heat Pumps - Liquid Pumping and Injection - Premium Cost : CD$ 45 000
"System-2"– phase 2 (2002/2003)	- Central DX Multiplex (Mechanical Room) - New Method for 100 % Heat Recovery- New - Fast Defrost Technique - Improved Liquid Pumping - Refrigeration Charge Reduction: 50 % vs. phase 1 - Premuim Cost Reduction: about 75 % vs. phase 1
"System-2"– phase 3 (2003/2004)	- Central DX Multiplex (Mechanical Room) - Efficient Refrigeration Secondary Fluids - Piping Reduction: 40 % vs. phase 2 - Refrigerant Charge Reduction : up to 30 % vs. phase 2 - Premium Cost Reduction: up to 20 % vs. phase 2

5. CONCLUSIONS

This report describes and validates recent improvements made in Canada in the supermarket multiplex refrigeration area, with special focus on some operating parameters: heat recovery, defrosting, sub-cooling and liquid pumping performances, and compressor and store energy consumption and power demand comparison. The first improved system ("*System1-phase1*") consists of a method to maximize the heat reclaim capability for space and water heating, and a high-speed defrost concept, while the second ("*System2-phase1*") proposes an original heat recovery and air-conditioning/dehumidifying method involving intermediate heat exchangers and multifunctional refrigerant-to-air heat pumps. These systems represent the first phase of their respective developments, since they have already been further improved, particularly in order to reduce total refrigerant charge and leak hazards, and are readily available. Field experimental evaluation proved that fossil combustible consumption for space heating in supermarkets could be completely eliminated in cold climates. To achieve this, the first system ("*System1-phase1*") raises the head pressures/condensing temperatures during the heating demand periods, and simultaneously modifies liquid flow and controls to eliminate the current limitation of heat recovery capability. A modulating valve governed by the ambient temperature, employed since 1999, could optimize this process by varying head pressures with the actual heating demand and the outdoor temperature variation. On the other hand, the second system (“*System2-phase1*”) was capable of reducing discharge pressures to an optimum, constant level (about 29 % lower than the other system), without compromising its own heat reclaim method. Consequently, savings in compressor energy consumption were obtained, mainly due to the presence of four refrigerant-to-air heat pumps having coefficients of performance of 3 or more. Indoor air heated by heat reclaim coils was supplied at about 30°C (86 °F) during the coldest periods of the year. Consequently, indoor store temperatures varied around 23°C during the winter. Liquid pumping technology used by “System-2” also helped its heat reclaim technique by increasing the liquid pressure by about 26 psig during the cold periods when the head pressure normally decreases. Outside the heat reclaim demand periods, both systems operated with floating head pressures controlled by condenser fan liquid injection (“*System2-phase1*”) or geothermal cooling (“*System1-phase1*”). Total annual specific electrical energy consumption (vs. sales area) of improved supermarkets was reduced by 6 %, and the corresponding energy costs, including power demand cost, by 12.5 % (“*System1-phase1*”) and 17.2 % (“*System2-phase1*”) compared to those of a conventional store. Consequently, the energy cost savings were $20,400 /year (“*System1-phase1*”) and $28,800 /year (“System-2”). Simultaneously, total annual specific energy consumption of the refrigeration compressors represented 32.5 % (“*System1-phase1*”) and 34 % (“System-2” – including heat pumps) compared with each respective store’s total electrical energy usage, while in Canada, average consumption varies from 28 % to 39 %. “ *System1-phase1*” results were slightly penalized by the use of less efficient screw compressors and excessive fouling of groundwater heat exchangers and a lack of proper maintenance. “System-2” was also penalized by heat pump supplementary energy consumption, even when compensated for by lower head pressure operation. The new high-speed defrost method proposed by “ *System1-phase1*” reduced defrost cycle duration time by up to 85 %, compared with a conventional hot-gas defrost method and control. Immediate beneficial effects were better quality and less perishable food losses. The actual maximum deformation of a representative case evaporator during a fast defrost cycle represents about 56 % of the elastic limit of the current materials, but the long-term effects are not known, as well as those of the existing conventional systems. Some further improvements proposed by “ *System1*” technology aim to replace the HCFC-22 by a new, environmentally friendly fluid, to reduce the total refrigerating piping length, fittings and thermal insulation by 60 % or more, and finally, to lessen total refrigerant charge by 60 % or more.

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APPENDIX

Table A.1 "*System1-phase1*": Display Cases and Walk-in Boxes – Low Temperature Rack (SST - Saturated Suction Temperature; CST - Condensing Saturated Temperature)

No	**Dimensions**	**Description**	**Capacity**	**SST**	**CST**
-	**ft**	**-**	**Btu/h**	**°F**	**°F**
1	8'	Frozen Tomb	10 860	-25	105
2	12'	Frozen Tomb	13 380	-25	105
3	15 doors	Ice Cream	25 500	-25	105
4	12 doors	Grocery	18 000	-15	105
5	14 doors	Meat	21 000	-15	105
6A	5 doors	Fish	7 500	-15	105
6B	3 doors	Bakery	4 500	-20	105
7A	26'x15'x8'	Grocery	12 500	-25	105
7B	26'x15'x8'	Grocery	12 500	-25	105
8	20'x13'x8'	Meat	15 900	-20	105
9	11'x8'x8'	Bakery	8 750	-20	105
10	12'	Frozen Tomb	3 960	-25	105
11		Spare 10 %	18 000	-25	105
13A	8'+2 (bouts)	Ice Cream	12 200	-35	105
13B	8'+2 (bouts)	Ice Cream	12 200	-35	105

Table A.2 ”*System1-phase1*”: Display Cases and Walk-in Boxes – Medium Temperature (SST - Saturated Suction Temperature; CST - Condensing Saturated Temperature)

No.	**Dimensions**	**Description**	**Capacity**	**SST**	**CST**
-	**ft**	**-**	**Btu/h**	**°F**	**°F**
20	28	Fresh Meat	44 800	15	110
21	36	Fresh Meat	57 600	15	110
22A	14	Delicatessen	15 400	15	110
22B	14	Delicatessen	15 400	15	110
23A	14	Cheese	10 850	15	110
23B	14	Cheese	10 850	15	110
24A	8	Pizza	9 600	15	110
24B	8	Spare	10 600	15	110
24C	8	Spare	2 400	20	110
24D	8	Spare	2 400	20	110
25	36	Delicatessen	52 800	15	110
26A	8	Fich	9 600	15	110
26B	8	Bakery	9 600	15	110
27	12	Start-up	9 300	15	110
28	Spare 10 %	-	26 000	15	110
29	A/C	AC offices	79 800	45	110
30A	*30’x20’x10’*	*Meat*	*17 550*	20	110
30B	*30’x20’x10’*	*Meat*	*17 550*	20	110
31A	*10’x9’x8’*	*Fish*	*8 200*	20	110
31B	*10’x9’x8’*	*Delicatessen*	*8 200*	20	110
31C	*12’x8’x8’*	*Bakery*	*8 000*	20	110
32A	8	Meat	3 040	20	110
32B	24	Delicatessen Service	9 120	20	110
33A	*26’x18’x8’*	*Milk*	*11 300*	20	110
33B	*26’x18’x8’*	*Milk*	*11 300*	20	110
34A	36	Milk	47 700	20	110
34B	12	Wine	15 900	20	110
35	38	Milk	50 360	20	110
36	40	Milk	53 000	20	110
37	44	Beer	63 800	20	110
38A	8	Fruits	3 800	20	110
38B	20	Fruits	26 500	20	110
39	48	Fruits	37 440	20	110
40	*24’x15’x8’*	*Fruits*	*20 150*	20	110
41A	*608*	*Preparing*	*19 760*	*30*	110
41B	*608*	*Preparing*	*19 760*	*30*	110
42	Spare 10 %	-	41 000	20	110
43	-	Sub-cooling	41 084	40	110
TOTAL	-	-	901 514	-	-

Table A.3 ”*System1-phase1*: Characteristics of the Low Temperature Compressors

Compressor	SST	Model	Power	Capacity	THR	RLA	kW
-	**°F**	**-**	**hp**	**Btu/h**	**Btu/h**	**A**	**-**
Copeland	-25	3DA3-0600-TFE	6	28 661	44 545	10,5	4,6
Copeland	-25	4DA3-1000-TSE	10	48 307	75 860	17,5	8
Copeland	-25	4DL3-1500-TSE	15	64 685	101 098	20,9	10,7
Copeland	-25	3DS3-1000-TFE	10	46 399	71 891	16,8	7,4
Copeland	-35	3DF3-0900-TFE	9	29 470	49 160	16,5	5,7
TOTAL	-	-	50	217 522	342 554	82,2	36,4

Table A.4 ”*System1-phase1*: Characteristics of the Medium Temperature Compressors

Compressor	TSS	Model	Power	Capacity	THR	RLA	kW
-	**°F**	**-**	**hp**	**Btu/h**	**Btu/h**	**A**	**-**
Carlyle	15	06TAH078C2EA	40	323 173	408 548	51,4	37,2
Carlyle	15	06TAH078C2EA	40	323 173	408 548	51,4	37,2
Carlyle	15	06TAH078C2EA	40	323 173	408 548	51,4	37,2
TOTAL	-	-	120	969 519	1 225 644	154,2	111,6

Table A.5 ”*System1-phase1*: Characteristics of the Air-Cooled Condensers

	Low Temperature	Medium Temperature
Model	CLD057	FDFCT-180-R
Capacity	424 800 Btu/h	1 800 000 Btu/h
AC Circuit; Control	8,4 A; 575 V/3 ph; 120 V/30 A	575 V/3 ph; 120 V/30 A

Table A.6 ”*System 1-phase 1*”: Example of Monitored Parameters

Type of Parameter	Observation
Store Total Power (kW)	-
Compressor’s Power (kW)	Both DT and MT Compressor
Total Electrical Power (kW)	LT and MT Compressor
Suction Pressures (psig)	Both DT and MT Refrigeration Line
Discharge Pressures (psig)	Both DT and MT Refrigeration Line
Suction Temperatures (°C)	Both DT and MT Refrigeration Line
Discharge Temperatures (°C)	Both DT and MT Refrigeration Line
Liquid Temperatures (°C)	Both DT and MT Refrigeration Line
Liquid Pressures (psig)	Both DT and MT Refrigeration Line
Cold Water Temperatures (°C)	Entering/Leaving Storage Tank
Groundwater Temperature (°C)	Entering/Leaving Plate Heat Exchanger
Control Valves Status (on/off)	Defrost, Heat Recovery, Flushing, etc.

Table A.7 ”*System 1-phase 1*”: Example of Calculated Thermodynamic and Energy Parameters

Type of Parameter	Observation
Mass Enthalpies (kJ/kg)	Both DT and MT Line (Suction, Discharge, Liquid)
Total Refrigerant Flow-rate (kg/s)	Both DT and MT Refrigeration Line
Title of Two-phase Refrigerant (%)	Both DT and MT Refrigeration Line
Refrigeration Power (kW)	Both DT and MT Refrigeration Line
Condensing Power (kW)	Both DT and MT Refrigeration Line
Refrigeration Load (kWh)	Both DT and MT Refrigeration Line
Energy Rejected from Condensers (kWh)	Both DT and MT Refrigeration Line
Refrigeration Efficiency	Both DT and MT Refrigeration Line
Saturated Suction (SST) and Discharge (SDT) Temperatures (°C)	Both DT and MT Refrigeration Line
Compressor Consumed Energy (kWh)	Both DT and MT Refrigeration Line
Energy Consumption (kWh)	Both DT and MT Refrigeration Line
Operating Time: Heat Reclaim, Defrost; Condenser; Geothermal	Space and Water Heating Systems
Defrost Cycle Status and Operating Time (sec/hr; hrs/day)	Fresh Meat Case and Cold Room; Ice Cream

Table A.8 "*System2 – phase1*": Characteristics of Dual (Low & Medium) Temperature Compressors (THR - Total Heat Rejection; EER – Energy Efficiency Ratio)

Compressor	TSS	Model	Condensing temperature	SUCTION Capacity	THR	EER
-	**°F**	**-**	**°F**	**Btu/h**	**Btu/h**	**-**
COP/Discuss (22) L/T	-35	2DA3-060E-TFE-200	100	16 000	27 270	5,07
COP/Discuss (22) L/T	-25	3DB3-075E-TFE-200	100	37 000	55 490	6,76
COP/Discuss (22) L/T	-20	2DL3-040E-TFE-200	100	25 000	36 380	7,07
COP/Discuss (22) L/T	-20	3DS3-100E-TFE-200	100	56 000	82 320	7,1
COP/Discuss (22) L/T	-20	3DS3-100E-TFE-200	100	56 000	82 320	7,1
COP/Discuss (22) L/T	-20	4DA3-100E-TSE-200	100	59 000	88 510	6,93
COP/Discuss (22) M/T	15	3DB3-100E-TFE-200	100	107 000	136 100	12,71
COP/Discuss (22) M/T	15	3DS3-150E-TFE-200	100	142 000	181 100	12,32
COP/Discuss (22) M/T	15	4DA3-2000-TSE-200	100	150 000	190 800	12,54
Total	-35	-	-	16 000	-	-
	-25	-	-	37 000	-	-
	-20	-	-	195 000	-	-
	15	-	-	399 000	-	-
Total Unit	-	-	-	-	880 200	-

Table A.9 "*System2 – phase1*": Characteristics of Medium Temperature Compressors (THR - Total Heat Rejection; EER – Energy Efficiency Ratio)

Compressor	TSS	Model	Condensing Temperature	SUCTION Capacity	THR	EER
-	**°F**	**-**	**°F**	**Btu/h**	**Btu/h**	**-**
COP/Discuss (22) M/T	20	2DA3-075E-TFE-200	100	83 000	103 600	13,78
COP/Discuss (22) M/T	20	3DB3-100E-TFE-200	100	120 000	149 300	13,82
COP/Discuss (22) M/T	20	3DL3-150E-TFE-200	100	159 000	199 200	13,34
COP/Discuss (22) M/T	20	4DH3-2500-TSE-200	100	222 000	276 800	13,69
Total	20	-	-	584 000	-	-
	-	-	-	-	729 000	-

Table A.10 “*System2 – phase1*”: Display Cases and Walk-in Boxes –Low Temperature

No.	**Description**	**Capacity**	**SST**
-	**-**	**Btu/h**	**°F**
13	Ice Cream	15 720	- 35
11	Pastry and Frozen Meat	5 280	- 25
4	Ice Cream 5 doors	25 500	- 25
10	Bakery and Frozen Crustaceous	16 680	- 20
8	Frozen Meat and Fish	21 720	- 20
3	Frozen Products	10 860	- 20
5	Frozen Products	23 160	- 20
6	Frozen Meat	17 640	- 20
7	Frozen Bakery	20 760	- 20
9	Frozen Meat and Fish	27 000	- 20
2	Frozen Products	24 000	- 20
1	Frozen Products	22 500	- 20
12	Spare	7 894	- 20

Table A.11 "*System2 – phase1*": Display Cases and Walk-in Boxes – Medium Temperature (SST : Saturated Suction Temperature)

No.	**Description**	**Capacity**	**SST**
-	**-**	**Btu/h**	**°F**
41	Combined Service. "BAS"	12 320	10
41	Shelves of Prepared Meals	7 200	15
41	Flowers	6 240	20
52	Prepared Meals	5 080	15
52	Delicatessen and mets préparés	12 420	20
31	Fresh Meat	30 280	15
30	Fresh Meat	44 800	15
32	Fresh Meat, Fish and Crustaceous	33 960	15
33	Delicatessen	52 800	15
34	Delicatessen	38 400	15
38	Prepared Meals	43 200	15
37	Pizza	31 200	15
38	Prepared Meals	11 670	20
35	Wine	26 500	20
35	Delicatessen	26 500	15
40	Vinaigrettes	37 100	20
46	Fruits and Vegetables	43 680	20
36	Fruits and Vegetables	18 720	20
42	Fruits and Vegetables	44 160	20
50	Milk Products	50 340	20
49	Milk Products	31 800	20
48	Milk Products	47 700	20
53	Milk Products	71 520	20
51	Beer	68 900	20
44	Milk	22 650	20
43	Fresh Meat and Fish	39 090	20
45	Bakery; Fruits and Vegetables	36 990	20
47	Preparation Area	32 400	30

Table A.12 "*System2 – phase1*": Example of Monitored Parameters

Type of Parameter	Observation
Store Total Power (kW)	-
Compressor Power (kW)	Each Low and Dual Compressor
Rack Power (kW)	LT and MT Compressor
Heat Recovery and Sub-Cooling Heat Pump Power (kW)	Each Unit
Suction Pressures (psig)	Very Low, Low and Medium
Common Discharge Pressures and Defrost Discharge (psig)	Each Rack
Suction Temperatures (°C)	Very Low, Low and Medium
Common Discharge Temperatures (°C)	DT/MT
Defrost Discharge Temperatures (°C)	DT/MT
Liquid Defrost Return Temperatures (°C)	DT/MT
Liquid Temperatures (°C)	DT/MT
Liquid Pressures (psig)	DT/MT
Liquid Pump Pressures (psig)	Entering/Leaving each LP
Control Valves Status (on/off)	Defrost, Heat Recovery, Flushing, etc.
Hot Gas Temperature at Plate Heat Exchangers Outlet (°C)	Each Unit
Natural Gas Valve Status (on/off)	Each Unit

Table A.13 "*System2 – phase1*": Example of Calculated Parameters

Type of Parameter	Observation
Mass Enthalpies (kJ/kg)	LT/DT (Suction, Discharge and Liquid0
Compression Mass Enthalpy Variation (kJ/kg)	LT/DT
Saturated Suction and Discharge Temperatures (°C)	Low and Dual Temperature
Compressor Energy Consumption (kWh)	Each Low and Dual Compressor
Total Energy Consumption (kWh)	Each Low and Dual Compressors
Heat Pumps Operating Time (sec/hr; hrs/day)	Space and Water Heating Systems
Defrost Status and Duration (sec/hr; hrs/day)	Fresh Meat Case and Cold Room; Ice Cream

Table A.14 "System *CON*" : Display Cases and Walk-in Boxes – Low Temperature

Circuit	**Description**	**Capacity**	**Operation Temperatures**	
-	**-**		**Evaporation**	**Condensing**
		Btu/h	**°F**	**°F**
1	Different Products	25 500	- 25	105
2	Ice Cream	22 100	- 25	105
3	Different Product	11 500	- 25	105
4	Fish	25 680	- 25	105
5	Meat	2 640	- 25	105
7	Meat	19 260	- 25	105
8	Meat	9 000	- 25	105
9	Pastry	5 700	- 25	105
Total	-	121 380	-	-

Table A.15 "*System CON*" : Display Cases (15 °F - Medium Temperature)

Circuit	Description	Capacity	Operation Temperatures	
-	-		Evaporation	Condensing
		Btu/h	°F	°F
19	Fresh Meat	57 600	15	110
11	Delicatessen	16 140	15	110
12	Fish	9 600	15	110
13	Mets cuisinés	24 000	15	110
14	Delicatessen	14 400	15	110
15	Cheese	10 800	15	110
20	Meat	23 400	15	110
Total	-	155 940	-	-

Table A.16 "*System CON*" : Display Cases (20 °F - Medium Temperature)

Circuit	Description	Capacity	Operation Temperatures	
-	-		Evaporation	Condensing
		Btu/h	°F	°F
17	Pastry	9 600	20	110
18	Farm Delicatessen	10 500	20	110
18	Milk	63 600	20	110
20	Beer	21 200	20	110
21	Jus/Vegetables	37 100	20	110
21	Refrigerated Table	5 600	20	110
22	Fruits/Vegetables	40 300	20	110
23	Milk	11 500	20	110
24	Delicatessen	4 500	20	110
25	Fruits/Vegetables	10 000	20	110
26	Beer	19 000	20	110
27	Meat	24 505	20	110
28	Fruits/Vegetables	10 800	20	110
Total	-	268 205		

Table A.17 “*System-CON*”: Monitored Parameters

Type of Parameter	Observation
Compressor Power (kW)	LT and MT Compressors
Electrical Oven Power – Cooking (kW)	Ovens #1, 2 and 3
Suction Temperature (°C)	Low and Medium Temperature Racks
Common Discharge Temperature (°C)	Each Rack
Hot Gas Temperature – Heat Reclaim (°C)	Entering and Leaving HR Coil
Defrost Valves Status (on/off)	Freezing Products; Fresh Meat

Based on the measured parameters, several thermodynamic, energy and performance calculations were performed (
Table).

Table A.18 “*System-CON*”: Thermodynamic, Energy and Performance Calculated Parameters

Type of Parameter	Observation
Energy Consumption of Compressor (kW)	Low and Medium Temperature Compressors
Total Refrigeration Consumed Energy (kWh)	Low and Medium Temperature Compressors
Cooking Energy Consumption (kWh)	Each Electrical Oven
Duration and Defrost Status (sec/hr; hrs/day)	Four Refrigeration Display Cases

IEA Heat Pump Programme, Annex 26:
Advanced Supermarket Refrigeration/Heat Recovery Systems
Country report from Denmark.

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Introduction.

Since the agreement on the Montreal protocol (1987) and its amendments with focus on ozone destruction fluids (ODP) there has been an increasing research and development of new refrigerants. As new refrigerant has been proposed new developed synthetic fluids, HFC, and a revival of the old natural refrigerants as ammonia, carbon dioxide, water and hydrocarbons (HC).
In Denmark the Montreal protocol has resulted in a tax on CFC from 1.1.1989 followed by a ban on use of CFC-refrigerant from 1.1.1995 and a ban on use of HCFC from 1.1.2001.
There is in Denmark concern about the replacement of CFC and HCFC with HFC due to the unknown influence of the HFCs on the environment in the long term, especially the contribution to the Green House Effect (GWP). Many people argued, that HFC only was an intermediate solution and it would be wiser to use as replacement for CFC and HCFC the natural refrigerants.
In the opening speech to the IIR conference on “Applications for Natural Refrigerants”, Aarhus, Denmark (1996) the Danish Minister for Environment and Energy, Svend Auken, in his conclusion said: “It is therefore my sincere hope that in ten years’ time, not a single fridge, freezer or cooling plant is being built in Denmark that requires HFCs or other greenhouse gases.” /1/
In Denmark emission of carbon dioxide (CO_2) had been taxed from 15.5.1992. The agreement on the Kyoto protocol has increased the concern about the GWP and Denmark has from 1.1.2001 extended the CO_2 taxation to industrial greenhouse gasses (HFCs, PFCs and SF_6) based on their GWP. The tax is US cents 1.27 (DKK 0.10) pr. kg CO_2 with CO_2 by definition having the GWP of 1. There is an upper limit of the tax of US$ 50.95 (DKK 400) pr. kg. /2/
Recently an announcement (in Danish ;Bekendtgørelse) /3/ has been issued which forbid the use of certain industrial warming potential gasses which includes the HFC´s. From January 1.th, 2006 the use of HFC’s is no longer allowed used in new products. Exemptions are e.g. the use in refrigeration plants, heat pumps, air conditioning plants and dehumidifiers with refrigerant charge between 0.15 kg and 10 kg and compact heat recovery plant factory assembled with charge less than 50 kg. According to this rule the conventional multiplex refrigeration system in supermarkets cannot be built after January 2006.
Based on what has been stated about it is easy to understand why the Danish research has focused on natural refrigerant and means to make the refrigeration plant more energy efficient.

Background.

The refrigeration systems in supermarket has traditionally been built as either multiplex refrigeration system or self-contained cases with direct evaporation of the refrigerant in the displace cabinets. As refrigerant has been used R12 for cooling and R502 or R22 for freezing. Due to the phase out of HCFCs a new refrigerant is needed. One can either use HFCs e.g. R404A and R507 or some of the natural refrigerants e.g. ammonia (NH_3, R717), HCs as propane (C_3H_8, R290) and propylene (C_3H_6, R1270) or carbon dioxide (CO_2, R744). Due to the Danish regulations only the natural refrigerants seems to be a viable refrigerants for the future. Except CO_2 the natural refrigerant mentioned above are either flammable and/or toxic. This eliminates the use of direct evaporation of the refrigerant in the salesroom so one has to use indirect systems with the refrigeration plant placed outside the sale area in a safe place and with as small amount of refrigerant as possible. At the same time the energy efficiency shall be as high as possible to minimize the emission of green house gasses (minimized TEWI). This limits the use of CO_2 due to its low critical temperature, but CO_2 can be used as refrigerant in a cascade plant in the freezers or as a secondary refrigerant. Used as secondary refrigerant CO_2 is a good choice because it has a very low viscosity, high heat transfer coefficient and the cooling performance is rather high when used as a two-phase fluid. The only problem using CO_2 is the relative high working pressure compared to the pressure of the other refrigerants.

The shift to use natural refrigerant was started already in 1994 when an old CFC (R11) centrifugal chiller to air-conditioning in a department store in the centre of Copenhagen (Magasin) was replaced with a roof mounted ammonia chiller. The new chiller is composed of two unit each with a cooling capacity of 900 kW. The amount of ammonia in each chiller is app. 100kg.

Demonstration plants.

In Denmark the authorities prescribe chilled food to be stored at temperature below +5ºC and frozen food to be stored at temperature below -18ºC. Hereafter these two temperature are referred to as high temperature and low temperature.

Indirect system with ammonia.

Already in 1995 a demonstration project at the Schou Epa supermarket, now Kvickly, in Roskilde, a fairly large supermarket according to Danish standard, has been carried out. The refrigeration plant was changed from a traditional refrigeration plant using CFC/HCFC (R12, R22 and R502) with direct expansion in the display cabinets/rooms to an indirect refrigeration plant using ammonia (NH_3) as primary refrigerant and Tyfoxit as secondary refrigerant performing the cooling of the display cabinets/rooms. The old installation was a self-contained system with 34 separate units. About 30% of the plant was connected to a main roof-mounted condenser outside the building, whereas the remaining part of the plant was condenser units with a local condenser located beside the compressor. Defrosting was electrical with defrost timer placed at every cabinet/room.
The new system is an indirect system due to the toxicity of ammonia. The cooling capacity of the plant is approximate 100 kW at the high display/room temperature and 40 kW at the low display/room temperature. The new plant is a two-stage ammonia plant with air-cooled condenser. The ammonia plant is mounted on the roof. All parts containing ammonia with exception of the air-cooled condenser are placed in a 20 feet

container. Defrosting is performed by warm brine. The brine is heated by sub-cooling the refrigerant from the condenser.
The evaporation temperatures are -12ºC and -33ºC and the amount of ammonia is 75 kg.
Figure 1 shows the principal layout of the new plant.

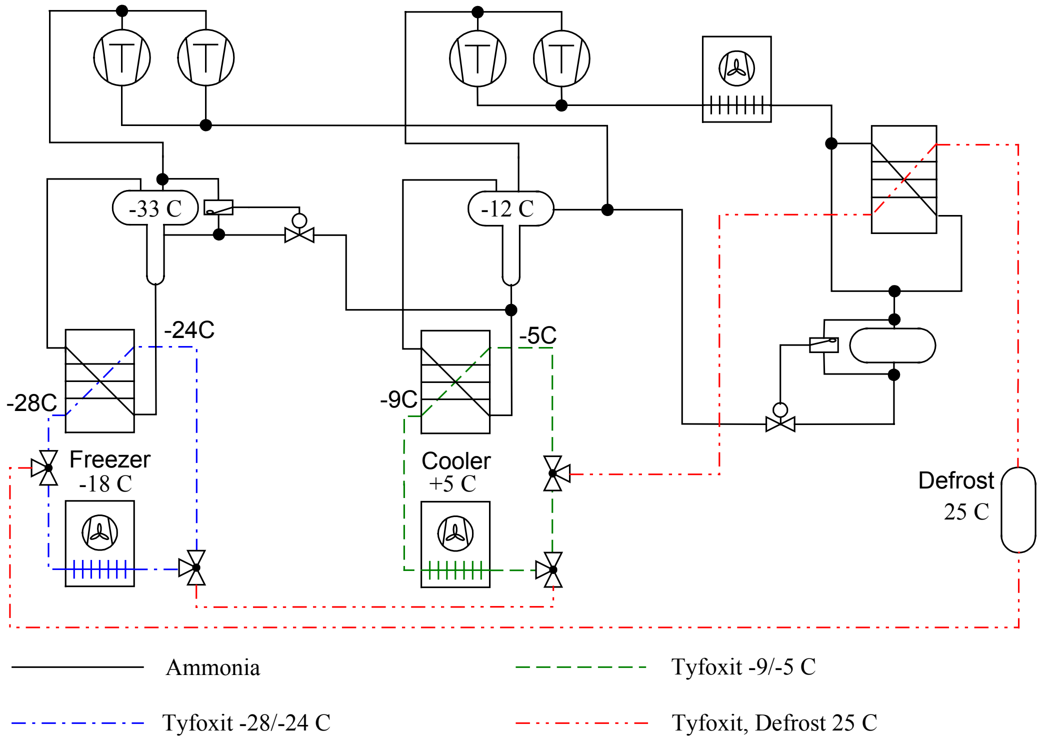

Figure 1. Principal construction of the indirect two stage ammonia plant at Schou Epa

Measurement has been performed on the old plant in the second half of 1995 and on the new plant in the second half of 1996. Correction has been performed to account for the difference in working conditions. A comparison of the energy consumption shows an energy saving of 35% for the new plant compared to the old plant. Part of this saving is due to elimination of electrical defrost, but it is concluded that the old plant has been working at not-expedient working conditions e.g. too high condenser pressure or too low evaporation temperature because of lack maintenance, lack of cleaning the evaporators and condensers and low air exchange for some of the condensers.
The plant and the results is described in /4,5/.
This demonstration project has been carried out in collaboration between the Danish Technological Institute (DTI), Sabroe + Søby A/S (now York Køleteknik), Danfoss A/S, LR Industri A/S, Schou Epa K/S (now Kvickly) and Grundfos A/S with economical support from the Ministry of Environmental and Energy and the Danish Environmental Protection Agency.

Cascade system with propane/carbon dioxide (Indirect/direct refrigeration).

Due to the toxicity of ammonia, one wishes to use another refrigerant than ammonia in supermarket refrigeration plant. Instead of ammonia propane, which is only flammable, can be used. Due to the flammability propane is not allowed used in the salesroom, and indirect system must be used. Instead of using a two-stage plant with indirect refrigeration both in the coolers and freezers, a cascade plant with carbon dioxide in low temperature cycle can be used with direct evaporation in the freezers while the refrigeration in the coolers is performed with a traditional secondary refrigerant.

To prove the ability of this concept the traditional refrigeration plant in a small Danish supermarket ("Dagli'Brugsen" in Odense) was replaced with a cascade plant. Propane is used at the high temperature level (-14°C /+30°C, evaporation /condensation temperature), while carbon dioxide (CO_2) is used at the low temperature level (–32°C /–10°C). The propane condenses directly in an air-cooled condenser mounted at the roof of the supermarket. The two refrigerants exchange heat in the cascade exchanger in which propane evaporates during dry expansion while CO_2 condenses. CO_2 is used directly in the low-temperature cabinets and cold stores of the supermarket. Glycol, cooled in a second cascade heat exchanger by evaporating propane, is pumped in a closed system to the high-temperature cabinets and cold stores. Figure 2 shows the principle layout of the plant.

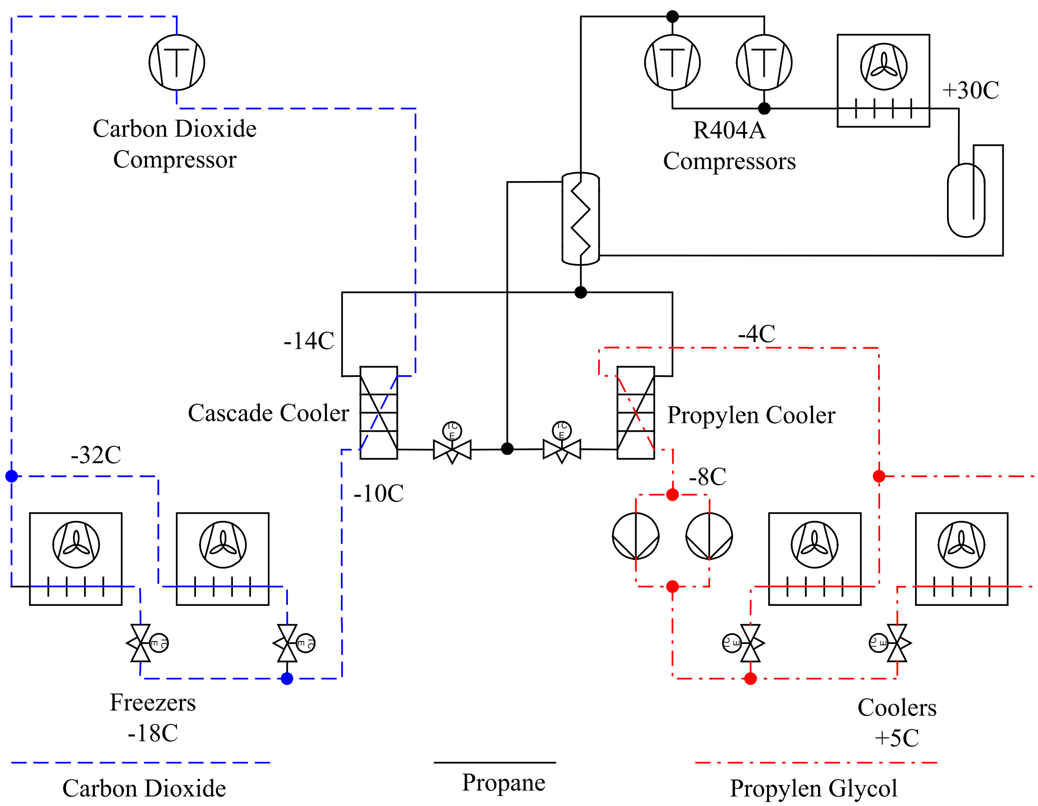

Figure 2. Principle layout of the direct/indirect system in Odense and Beder.

The size of the plant is based on a capacity of 10 kW at high temperature and 6 kW at low temperature.

The propane system is constructed with steel pipes and most of the other components are also made of steel. This has been necessary for the approval of the system at that time. All of the components used for the propane system are commercially available. In the future it is expected that the propane system can be constructed with copper pipes.

The carbon dioxide system is constructed with copper pipes. The compressor is a semi-hermetic prototype compressor. The system design pressure is 19 bar at the suction side and 32 bar at the discharge side. Relief valves are mounted on the suction and discharge side of the compressor as well as on each heat exchanger.
The cascade cooler between evaporating propane and condensing carbon dioxide is a 45 bar plate heat exchanger.
All evaporators are dry expansion evaporators.
The total refrigerant charge is 6 kg propane and 6 kg carbon dioxide.
All components containing propane except the air- cooled condenser together with the carbon dioxide compressor and circulations pumps are set up in a in a tight and ventilated box, which serves as a separate engine room. Mechanical ventilation ensures a constant pressure below the atmospheric pressure. The box is equipped with both propane and carbon dioxide detector.
The plant was set into operation May 26, 2000. During the summer 2000 data has been collected from the system (temperature, pressure and total energy consumption). The refrigeration system has met the desired conditions and keeps -20°C in the freezing display cabinets and +2°C cooling cabinets.
The total energy consumption has been measure for both the old and the new plant and a reduction of 10% has been achieved. But it has to be emphasised the old plant was worn-out.
In the autumn 2000 and spring 2001 further more detailed measurement has been performed in connection with a master project. From these measurements it can be conclude that the COP for the propane cycle is 3.3 and the COP for the carbon dioxide cycle 2.5. The reason for the low COP for the carbon dioxide system is due to a much to big compressor. The compressor is equipped with speed control, but even at the lowest permissible speed the capacity of the compressor is to high, so it is necessary to have the compressor working intermittently (on/off). The measured cooling load in the period is 2.4 kW but the dimensioning cooling load is 6 kW.
Based on the result from this demonstration plant is concluded that the additional cost for a propane/carbon dioxide cascade plant for a medium sized Danish supermarket (30 kW freezing load and 60 kW cooling load) will amount to approximate 15% of the total installation.
Based on the result form the first demonstration plant a new cascade plant using propane/carbon dioxide has been built in a medium-size supermarket (“Fakta”, Beder). The chosen supermarket is part of a chain (“Fakta”) with app. 250 shops so it is possible to compare the propane/carbon dioxide plant with a newer traditional plant with the same cooling capacity, 21 kW at -10°C (coolers) and 10 kW at -32°C (freezers).
Propane is used at the high temperature level (–14°C /25°C), while CO_2 is used at the low temperature level (–32°C /–10°C). The propane condenses directly in an air-cooled condenser at the roof of the supermarket. The two refrigerants exchange heat in the cascade exchanger in which propane evaporates during dry expansion while CO_2 condenses. CO_2 is used directly in the low-temperature cabinets and cold stores of the supermarket, while propane heat exchanges to an indirect system with glycol. The glycol is pumped in a closed system to the high-temperature cabinets and cold stores. The system lay-out it shown in Figure 2.
Suction gas cooled semihermetic compressors are used for both propane and CO_2. The propane compressors are equipped with an oil pump, while the CO_2 compressor is splash lubricated.

The refrigeration system is built on a rack (compressor rack), where compressors, cascade exchanger, brine cooler, brine pump, valves and vessel are mounted. The aggregate is set up in a separate machine room from which pipes connect the cabinets to the system. The components, which contain propane, are built in a ventilated box. The propane system operates with two evaporators: one cascade exchanger and one brine cooler. The cascade exchanger and the brine cooler are both plate heat exchangers. On both exchangers electronic expansion valves are used for injection of refrigerant. The propane system is furthermore built without an oil separator. A mineral oil is used which is fully miscible with propane so that the oil returns from the system to the compressors. An identical oil level between the compressors is ensured by means of mounted pressure equalisation pipes.

The CO_2 system is constructed as a traditional refrigeration system. The cascade heat exchanger functions as the condenser of the CO_2 system, where CO_2 condenses against evaporating propane. The liquid feed to the evaporators located in the low-temperature cabinets and cold stores of the supermarket. Pulse width modulated As injection valves are used, which maintained a superheat of 8 K. The gas is then sucked back to the compressor. The system does not have an oil separator.

The amount of oil entering the system will due to fine solubility with the refrigerant (ester oil/CO_2) and relatively high gas velocities be transported with the refrigerant back to the compressor. The first system in Odense had an oil separator, but there has not been observed any problems by not having an oil separator.

In contrast to the system in Odense both the carbon dioxide and the propane circuits are constructed with copper tubes. The glycol part is made of ABS pipes.

The plant has been set in operation in the autumn 2001. Based on measurement of the energy consumption for refrigeration in Beder and 8 reference “Fakta” supermarkets of equivalent size using conventional R404A refrigeration plant it can be concluded, that the refrigeration plant in Beder is neutral compared to modern and optimised conventional systems with R404A.

The carbon dioxide/propane system and the chosen installation in “Fakta” Beder amount an additional investment of approximate 20% compared to conventional R404A plants.

For details se Appendix A, Appendix B and /6 in Danish/.

The demonstration project at “Dagli’Brugsen” has been carried out in collaboration between Danish Technological Institute (DTI), Super Køl and FDB (now Coop Denmark) with economical support from the Danish Energy Agency and the demonstration project at “Fakta”has been carried out in collaboration between Danish Technological Institute (DTI), Super Køl and Fakta with economical support from Danish Environmental Protection Agency.

Cascade system with R404A/carbon dioxide (Direct refrigeration).

The use of a secondary refrigerant at the high temperature (+5ºC air temperature) implies an extra heat exchanger compared to direct evaporation. The evaporation temperature of the primary refrigerant has therefore to be lower, which increases the compressor power. Normally used secondary refrigerants also exhibit poor heat transfer properties. This disadvantage can be eliminated using carbon dioxide as refrigerant also at the high temperature.

The supermarket chain ISO has in a new supermarket (ISO Friheden) installed a refrigeration plant according to this concept /7,8/. The plant has 190 kW cooling capacity and 60 kW freezing capacity. Furthermore is the condenser heat utilized to supply the shop with heat and sanitary hot water. No further heat input is needed.

The refrigeration plant is a cascade plant with R404A in the high temperature stage and carbon dioxide in the low temperature stage. The freezers are cooled by direct evaporation of carbon dioxide at sufficient low saturation temperature. The vapour is then compressed to the saturation pressure in the cascade cooler by a carbon dioxide compressor. The coolers are also cooled by direct evaporation of carbon dioxide circulating between the cascade cooler and the evaporators by means of circulating pump. This means, that the carbon dioxide is condensed in the cascade cooler at a temperature equal to the evaporation temperature in the coolers.
The principle layout of the plant is shown in Figure 3.

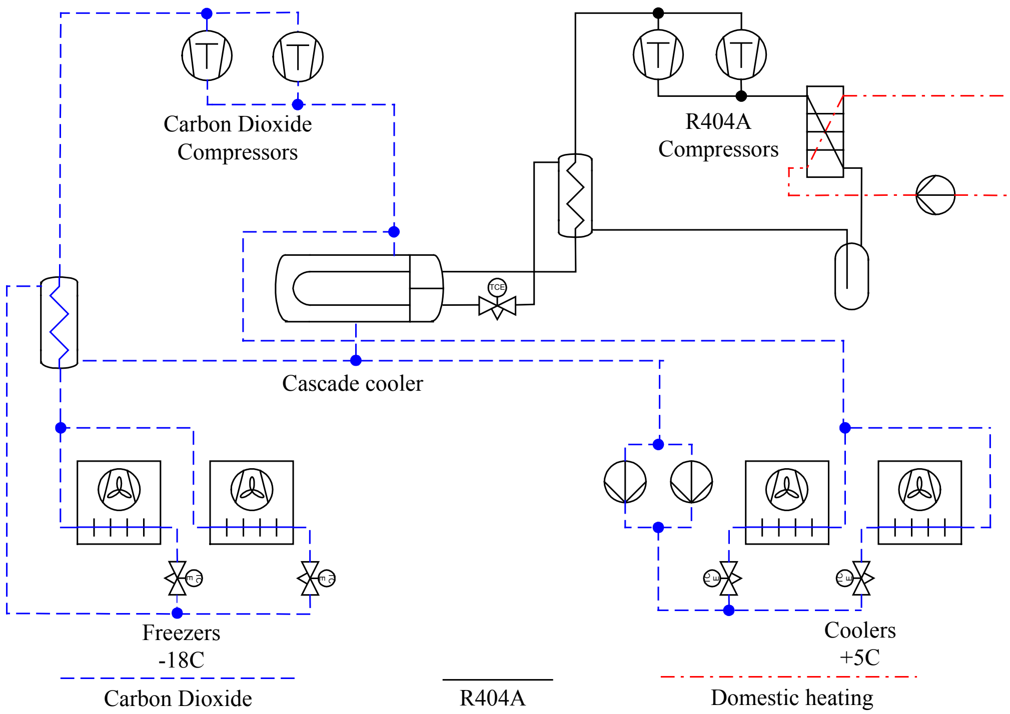

Figure 3. Principal layout of the direct expansion plant at ISO.

In this case the high temperature stage uses R404A instead of e.g. propane or ammonia. This is due to the placement of the cooling plant in the basement of the shop, where it is difficult to get permission to install flammable/and or toxic refrigerant. Due to the design with only carbon dioxide in the coolers and freezers instead of the usual direct evaporation of R404A, the amount of R404A has been reduced to only 120 kg, approximate 10% of a normal charge. Taking into account the Danish tax on R404A this reduction reduces the investment in refrigerant with approximate US$ 37,000. (DKK 290,000). The total investment in the refrigeration plant is approximate 10% higher than a usual HFC-plant, but the payback time for this extra investment is very short due to higher efficient of performance and reduced expenses to refrigerant due to leakage.
All components in the carbon dioxide circuit are designed for 35 bar and the test pressure has been 50 bar.
The entire refrigeration plant – including the display cases and the cool rooms - is equipped with an electronic control system that enables remote monitoring and control.

The system is connected to a central alarm processing facility.
The plant has been operating since February 17, 2002. Data are collected and compared with data from other ISO shops in the area with conventional refrigeration system. A comparison with the other shops shows a significant lower energy consumption /9/.
This project has been carried out in collaboration between ISO, York Køleteknik and Findan. Data acquisition is performed by Super Køl in Odense

Conclusion.

Based on the result from the demonstration plants mentioned about it can be concluded, that it is possible to built refrigeration plant using natural refrigerants to supermarkets with an energy consumption equal to or lower to the conventional HFC-plants. The investment will be 10% to 15% higher for the plant with natural refrigerants than the conventional HFC plants.

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1US$ = 784.93 DDK

Appendix A.

Refrigeration Systems in Supermarkets with Propane and CO_2 – Energy Consumption and Economy

By Kim Christensen, Danish Technological Institute, Energy

In 2000, a new refrigeration technology was implemented in the small supermarket "Dagli'Brugsen" in Odense, Denmark. The system has been in operation since then and has demonstrated that it is possible to build cascade systems based on propane and CO_2. The small supermarket was however very atypical for which reason it was difficult within the frames of the project of that time to provide qualified conclusions concerning the energy consumption and economy.

It was thus decided to carry out a similar project – in a typical supermarket – in order to provide comparative data from new conventional systems based on direct expansion and with modern scroll compressors. The project was carried out in collaboration with Superkøl, the Danish Technological Institute and Fakta. The project was furthermore partly financed by the Danish Environmental Protection Agency.

The supermarkets of the Fakta retail chain (in total 238 supermarkets) have the advantage that the newest ones are very standardised and a statistical comparison therefore can be carried out without great uncertainty. The project group chose to take its starting point in a new building in Beder, Denmark, where the new Fakta supermarket was opened in June 2001.

The previous project focused on investigation of function and operation, whereas this project focused on documentation of the system with regard to energy consumption and economy. The unanswered questions of the industry in these areas have to be answered. I hope that this article might be of help in doing so.

Improvements Compared to the System in Odense

Experience from the project in Odense made the project group focus on several items:

1. The propane part of the new system in Beder should be based on copper pipes and conventional components where the propane part of the system in Odense was welded in steel pipes.
2. The brine system should be optimised with regard to construction (use of flexible tubes, pre-insulated pipes, flow valves and more intelligent control).
3. Use of frequency converter and CO_2 compressor only, not propane compressors.
4. New and sturdier control of the propane injection for the cascade exchanger is investigated.
5. Energy consumption is compared with identical supermarkets in the retail chain.
6. Investment and service costs are documented and compared to other supermarkets in the retail chain.

System Construction

The system, which is set up in Fakta Beder, is constructed as a cascade system. Propane is used at the high temperature level (–14/25°C), while CO_2 is used at the low temperature level (–32/–10°C). The propane condenses directly in an air-cooled condenser at the roof of the supermarket. The two refrigerants heat exchange in the cascade exchanger in which propane evaporates during dry expansion while CO_2 condenses. CO_2 is used directly in the low-temperature cabinets and cold stores of the supermarket, while propane heat exchanges to an indirect system with glycol.

The glycol is pumped in a closed system to the high-temperature cabinets and cold stores.

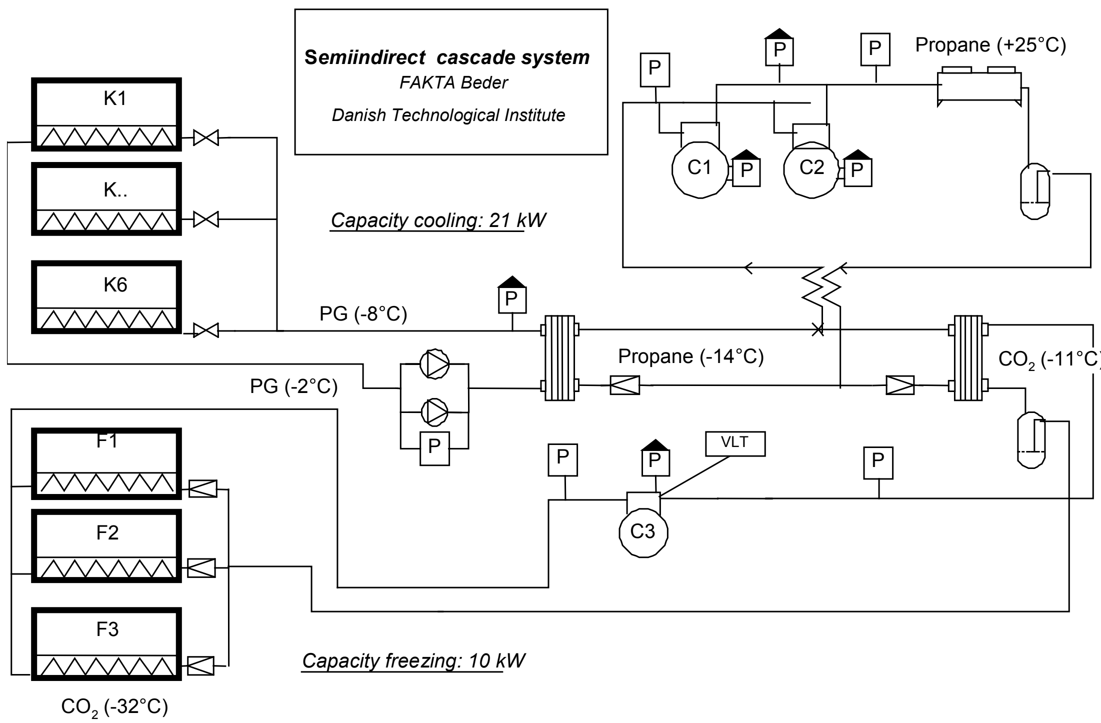


Figure1: Outline of the cascade system

Suction gas cooled semihermetic compressors are used for both propane and CO_2. The propane compressors are equipped with an oil pump, while the CO_2 compressor is splash lubricated.

The refrigeration system is built on a rack (compressor rack), where compressors, cascade exchanger, brine cooler, brine pump, valves and vessel are mounted. The aggregate is set up in a separate machine room from where pipes and tubes connect the cabinets to the system. The components, which contain propane, are built in a ventilated box.

The propane system operates with two evaporators: one cascade exchanger and one brine cooler. The cascade exchanger and the brine cooler are both plate heat exchangers. On both exchangers, electronic expansion valves are used for injection of refrigerant. The propane system is furthermore built without an oil separator. A mineral oil is used which is fully miscible with propane so that the oil returns to the compressors from the system. An identical oil level between the compressors is ensured by means of mounted pressure equalisation pipes.

The internal heat exchanger (plate heat exchanger) ensures supercooling of the propane liquid before the expansion valves and superheating of the suction gas to the compressor. Compression with propane gives very low compressed gas temperatures, and the compressor will therefore without an internal exchanger be relatively cold with high solubility of refrigerant in the oil as a result. This may reduce the lubricity of the oil and increase the amount of oil transported from the compressor to the system.

The compressor is furthermore supplied with a relatively large electric heating element for the oil sump during standstill.

Finally, a filter dryer and dirt filter are mounted in the liquid line of the system, but is without receiver.

The CO_2 system is constructed as a traditional refrigeration system. The cascade exchanger functions as the condenser of the CO_2 system, where CO_2 condenses against evaporating propane. The liquid circulates to the evaporators located in the low-temperature cabinets and cold stores of the supermarket. As injection valves, pulse width modulated valves are used where a superheating of 8 K is maintained. The gas is then sucked back to the compressor. The system does not have an oil separator.

The amount of oil entering the system will due to fine solubility with the refrigerant (ester oil/CO_2) and relatively high gas velocities be transported with the refrigerant back to the compressor. The first system in Odense had an oil separator, but there has not been observed any problems by not having an oil separator.

Figure 2: CO_2 compressor and receiver

Refrigerants

The refrigeration system of the supermarket uses the following refrigerants:

- *Propane (R290) – min. 97.5% pure propane, supplied by Calor Gas, less than 50 ppm of humidity in the system*
- *CO_2 (R744) – 99.9% pure CO_2, less than 50 ppm of humidity in the system*
- *Technical propylene glycol (Dowcal N/ 40% wt.) – inhibited and approved for foods.*

Propane and CO_2

Propane is an odourless and nontoxic gas. Propane is however an explosive gas with lower and upper explosion limits of 2.1-9.5 % v/v (0.038-0.171 kg/m^3). The automatic

ignition temperature is at 470°C. The gas is heavier than air and it will thus settle at the lowest level.
CO_2 is also odourless and nontoxic, but the gas can be dangerous to human beings at higher concentrations than 0.5% v/v (5000 ppm). The gas is heavier than air, and attention should be paid to the fact that expansion of liquid to pressures lower than 5.18 bar (–56,6°C – triple point) develops solid phase (dry ice). CO_2 is moreover characterised by very high saturation pressures. At 25°C, the corresponding saturation pressure is at 64.4 bar. Finally, the low critical temperature and pressure of CO_2 at 31°C and 73.8 bar should be taken into account, as the cycle with CO_2 becomes transcritical at higher temperatures.

Brine

Technical propylene glycol Dowcal N is used. The concentration is 40% wt., which ensures a freezing point at –21°C.

The glycol is inhibited and approved for foods (FDA approved).
– Nontoxic, but should not be drunken
– Non-inflammable.

Dowcal N is compatible with normally used metals (copper, steel and brass), plastics and elastomers (PE, PP, ABS, PVC, IIR, PTFE, EPDM, NBR and NR).

Cabinets and Heat Exchangers

All the cabinets used are from Arneg. The evaporators are modified to the refrigerant (brine/CO_2). This applies both to the evaporators for the cold stores and for the cabinets. All the evaporators for CO_2 are made as traditional fin coils with 3/8" copper tubes and aluminium fins.

	Air on	*Capacity [W]*		*Dimensions*
	[°C]	Cooling (+2°C)	Freezing (–20°C)	LxWxH [mm^3]
Multideck with cold store behind (glass doors), cooling	–2°C	4500	-	5100x3300x260
Multideck with cold store behind (glass doors), freezing	–26°C	-	4200	3600x3300x2450
Multideck with cold store behind (glass doors), ice cream	–28°C		2800	2100x3300x2450
Open multideck	–2°C	14400	-	10 m in length
Low-temperature cabinet	–27°C		2200	3750x1000x960
Low-temperature cabinet	–27°C		800	1985x1000x960
High-temperature cabinet	–2°C	1400		3750x1000x960
High-temperature cabinet	–2°C	700		1985x1000x960
In total		*21000*	*10000*	

Table 1: Survey of cabinets and cold stores

The cold stores are equipped with individual exchangers. Pulse width modulated valves from Danfoss control all the exchangers. The valve for injection of CO_2 uses a cycle time of 6 sec., while the one for brine uses a cycle time of 6 min.

Figure 3: The new supermarket – without products

Construction of the System

In connection with the system construction, Directive 97/23/EF of the European Council (i.e. the so-called Pressure Equipment Directive) is used. The Pressure Equipment Directive is adopted in Danish legislation by Bekendtgørelse nr. 743/99 (Danish executive order no. 743/99). The use of pressurised equipment is on the other hand a public affair and in this case Bekendtgørelse nr. 746 (Danish executive order no. 746) applies at present.

A calculation method for supporting the most important safety requirements in connection with pipes, fittings and vessels is used. The construction is simple, but it has not been possible to procure copper fittings with the required 3.1B certificate. Moreover, it has not been possible to get declarations of conformity with regard to valves for propane, only the declaration of the producer stating that the valve is compatible with refrigerants and can stand up to maximum operation pressure. There is however a substantial and relatively rapid development in this field, and it is expected that both fittings and other components are available in the near future both for propane and CO_2.

Both the propane part and the CO_2 part are constructed of copper pipes, while the brine part is made of ABS pipes and flexible tubes with regard to the connections to the heat exchangers of the cabinets.
In the brine system, valves – ASV-PV and ASV-I – from Danfoss are mounted in each circuit. The valves must ensure uniform pressure loss in each circuit. The problem with "unequal" load on the substrings has become more pronounced in large brine systems.

Components

Propane

Description	Supplier
2 compressors	Bitzer, 4T-8.2P
Air-cooled condenser	ECO FCE 071C63
Plate heat exchanger	R290/PGl: 21 kW, Swep, B25x70
Plate heat exchanger	R290/R744:10 kW, Swep V27x80 HP
Plate heat exchanger	R290/R290: 2 kW, Swep, B12x70
Plate heat exchanger	R290/PG: 3 kW, Swep B27x50
Electronic expansion valve	Danfoss ETRE – 30 kW
Electronic expansion valve	Siemens Staefa – 14 kW

CO_2

Description	Supplier
Compressor for CO_2	Bitzer, X2KC-3.2, BSE 55
Expansion valve	Danfoss AKV (nozzle 2-4)
Safety pressure switches	Danfoss KP 5
Filter dryer	Danfoss DU 303

Safety

It is important to take the special properties of propane (flammability) and CO_2 into account. This applies in connection with start-up, operation and standstill. In connection with servicing and maintenance, it is very important that the installers have the right qualifications.

The propane system is built in a ventilated box, where the concentration under all circumstances is maintained below 25% of the lower explosion limit. The zone inside the box is classified as zone 2 with requirements on minimum IP 54 for the electrical equipment and that the equipment is equipped with intrinsic safety circuits.

The safety in connection with the use of propane and CO_2 will not be explained further, but is referred to in /1+2/.

Control

The control of the system is composed of separate controls of compressors and cabinets respectively. Controls from Danfoss are used for both cabinets and compressors (AKC).

The compressors are controlled according to constant suction pressure. The propane system operates at –14°C by means of 4 capacity stages, which give a flow temperature of the brine at approx. –8°C and a condensation temperature for the CO_2 system at –10°C. The condensation temperature of the propane system is kept constant at 25°C by means of modulating operation of the ventilators. The CO_2 system operates with constant suction pressure at –32°C by means of a frequency converter, which can regulate the compressor in the area of 30-60 Hz.

The cabinets are controlled by means of Adap-Kool controllers depending on the refrigeration system. In the cabinets, fans, pan heater and defrost can be controlled. Moreover, the condenser fan and pump are controlled independently of the compressors. The pump operates with constant differential pressure. Defrost is carried out electrically.

The expansion valves for the two propane evaporators (brine cooler and cascade exchanger) use their own controller, where there is controlled according to superheating only.

In the injection valve on the cascade exchanger, both an ETRE valve from Danfoss and an electronic Siemens-Staefa valve which is controlled by a Siemens controller (PolyCool) are used. Optimum control of the cascade exchanger is still worked on. During start-up of the CO_2 system, the pressure on the CO_2 side increases quickly. The superheating signal on the propane side increases more slowly as a result of a high proportional band, and the CO_2 compressor stops at high pressure. The high proportional band is necessary so that the valve does not become too "eager", as the compressed gas temperature of CO_2 is +45°C at the inlet and –10°C at the outlet of the exchanger.

In order to compensate for the high proportional band, the settings of the derivative action is set high so that relatively small changes in the process variable (superheating) gives a reaction in the valve. Unfortunately, the valve reacts very quickly by closing when the liquid starts to parboil. It is assessed that it is necessary with a special control of the valve during start-up in particular, which is not based on a PID controller only.

Energy Consumption

The supermarkets being the standard of comparison for Fakta Beder are all of a more recent date and are roughly of the same size. The supermarkets have refrigeration equipment of 32 to 39 metres, i.e. number of meter cabinets, multidecks and cold stores. Eight supermarkets have been selected which are within the standard limits and energy consumption. All supermarkets have been equated to 32 metre of refrigeration equipment. Scroll systems with direct expansion have been installed in all of the reference supermarkets with separate systems for cooling and freezing respectively.

The distribution scale follows ”Brancheenergianalyse – Supermarked” (Danish industry energy analysis - supermarket) category 1 /3/:

Refrigeration system: 64%
Lighting: 33%
Other: 3%

Agreement between the distribution scale of the Danish industry energy analysis and the energy distribution in Fakta has been tested and determined by internal analysis of 30 supermarkets in the Fakta retail chain. The agreement of the scale has moreover been confirmed in a new internal Fakta report, where COWI /3/ has investigated ten different supermarkets.

Energy Meters in Fakta Beder

In Fakta Beder, the electricity consumption is registered three places. In addition to the total consumption of the supermarket, the total consumption of the refrigeration installations is registered, i.e. compressors, ventilators, pan heater, lighting in cabinets, pump and defrost heat elements. Furthermore, there is an electrical meter for the brine pump.

Sales area:	490 m^2 (720 m^2 in total)
Opening hours per day:	12 h

The consumption of the supermarket in July is distributed in this way:

	Number	kW/pcs.	Hours/days	kWh/ July
Refrigeration system	1	-	24	8682
Indoor lighting	210	0.058	12	4531
Outdoor lighting	10	0.1	12	372
Ventilation in supermarket	1	2.62	14	1137
Cash registers	3	0.4	12	446
Bottle machine	1	1	1	31
Sundries	1	2	12	372
In total				15571

The total energy consumption of the refrigeration system amounts to 8682 kWh/month.

Distribution of the Energy Consumption in the Refrigeration System

	Energy consumption [kWh for July]
Propane compressor 1	~3565
Propane compressor 2	~1240
CO_2 compressor	~605
Cabinets and multidecks:	
Lighting	107
Ventilation	487
Cold stores:	
Lighting	391
Ventilation	744
Other:	
Brine pump	~126
Pan heater	~1116
Defrost (cabinets, multidecks and cold stores)	~84
Condenser fans	140
Electronics	78
Total	8682

Units, where the power consumption has a preceding (~), are average values. The values have appeared on the basis of the change in the total energy consumption by repeated starts of the unit. The other values are found on the basis of the type plate values.

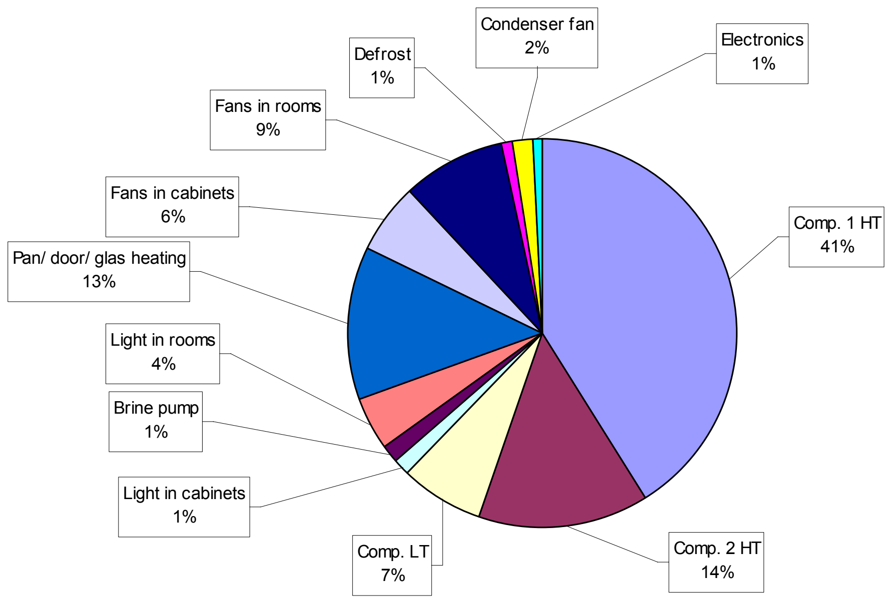


Figure 4: Distribution of the electrical energy consumption of the different components

The energy consumption of the compressors amounts to 62% of the total energy consumption of the refrigeration system, while the cabinets and cold stores altogether use 34%, and the remaining 4% is used for the brine pump, condenser fans and electronics. The energy consumption for the brine pump is surprisingly small. With regard to energy savings, it could be interesting to look at the compressors (including inexpedient operation, pressure loss, etc.), pan heater and ventilation in connection with cabinets and cold stores.

Comparison of Fakta Beder and the Eight Reference Supermarkets

There are no great variations in the energy consumption between the reference supermarkets in July, as a matter of fact the consumption of the reference supermarkets only varies 3-4% compared to the average. The month of July 2001 was very warm with an average temperature of 17.9°C (normally approx. 16°C) with a relative humidity below the average of approx. 65%. The measured energy consumption of the refrigeration system of Fakta Beder amounts to 60,1% of the equated total consumption of the reference supermarkets.
During the next months from July to December different improvement were carried out on the regulation of the brine system a the control of the cascade system and the energy consumption were reduces as seen on figure 5. These modifications caused fewer capacity jumps on the propane compressors and therefore low lower energy consumption. These modifications is described later.

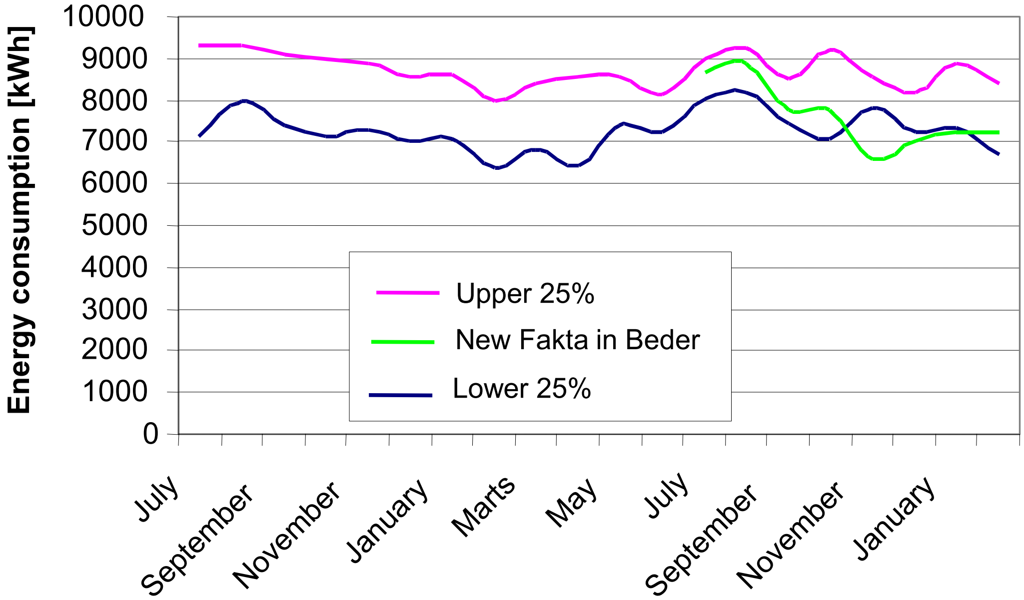


Figure 5: Energy consumption of the refrigeration system in Fakta Beder compared to the other eight Fakta supermarkets

It can thus be concluded that the energy consumption for the new system is neutral compared to modern and optimised conventional systems with R404A. This is an important conclusion for the technology in the future.

Fakta Beder is the result of the newest concept in the Fakta retail chain which implies more square metres, more outdoor lighting, the cabinets are adjusted to lower temperatures with more hot wires in order to avoid condensation, etc. Despite these variations, the energy consumption is neutral compared to the other supermarkets.

Possibilities for Optimisation

Both the brine cooler and the cascade exchanger can be optimised with regard to control. The insufficient control of the brine circuit and the two injection valves result in too many starts of refrigeration compressor 2. The reason for these frequent starts is due to the fact that both flow and temperature change with great speed, and the compressors must then as a result of great load on the plate exchanger quickly get the suction pressure down to normal level again. This requires rapid starts and stops of the compressor stage, which is not economically. The smallest CO_2 compressor on the market at present has an efficiency of approx. 10 kW during the actual operating situation. As our maximum requirement on the freezing is also 10 kW, the only chance for capacity control is by varying the rotational speed. It has however been demonstrated that the efficiency decreases distinctively during partial load.

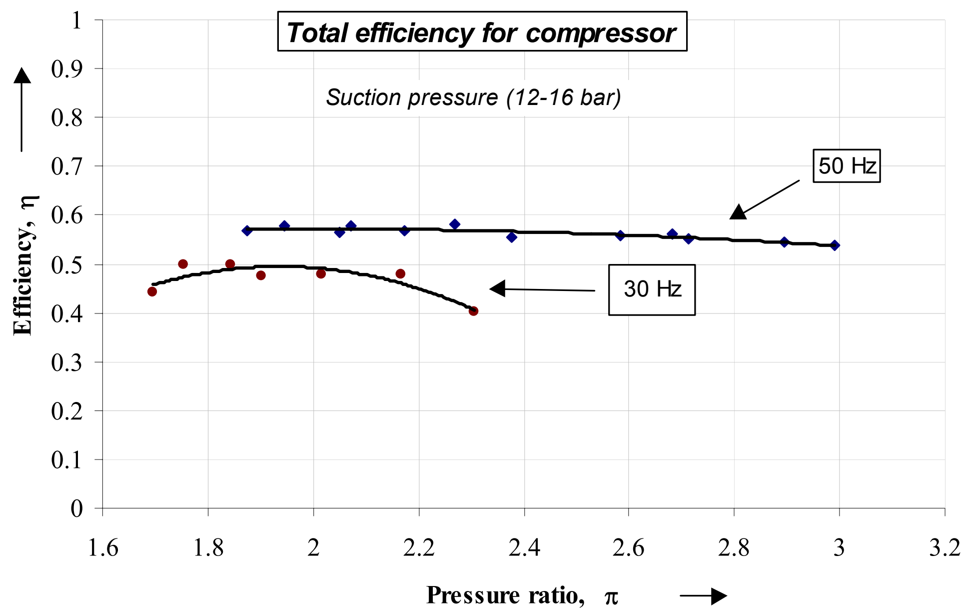


Figure 6: The total efficiency of the CO_2 compressor (X2KC-3.2) at 30 and 50 Hz respectively

The decrease in efficiency is naturally due to several circumstances with regard to the compressor. The mechanical losses in the compressor are almost independent of the rotational speed and will thus have relatively greater influence with a smaller mass flow. Moreover, the losses in the engine will be increased during operation outside the design area.

As the system in the supermarket will operate at partial load 95% of the time, these conditions are naturally of significant importance to the total energy consumption of the system. In large systems, where the capacity control can take place by means of start/stop of e.g. four compressors, the operation will be optimum.

Economy

Refrigeration assembly:	System	Cabinets	Assembly Mat.	Assembly Hours	Total
Total cost – cooling	100	244	64	91	499
HFC concept system incl. tax[1]	85	237	49	72	443
Additional charge CO_2/propane	15	7	15	19	56
In %	18%	3%	31%	26%	13%

[1] *first charge*

Elec. assembly:	Board Adap-Kool	Elec. board	Installation	Total
Total cost – electricity	32	56	52	140
HFC concept system	30	23	38	91
Additional charge CO_2/propane	2	33	14	49

Total project:	CO_2/ propane	Conven. HFC system	Difference Total	Difference In %
Refrigeration system	499	443	56	13%
Elec. system	140	91	49	54%
In total	639	534	105	20%

The total additional charge is primarily influenced by three main areas.

1. Elec. board
2. Assembly – refrigeration system
3. Construction – refrigeration system

Ad. 1 - Elec. board

The elec. board is especially influenced by:
- Frequency converter
- Special FI relay AC/DC
- Construction of safety circuit
- 2 x compressor control, conditional on frequency converter
- Oil control with PLC

If the task is simplified and certain component choices are different (especially if AC/DC relay and compressor control are changed to oil control construction, the task is assessed to be solved to index 40 compared to index 56 in the project.

Ad. 2 - Assembly – refrigeration system

The refrigeration assembly is influenced by lack of assembly experience and a certain training element of the installers involved in the project. It is estimated that the refrigeration assembly when having a certain experience can be changed to index 75 compared to index 91 in the project.

Ad. 3 - Construction – refrigeration system

Increased experience with system construction and competition on exchangers and compressors in particular is estimated to reduce the index to 90 compared to index 100 in the project.

On the grounds of this, it is assessed that the total additional charge for the propane/CO_2 system can be reduced to 10%. It is especially in large systems, the technology becomes competitive at first.

Conclusion

It is not surprising that the energy consumption is neutral with conventional R404a systems. The systems can be modulated and calculated and in the way it was seen very early that propane and CO_2 could compete. It has now in practice been demonstrated that the energy consumption is neutral on the basis of a comparison of eight identical Fakta supermarkets.

The CO_2/propan system and the chosen installation in Fakta Beder amount to an additional charge of between 12% and 20%, or separately on the system and assembly between 27% and 18%.

The additional charge of the system will be reduced if the total kW requirement increases. It is estimated that the additional charge of the system can be reduced to approx. 10% if the system was three times larger (approx. 60 kW cooling and approx. 30 kW freezing).

The success of the project can naturally be measured on the conclusion above, but also the dissemination of the technology to the industry can be included. After the two projects in Odense and Beder have been carried out, the industry has started to build systems for supermarkets based on CO_2. A Føtex supermarket has been built, and an ISO supermarket is on its way. The interesting thing about the system in ISO is that CO_2 as refrigerant is used both in connection with cooling and freezing. This system was already described in connection with the project in Odense, but at that time it was decided "to take one step at a time".

Well, it might be true that the refrigeration industry is conservative, but a lot of initiatives are in fact being made.

References

/1/ ”Demonstation af naturlige kølemidler i supermarkeder”,
ENS, J.nr. 731327/99-0199
/2/ ”Kulbrinter i mellemstore køleanlæg”, MST, J.nr. M126-0054
/3/ ”Brancheenergianalyse – Supermarked”, 1994
/4/ ”COWI rapport, Fakta”, 2001

Specifications on cascade plant in Beder (DK)

System measured energy usage (electric and gas)

- low-temperature (frozen food) refrigeration: 10 kW
- medium-temperature (chilled food) refrigeration: 21 kW
- heating, ventilation, air-conditioning (HVAC) 0 kW
- total store energy usage (July 2001) 15.571 kWh/month

Ambient operating conditions

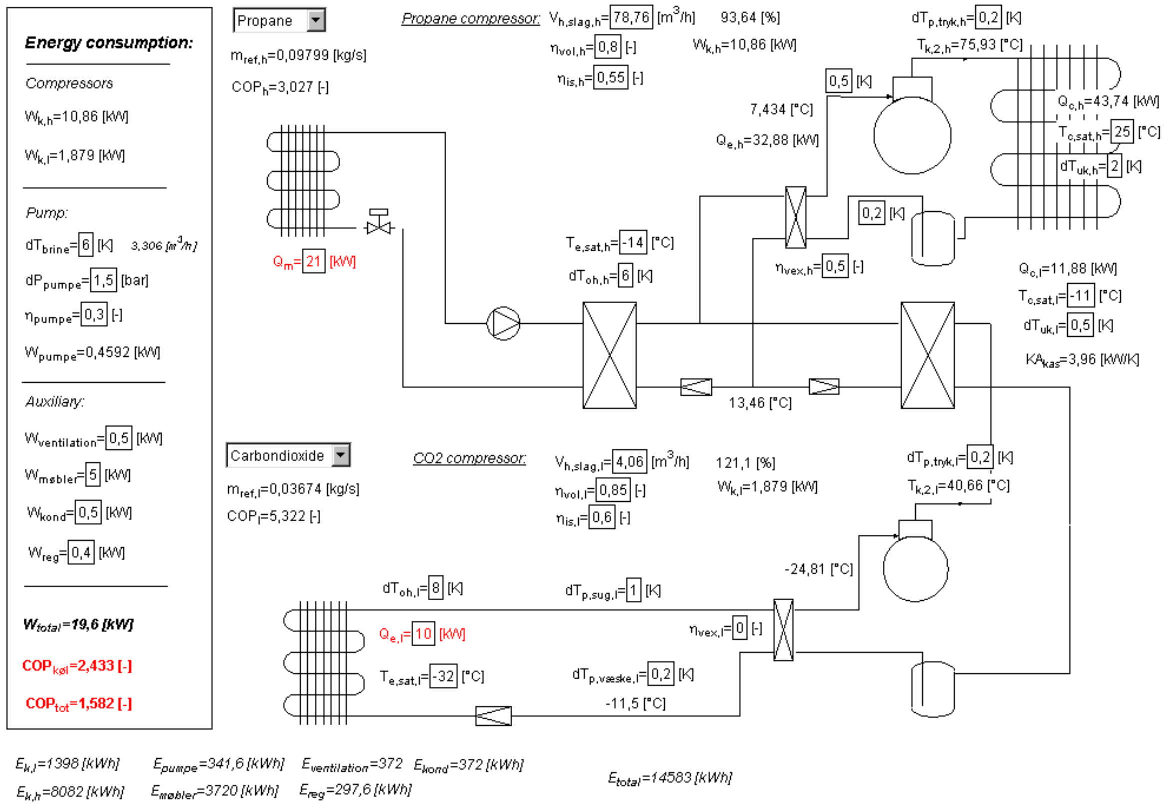
- outdoor temperature and relative humidity 17,9 °C (Avg.) / ?
- indoor temperature and relative humidity 22,5 °C (Avg.) / ?

System operating data

- refrigeration system saturated discharge and suction refrigerant temperatures

Propane system: Condensation Temperature: Evaporation temperature: Capacity (design):	 25°C (avg.)/ 22°C (min) / 30°C (max) -14°C (avg.) 33 kW
CO2 system: Condensation Temperature: Evaporation temperature: Capacity (design):	 -11°C (avg.) -32,5°C (avg.) 10 kW
Brine PG(40%): Outlet temperature: Return temperature: Capacity (design):	 -8°C -2°C 21 kW

Operation in full load/ capacity is seen from simulation:



- actual fraction of refrigeration reject heat recovered by HVAC system: 0%

Store and system design details

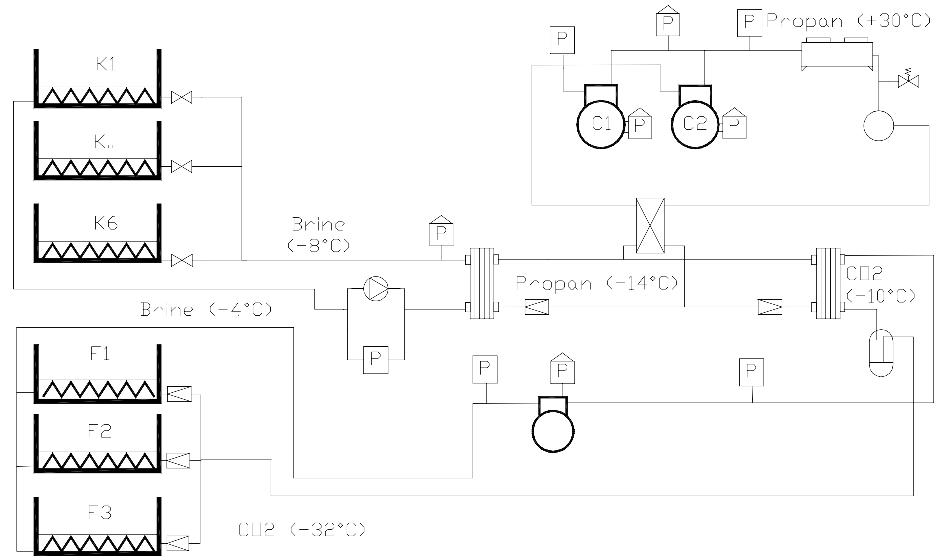
The supermarket in question is of medium size at least in Denmark. There would be approximately 700 supermarkets in DK of similar size in DK. The supermarket does not have bakery, restaurant or other speciality areas located within store! The store is built in one level (ground level) and there is no cellar (flat roof).

Store/ sales area and layout details: Total store: 720 m^2/Sales area: 490 m^2
Store operating hours (hours open per day) 12 h

	pcs	kW/pcs	Hours/day	kWh/ July
Refrigeration plant	1	-	24	8682
In store lightning	210	0,058	12	4531
Out store lightning	10	0,1	12	372
Ventilation store	1	2,62	14	1137
Tellers	3	0,4	12	446
Bottle machine	1	1	1	31
Various	1	2	12	372
Total				15571

Refrigeration system design parameters

Cascade plant working with propane and CO_2 and propylene glycol:



Design parameters:

Propane system:	
Capacity (design):	33 kW
System Capacity:	35 kW
Refrigerant charge:	10 kg
Annual loss:	2%
CO_2 system:	
Capacity (design):	10 kW
System Capacity:	9,9 kW
Refrigerant charge:	6 kg
Annual loss:	15%
Brine system:	
Refrigerant charge:	120 l

Number and type of display cases and refrigerated storage rooms:

	Air on	*Capacity [W]*		*Dimensions*
	[°C]	Medium (+2°C)	Low (-20°C)	LxBxH [mm^3]
Storage (glass doors to shop)	-2°C	4500	-	5100x3300x2360
Storage, frozen food (doors)	-26°C	-	4200	3600x3300x2450
Storage, ice cream (doors)	-28°C		2800	2100x3300x2450
Vertical cabinet (no doors)	-2°C	14400	-	10 m in length
Horizontal case	-27°C		2200	3750x1000x960
Horizontal case	-27°C		800	1985x1000x960
Horizontal case	-2°C	1400		3750x1000x960
Horizontal case	-2°C	700		1985x1000x960
Total		*21000*	*10000*	

* Type of defrost method and control used: Electrical
* HVAC system type: None

Control

Propane:
Condensation temperature: Constant by modulating fans (+25°C)
Suction pressure/ temp.: Constant by using 4 capacity steps on compressors (-14°C)

CO_2:
Suction pressure/ temp.: Constant by using frequency converter: 30-60 Hz (-32°C)

Pump:
Constant differential pressure of 1,5 bar

Energy consumption (refrigeration system):

Total energy consumption in July 2001: 8682 kWh

Consumer	Energy consumption [kWh for July]
Propane compressor 1	3565
Propane compressor 2	1240
CO_2 compressor	605
Cases and cabinet:	
Lightning	107
Ventilation	487
Storage rooms:	
Lightning	391
Ventilation	744
Brine pump:	126
Heaters (Edge):	1116
Defrost (cases, cabinet and storage)	84
Ventilation condenser	140
Electronics	78
Total	8682

Appendix B

"Demonstration of the Use of Natural Refrigerants in Supermarkets"

Framework and Organisation of the Project

The project is undertaken by the Danish Technological Institute in collaboration with Super Køl and FDB. The project is moreover sponsored by the Danish Energy Agency under J. No. 731327/99-0199.
Furthermore, a number of companies have participated in the project by supplying components.
The demonstration system is set up in Lokal Brugsen on Juelsmindevej in Odense, Denmark. The project was commenced in February 2000 and the system was started on 26 May 2000. The project is completed on 1 October 2000.

Purpose of the Project

The purpose of the project has been to demonstrate the use of natural refrigerants in connection with refrigeration systems in supermarkets.
And at the same time to demonstrate that the energy consumption can be reduced slightly in this type of system seen in relation to the optimised systems which are known today. The price of a system has moreover been assessed.

Description of the Project and the System

The project takes its starting point in a previous project (J. No. ENST 731327/97-0164 and MST 128-0428) in which different systems with natural refrigerants have been investigated theoretically and through laboratory work. Through this work, different concepts of systems have thus been evaluated. The theoretical analysis indicated that ordinary indirect systems will increase the total energy consumption of the system by 5-10%. A cascade system using CO_2 (R744) directly in the freezing display cabinets will however be a much more efficient solution where the energy consumption compared to an optimised direct R404A system (subcooling system) could be reduced by approx. 5%. Here the energy consumption includes the energy consumption for condensers, ventilation in engine room, control, cabinets (light, frame jamb heating, defrosting and ventilators) and energy consumption for pumps. The relative energy consumption distributed on components is like this:

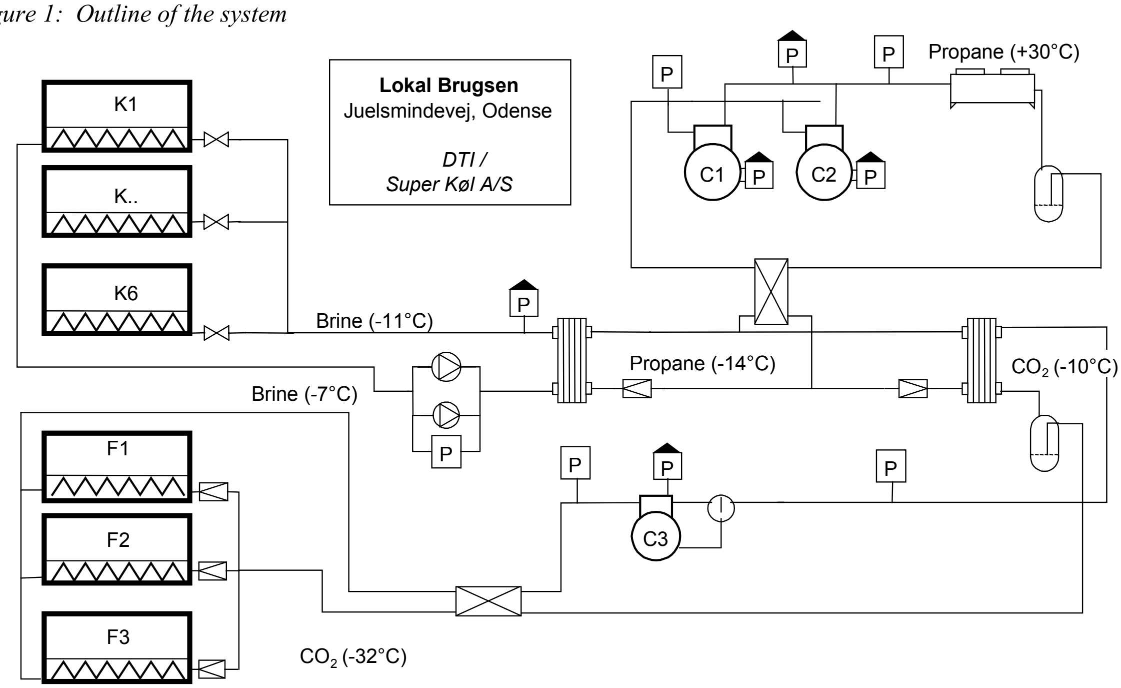
Energy consump.	Propane/CO2	Opt. R404A
Condensers	7%	6%
Ventilation	1%	0%
Control	1%	1%
Cabinets	40%	38%
Pumps	4%	0%
Compressors	47%	55%
Total	**100%**	**100%**

The system chosen is thus built as a cascade system. Propane is used at the high temperature level (-14/+30°C), while carbon dioxide is used at the low temperature level (-32/-11°C). The two refrigerants exchange heat in the cascade exchanger where propane at dry expansion evaporates and CO_2 condenses. CO_2 is used directly in the supermarket, while propane exchanges heat to an indirect system with glycol. The glycol is pumped from a closed system to the cabinets. The performances of cold dimensioned are:

Cooling: 6 kW at –32 °C (CO_2)
Freezing: 10 kW at brine flow approx. –10 °C

Please see Figure 1 for an outline of the system.

Figure 1: Outline of the system



Construction of the Plant

It has been attempted to construct the system with existing and commercially available components. Due to the fact that propane (R290) is inflammable and CO_2 (R744) operates with relatively high pressures, it is however necessary during the design and construction to make allowances for these special properties. The system is filled with 6 kg of propane and 6 kg of CO_2.

The propane system is set up in an engine room in a tight and ventilated box. The mechanical ventilation that always operates ensures a constant low pressure in the box. Pumps in the engine room ensure circulation of brine for the cooling display cabinets.
The CO_2 system is set up in the engine room from where fluid and suction pipes are connected to the freezing display cabinets. A CO_2 detector has moreover been installed which registers CO_2 gas at 4000 ppm.

The propane system is constructed with steel pipes, and most of the components are made of steel. This has been necessary for the approval of the system since certified components in copper are not available in several areas. The system is designed without an oil separator in which the miscibility between refrigerant and oil (mineral) and the construction of the system ensure oil return to the compressor. All of the components used for the propane system are commercially available.

Figure 2: Semihermetic prototype compressor from Bitzer for CO_2

The CO_2 system is constructed with copper pipes and components of copper are used. The compressor is a semihermetic prototype compressor. The evaporators used in the supermarket are custom-made by the Danish Technological Institute and 3/8" copper pipes are used in the exchanger.
An oil separator ensures that only a small amount moves around in the system. The cascade exchanger between evaporating propane and condensing CO_2 is a plate heat exchanger of 45 bar. The system is designed for 19 bar on the suction side and 32 bar on the delivery side. The relief valves with the correct adjusting pressure are placed on the suction and delivery sides of the compressors and at each heat exchanger.

Results of the Project

During the summer of 2000, data has been collected from the system (temperatures, pressure and total energy consumption).
The refrigeration system has from the start-up been running without failure or problems. It has met the desired conditions and keeps -20°C in the freezing display cabinets and +2°C in the cooling display cabinets. Particularly under very hot conditions, the system has shown good performance.

Figure 3: The refrigeration system. The propane system has been mounted at the upper side, the brine pumps and the CO_2 system at the lower side.

The defrosting times have furthermore been reduced on the freezing display cabinets and take between 12-15 min. On the basis of the energy measurements collected from the new and the old systems respectively, the energy savings can be evaluated. However, several factors play a part in this comparison. It has been attempted to adjust different performances of cold on cooling and freezing respectively between the systems, other cabinets and different temperatures in the cabinets as the old system could not keep the temperatures in the cabinets. It is moreover emphasised that the old system in several respects was worn-out!
The total energy consumption has been reduced by 10%, while the energy consumption for the refrigeration system (compressor + condenser + pump) has been reduced by 20%.
Based on the experience from the project, it is assessed that the additional charge for a propane/CO_2 system in a medium-sized supermarket (30/60kW) will amount to approx. 10-15% of the total installation incl. assembly.

Future Refrigeration Systems in Supermarkets

The phase-out of CFC and HCFC refrigerants as well as the restrictions against the use of HFCs will imply that the focus on the natural refrigerants will be intensified in the future. However, the direct use of natural refrigerants cannot be accepted, while the use of typical indirect systems cannot be preferred either due to the energy consumption. The system described with direct use of CO_2 as refrigerant has in every area shown great potential. It is thus claimed that this type of system will gain a footing in the future.

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IEA Heat Pump Programme, Annex 26: Advanced Supermarket Refrigeration/ Heat Recovery Systems

Report from Sweden.

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1. Introduction

Increased sales in supermarkets, stricter environmental legislation for CFC and HCFC refrigerants and a major consideration of the use of energy and the effect on the environment have influenced the supermarket sector during the last year. Sales in supermarkets have increased 4.6 % during year 2001 in comparison with the year 2000 [30]. The consumption of deep-frozen products in supermarkets during 2001 was 231788 metric tons while during 2000 it was 216427 metric tons, which is an increase of 7% [12]. The increase in sales in supermarkets places a demand on more cabinets, cold rooms and refrigeration systems that increase the energy consumption from food stores.

A survey made by one of the Swedish supermarket chains shows that the average energy consumption in 256 supermarkets is about 421 kWh/m^2 a year. The total energy consumption in a hypermarket (about 7000 m^2) is about 326 kWh/m^2 a year while the total energy consumption in small neighbourhood shops (about 600 m^2) is about 471 kWh/m^2 a year [26].

Supermarkets are using large amounts of energy; approximately 3% of the electric energy consumed in Sweden is used in supermarkets (1,8 TWh/year). A breakdown of the energy usage in figure 1.1 shows that, typically, 47% is used for medium and low temp refrigeration, 27% for illumination, 13% for fans and climate control, 3% for kitchen, 5% for outdoor usage and 5% for other uses [13].

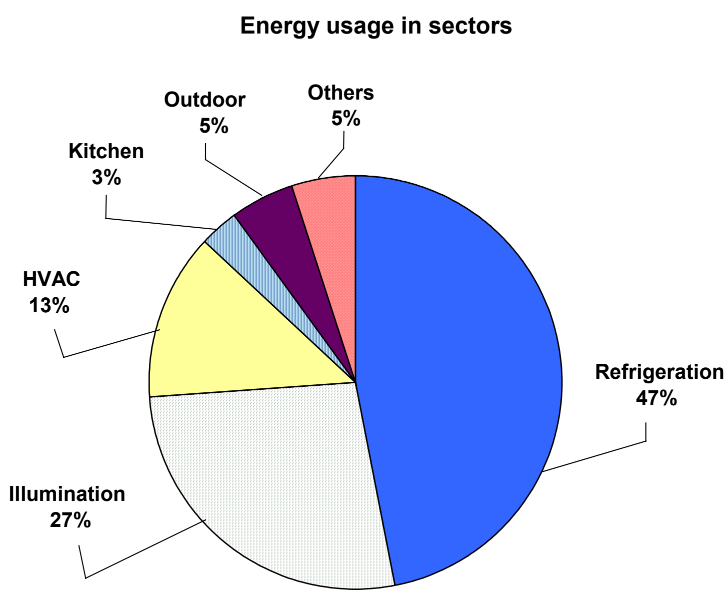

Figure 1.1: A breakdown of energy usage in supermarkets.

An analysis of the overall cost structure of a typical supermarket including the profit [23] is shown in figure 1.2. 76% are product costs, 11% are wages costs, 3% are rent costs, 2% are marketing costs, 4% are other costs, 1% is energy cost and 3% is profit. The cost of energy for this typical supermarket is only 1% of the total turnover. Since the profit is 3% of the turnover and a 50% reduction of energy consumption gives a 15% increase in profit, this sounds a little more attractive to the owner.

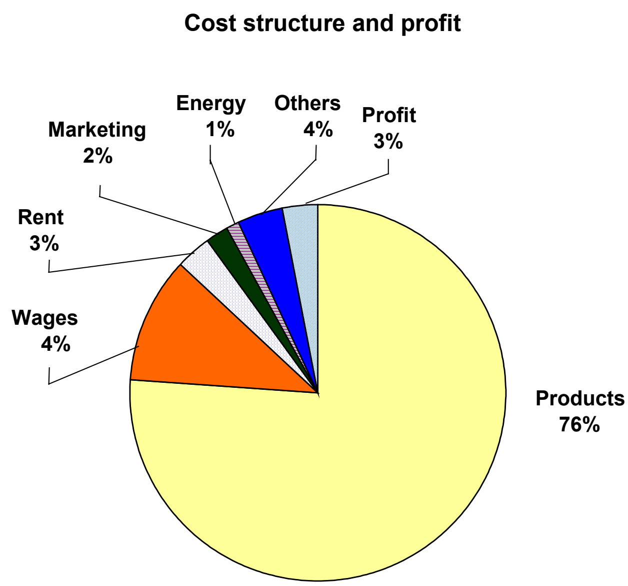

Figure 1.2:Cost structure and profit of a typical supermarket.

There are many indications that large gains in energy efficiency are possible using economic arguments only. The important question is how to address these issues and furthermore, identify the key players in the process. It is important to understand that the overall objective of these actors differs considerably.

The shop owner is responsible for the operation and investments in the supermarket. He usually makes the principle decisions. Different supermarket chains have different structures. A more centrally oriented Swedish chain such as COOP (member of Euro Coop) may take decisions with different rationales than a more decentralized organization. The problem for the shop owner is that many different potential investments compete. Since the cost of primary energy is a low percentage of the overall turnover, a more attractive display of foodstuff or interior decoration of the shop may show shorter payback figures. Energy efficiency is a driving force that is used in various forms of environmental labelling in supermarkets. A green profile may attract customers. Relatively short paybacks on simple energy efficiency measures such as night lids/curtains, timers, cleaning of fan coils etc. are attractive today.

Energy consultants designing supermarkets are making selections among various overall solutions and concepts for the supermarket owner. The consultant is knowledgeable in various system solutions and techniques and the way these affect the energy efficiency of a supermarket. The influence on the owner is considerable. The “concept” is important for the design. Low life cycle cost is a weak argument in these discussions. Low first cost is often given high priority. A Future development of the energy market towards more “Energy Services” may change this.

Manufacturers of components or sub-systems and service companies are important players. These companies develop new technologies. Energy efficient display cases, low charge central chillers, energy efficient illumination systems etc. are typical examples.

Suppliers of foodstuff such as beverages, milk products etc are offering concepts such as the ”milk market” a special low temperature zone within a supermarket. A potential development of water-cooled plug-in units (instead of air cooled) may, for example, be initiated by companies that produce soft drinks.

It is clear that several important actors need to co-operate in order to attain more energy efficient supermarkets in the future. Other key players are, of course, energy labelling institutions such as Environmental Protection Agencies, governmental research councils, refrigerant manufacturers, customers etc. Education and research support for new energy efficient technologies are important actions that need to be taken. Public awareness of energy efficiency issues and long-term climate problems are growing and lead to competitive advantages for those supermarkets that manage to adopt, and present, an energy efficient profile for the future.

Refrigeration systems, display cases, indoor climate control and illumination are the areas with the greatest potential for improvement. Since the energy systems of a supermarket are relatively complex, improvements in one subsystem affects other systems, thus making an analysis of potential improvements non-additive. Typical efficiency improvements may involve refrigeration systems, heat recovery, more efficient illumination, more efficient display cases with night lids, more efficient control, floating condensation etc.

Several new system solutions as completely, partially and cascade indirect systems have been developed and introduced in recent years in Sweden to lower the refrigerant charge and, at the same time, minimize potential refrigerant leakage. For example, in a supermarket in the city of Lund in the south of Sweden, a refrigeration system with 500 kg refrigerant CFC was replaced with a new one that uses 36 kg ammonia as refrigerant and CO_2 as secondary refrigerant [22].

Growing interest in secondary loop systems has led to the development of some new secondary refrigerants based on potassium formate and potassium acetate alone or mixed. Water solutions of glycols, alcohols and chlorides have long been used as secondary refrigerants. A number of non-aqueous heat transfer liquids are also used. For low temperature applications coils in display cases must be re-designed for the laminar flow regime to avoid excessive pressure drop. Experiences from practical installations of laminar flow coils are promising with temperature differences in the range of 1.5 –2 K on the liquid side.

Another very promising development is CO_2 as a secondary refrigerant. This technology is implemented in more than 30 supermarkets throughout Sweden. CO_2 systems require much lower tube diameters and the pressure drop is negligible when compared to conventional systems. Practical problems with high pressure during a shut down have been satisfactorily tackled [26]. Cascade systems with CO_2 in the low temperature stage and mechanical sub-cooling from an Ammonia system are an interesting solution [8].

A third promising development in this field is ice slurries. These systems offer additional advantages with enhanced thermal capacity and a "built-in" thermal storage in the system without increased pressure drop if ice-slurry with the right consistence is produced [9]. The mixture liquid water/ice may be "tweaked" with small amounts of antifreeze such as propylene glycol.

Secondary systems and the minimisation of refrigerant charge may lead to an unwanted trade off in overall energy efficiency. Theoretical calculations confirm this due to the obvious extra temperature differences introduced in the system [10]. An evaluation using a concept like TEWI (Total Equivalent Warming Impact) [29] may be used to estimate the overall environmental impact. Calculations made for the two extremes: Sweden (Nuclear and Hydro) and Denmark (Coal and gas) show that the impact from large refrigerant leakages always dominate over CO_2 releases related to the production of electricity [1]. Recent practical experiences and experimental studies [18], [19], [21], however indicate that indirect systems are as energy efficient as direct systems, if properly designed. More research is clearly needed to clarify the reasons for this. More efficient defrosting systems, better part load characteristics and more reliable systems are believed to contribute.

The development of energy efficient supermarket refrigeration systems is strongly related to the national refrigerant policies. Environmental legislation has, from an historic perspective, never been eased. This implies stronger future regulation of HFC refrigerants. This is likely to lead to limitations in permitted refrigerant charge (for a given capacity) and maximum leakage levels. The acceptance of flammable fluids has increased considerably in Northern Europe during the last five years and the development is not likely to go back unless severe accidents occur. Strong initiatives to phase out HFC refrigerants completely have, for instance, been taken by the Danish Environmental Protection Agency, for example. The regulation of the refrigerants CFC, HCFC and HFC in Sweden is presented in figure 1.1.

ASHRAE Number	Primary Replace-ment	Type of refrige-rant	Stop for import or new installations	Stop for refill	Stop for use	Primary environmental concern
R12, R500, R502	**R134a R404A**	**CFC**	**1/1 1995**	**1/1 1998**	**1/1 2000**	**Ozone depletion**
R22	**R407C R417A**	**HCFC**	**1/1 1998**	**1/1 2002**	**N/A**	**Ozone depletion**
R134a, R407C, R404A, R417A, R410A, etc.	**?**	**HFC**	**1/1 2006?**	**?**	**N/A(?)**	**Global warming**

Figure 1.1: Regulation of refrigerant CFC, HCFC and HFC in Sweden.

The development of alternative cycles is interesting, but the possibility to compete with vapour compression systems in supermarket applications is limited. For large temperature lifts air cycles may be used for freezing applications and desiccant cooling may be an interesting compliment in hot and humid climates for air conditioning and dehumidification.

In 1998 the Department of Energy Technology of the Royal Institute of Technology of Sweden started a project in co-operation with different companies and the Swedish National Energy Administration. Modelling and field measurements of supermarket energy systems have been undertaken in the research project "The energy efficient supermarket" for more than four years. The results have been part of the Swedish contribution to the IEA Annex 26

(Advanced Supermarket Refrigeration/Heat Recovery Systems) under the IEA Implementing Agreement on Heat Pumping Technologies.

The overall aim of the project “The energy efficient supermarket” is to develop a sound simulation model where different system solutions can be compared in detail with focus on energy usage, environmental impact (TEWI) and LCC (Life Cycle Cost).

The program, “CyberMart” is built in modules dealing with subsystems such as in- and outdoor climate, display cases, cooling and freezing rooms, refrigeration machinery, the building envelope etc. CyberMart is a day-to-day simulation program that allows the user to see the variation of different variables such as compressor power, refrigeration capacity and temperatures in the supermarket during one year.

The model is currently under validation with four different supermarkets in Sweden in cooperation with COOP and ICA, two major Swedish supermarket chains. The model development activities are now treating issues such as illumination, indoor climate and heat recovery, comfort cooling etc. Interesting issues to study are the potentials in heat recovery in the winter, moisture control/dehumidification and floating condensation. Additional Field measurements intended to validate data are currently being set up.

2. New system designs

The quest for increased energy efficiency and the phase out of ozone depleting substances have affected the system design for supermarkets considerably. The traditional refrigerants are replaced today with R404A, R134a etc. A renewed interest in natural refrigerants such as Ammonia, Propane and CO_2 has resulted in charge minimisation and relatively leak proof systems placed in machine rooms. Supermarkets appear in all kinds of sizes from small neighbourhood shops to hypermarkets and the choice of overall system solutions vary considerably.

Energy efficiency improvements in supermarkets demand more efficient refrigeration systems, illumination and display cases. Display cases are commonly large energy consumers especially vertical open display cabinets. The reason is the large amount of food that this kind of cabinet displays on a small surface in the store and the large open area. Energy efficient display cases have been developed in a technology procurement competition launched by the Swedish Energy Administration. The winning brine-cooled display case uses approximately 50% less energy than similar older designs [1].

New attractive designs for supermarket refrigeration systems with energy efficient equipments have been developed to decrease the energy use, to lower the refrigerant charge and, to minimize potential refrigerant leakage [3], [27].

The most traditional refrigeration system design in supermarkets is the Direct System, figure 2.1. In Direct Systems the refrigerant circulates from the machine room, where the compressor is found, to the display cases in the sales area where it evaporates and absorbs heat. The system requires long pipes to connect the compressor and the display cases. This implicates very large refrigerant charges.

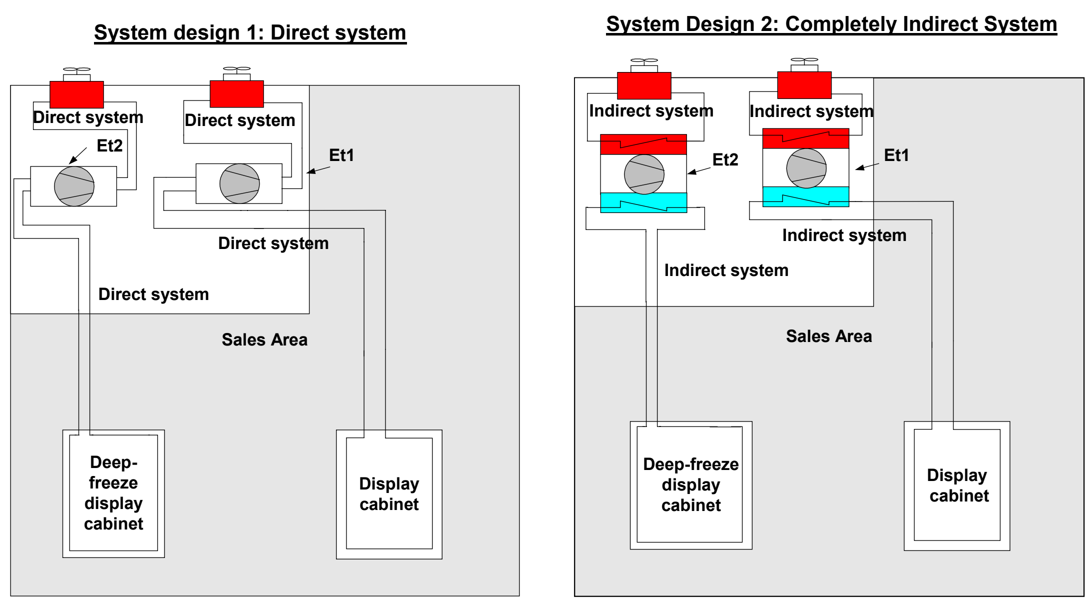

Figure 2.1: System design 1, Direct System and System design 2, completely indirect system

Refrigeration systems with indirect systems have been introduced in supermarkets to decrease the refrigerant charge and to minimize potential refrigerant leakage. A design with a completely indirect system is introduced in figure 2.1. In this system design two parallel refrigeration systems (chiller) with different brines and level of temperatures exist.

The brine in the intermediate temperature level often has an approach temperature around -8°C and a return temperature around -4°C. A typical value of brine temperature to deep-freeze display cases is about -32°C and the return temperature is about -28°C. Secondary refrigerant as water solutions of potassium formate or CO_2 vapour-liquid are used as the heat transfer fluid in the low temperature system. CO_2 is a very interesting heat transfer fluid for low temperature systems because the transport properties and the low viscosity at temperatures below -20°C. The problem with CO_2 is the high pressure (19.7 bar at -20°C).

Two other secondary loops are used in the system to transport the heat rejected from the condensers, in the machine room, to two different dry coolers located in the roof of the supermarket. The waste heat from the condenser can be recovered during the winter with substantial energy saving.

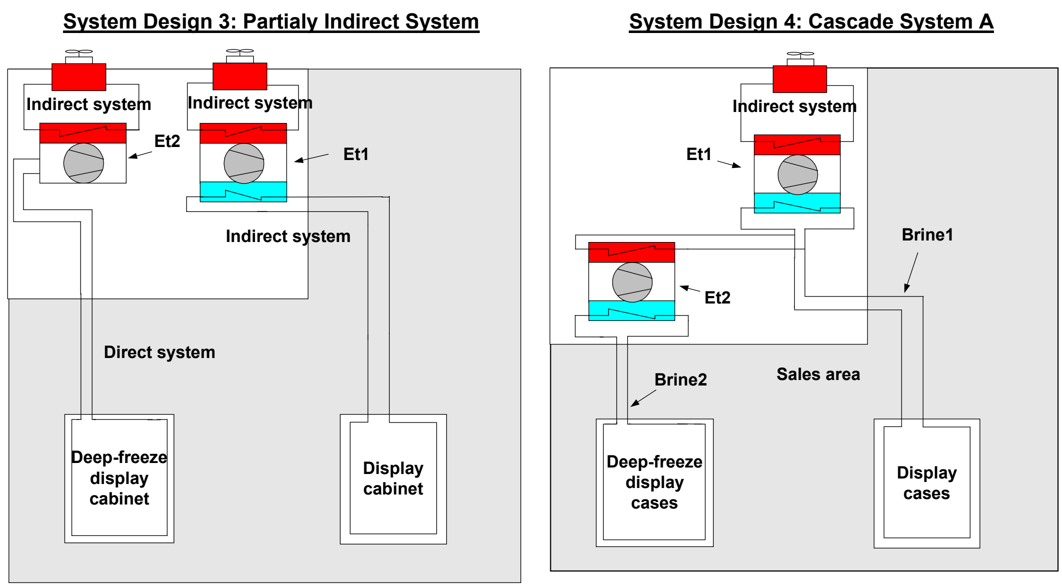

Figure 2.2: System design 3, partially indirect system and System design 4, Cascade system A

The most common Partially Indirect system in supermarkets is shown in figure 2.2. The heat from the condensers is rejected by a dry cooler in the roof of the supermarket to the environment. The low temperature system has a direct system between the compressor and deep-freeze display cases and the intermediate temperature system has an indirect system between the cabinets and the chiller.

The Cascade system, in figure 2.2, is a favourable solution that avoids the large pressure ratio in the low temperature system obtained in the completely indirect system. The installation operates with two different temperature levels and secondary loops. The temperature of the brine in an intermediate temperature level has, as in completely indirect systems, an approach temperature of about -8°C and a return temperature of about -4°C. The approach temperature of the brine in the low temperature system is about -32°C and the return temperature is about -28°C.

The condenser heat from the low temperature system is rejected to the secondary refrigerant on the intermediate temperature. The condensing temperature of the low temperature system is about 0°C, which increases the coefficient of performance of the refrigeration cycle and decreases the energy consumption of the low temperature system. The drawback with this system is the increase of refrigeration capacity and compressor power of the intermediate temperature system due to the condenser heat from the low temperature system.

The heat from the other condenser is rejected to the outside through a secondary loop that connects the condenser to a dry cooler located in the roof of the supermarket. The waste heat from the condenser can also be recovered during the winter.

Another design utilizing a Cascade system is introduced in figure 2.3. The secondary refrigerant in the intermediate temperature level cools the display cases and the condensers of the low temperature system as in the previous design. The difference is in the refrigeration system of the low temperature level contained in the deep-freeze display cases. The reason for the system is to decrease the pump power of the low temperature system because the high viscosity of the secondary refrigerant at low temperature that affects the pressure drop of the fluid between the chiller and deep-freeze display cases.

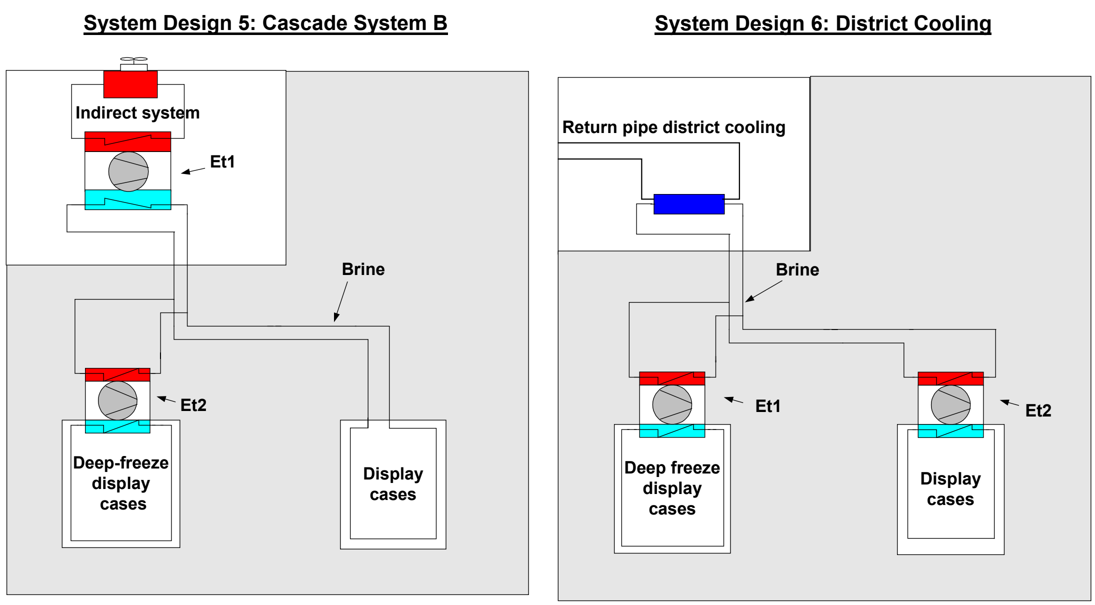

Figure 2.3: System design 5, Cascade system B and: System design 6, District Cooling

Figure 2.3 also present system design 6 with *District Cooling*. In this installation the display cases contain compressors, condensers and evaporators. The district cooling return pipe cools the condenser from the display cases throughout a secondary refrigerant. The brine temperatures fluctuate between 12°C and 16°C, which is higher than normal. The condensing temperature of the refrigeration systems is about 20°C, which implicates lower compressor power in comparison with the other system designs. Disadvantages with this solution are the price of the district cooling that might increase the total energy consumption, the dependence on the contractor of district cooling and the impossibility of heat recovery

A variant of the design with Completely Indirect System that improves the performance of the system is presented in fig 2.4. In this case, the refrigerant in the low temperature system is sub-cooled with the brine of the intermediate temperature level [31]. The temperature of the refrigerant after sub cooling can be about +5°C.

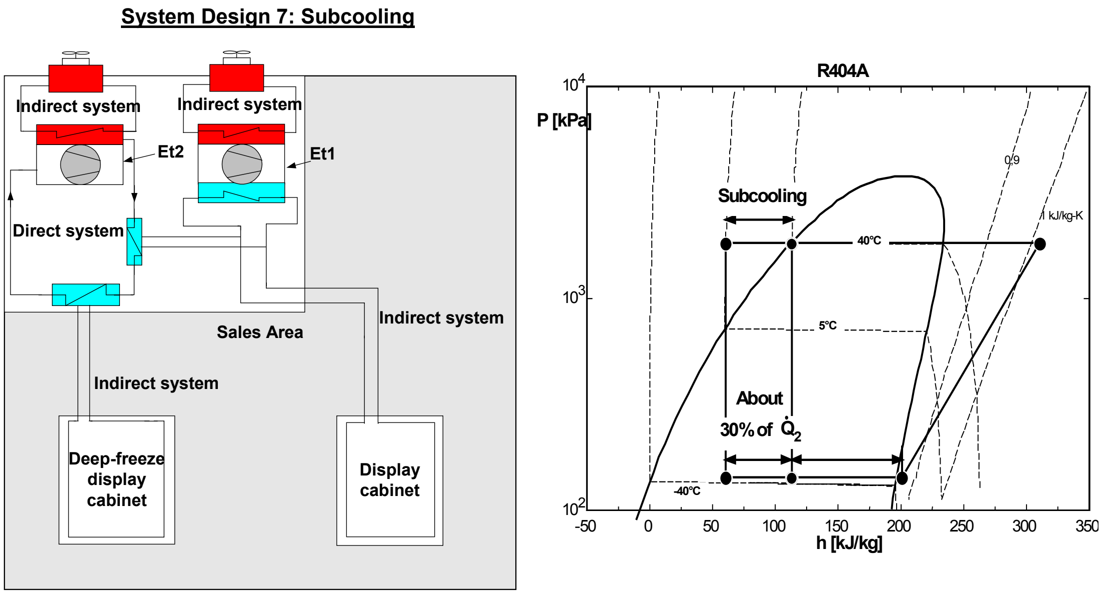

Figure 2.4: Completely Indirect system with sub cooling and P-h diagram for R404A with the refrigeration cycle of the low temperature level.

The effect of sub-cooling on the low temperature system is presented in the P-h diagram for R404A in fig 2.4. The influence of sub cooling on the refrigerating effect depends, among other things, on the kind of refrigerant. For R404A the increase in the refrigeration capacity for the low temperature system can be approximately 30% as shown in fig 2.4. The heat from the sub-cooling process is rejected to the intermediate temperature system that has a higher Coefficient of Performance COP_2 than the low temperature system. This implicates lower energy consumption than for a completely indirect system.

3. Model

3.1. Model overview

Many new ideas and concepts have been introduced in supermarkets, during the last few years, with the intention of decreasing energy usage and to minimizing refrigerant charge. Some of these new ideas and concepts are beneficial while others are less beneficial. One of the general characteristics of energy efficient measures is that several options are often available at the same time. One obvious example for supermarkets is floating condensation or heat recovery (at a higher condensation temperature). Many evaluations tend to deal with this issue as if the different options were independent. This is not generally true. A system approach must be taken, evaluating various energy efficient measures at the same time. In order to do so a "tool" is needed. The vision of a tool is a user-friendly computer code for the evaluation of energy efficient measures in supermarkets and other premises related to the storing or handling of refrigerated food.

A computer model capable of dealing with the actual system solutions, investment economics (LCC) and environmental impact (TEWI) has been developed. Focus in the model is on refrigeration technology but other aspects have been considered when relevant. The model is built in modules dealing with subsystems such as display cases, refrigeration machinery, the building envelope etc.

The supermarket here is regarded as a system consisting of several sub-systems (figure 3.1). This means that improved energy efficiency is achievable by a number of measures such as: more efficient display cases, more favourable indoor climate, increasing the COP of the refrigeration system or simply by an overall reduction of chilled or frozen foodstuff etc.

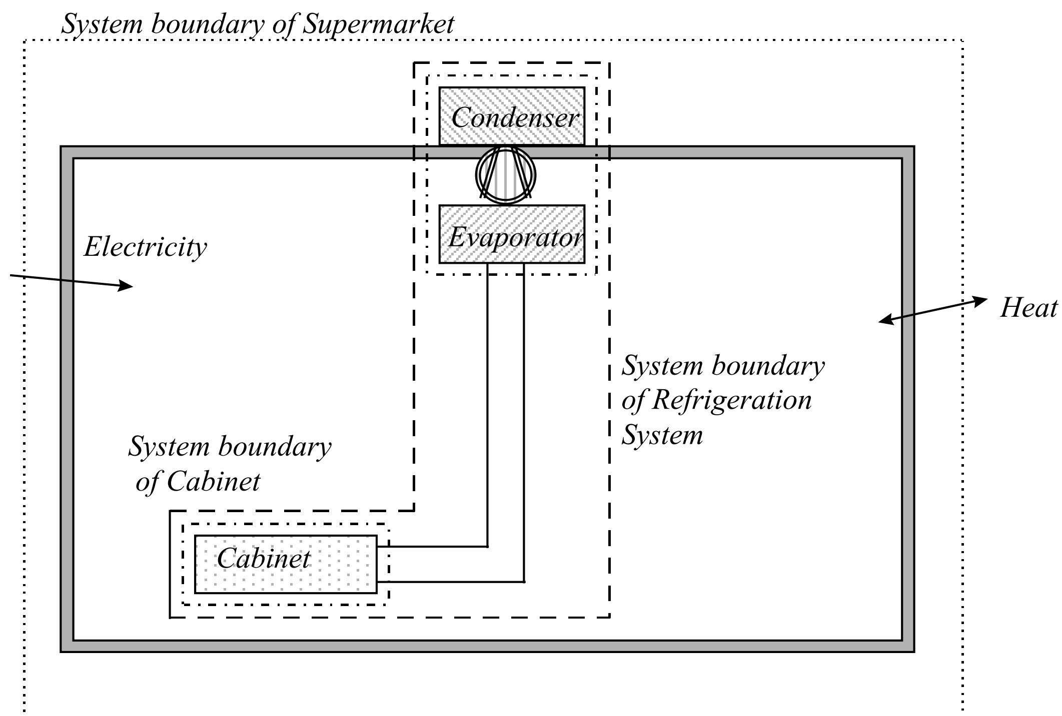

Figure 3.1: System boundaries in a supermarket.

The users of the program are designers and technicians from different companies involved in the project. The idea is to have a program that is user-friendly with little input data and reliable results. The properties of the different components in the model have been taken from performance data of different manufacturers, which is the data that technician and designers have access to.

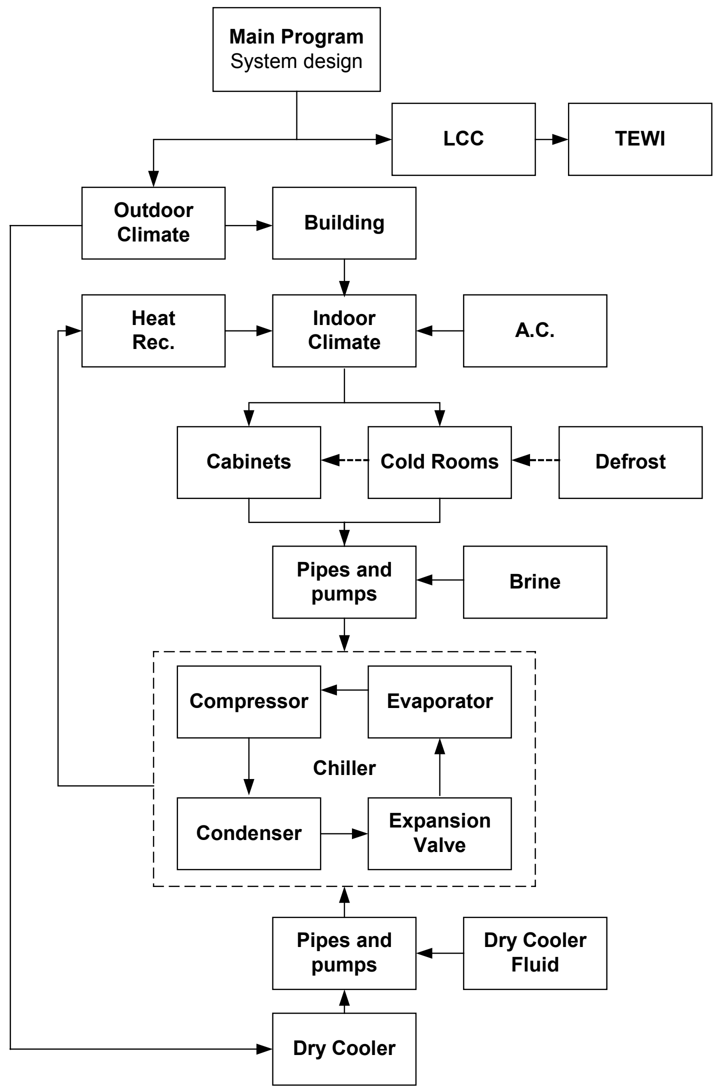

Figure 3.2: Conceptual schema of the modules in Cybermart.

The diagram in figure 3.2 shows a conceptual schema of the different modules in the program. The climates and building modules that simulate the outdoor and indoor conditions are under modelling and implementation. The values of indoor temperature, outdoors temperature and indoor moisture used in the actual model have been acquired from the measurements carried out in two supermarkets in two cities in Sweden.

CyberMart is built in Delphi that is an application development product for writing Windows applications. Delphi is an object-oriented programming language with a database function. In CyberMart it is possible to simulate seven different refrigeration system designs that are the most representative in supermarkets in Sweden. The properties of the different components of the refrigeration system as compressors, heat exchangers, display cases, and others, are in a database that is called from different modules in the program in order to simulate the variation of the refrigeration system in different conditions.

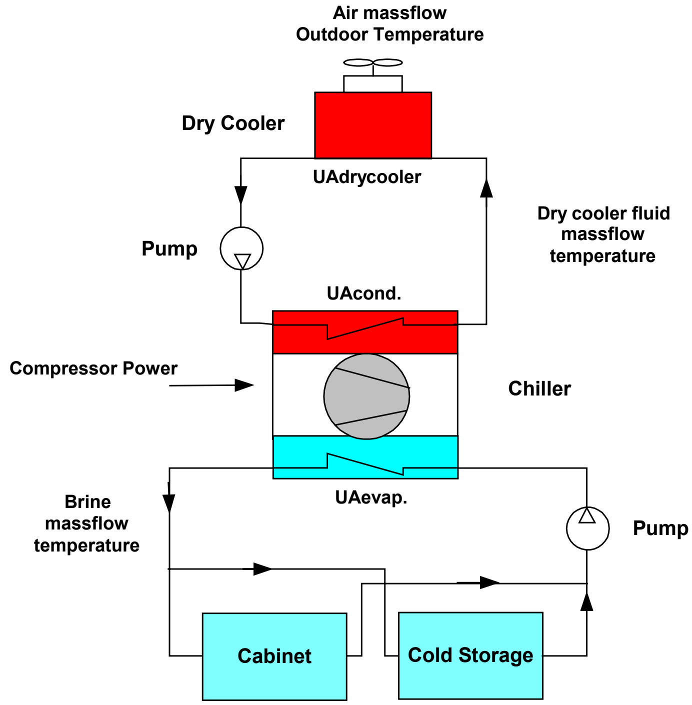

Figure 3.3: The basic model of the indirect system in CyberMart

The refrigeration systems capable of being simulated in CyberMart are solutions either with direct or indirect systems. The models of the indirect and direct systems are presented in figures 3.3 and 3.4.

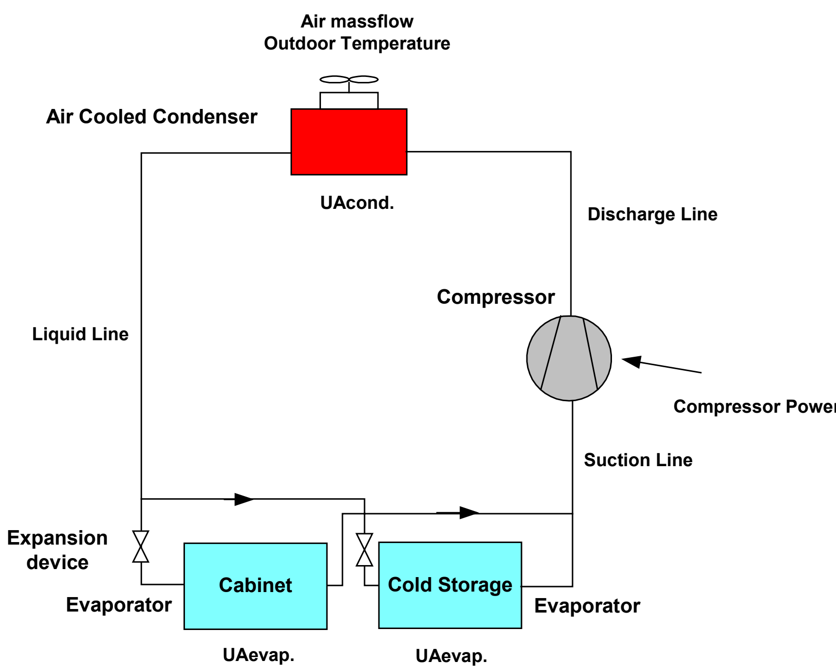

Figure 3.4: The basic model of direct system in CyberMart

3.2. Indirect System Modelling

The components of the indirect system that are simulated in CyberMart are presented in figure 3.3. The most important device in the model is the chiller that is represented by four components, condenser, expansion valve, evaporator and compressor. The components that influence the operating condition of condenser and evaporator are cabinets and dry coolers. Figure 3.5 is an attempt to illustrate the temperature levels and the energy flow through an indirect system.

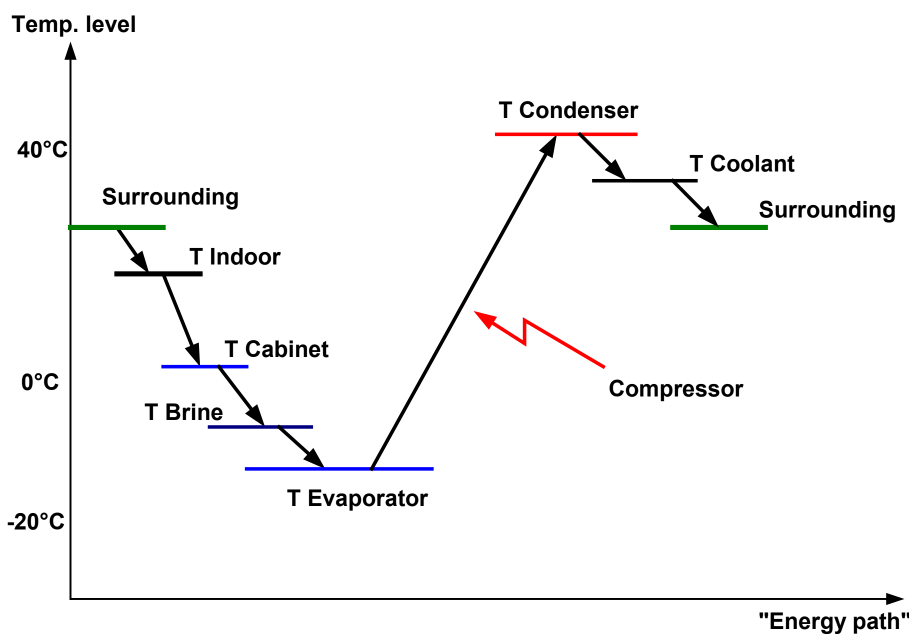

Figure 3.5: Temperature levels and "energy path" of an indirect system.

3.2.1. Cabinets and deep-freeze cabinets

The performance data of the cabinets and deep-freeze cabinets in CyberMart have been taken from manufacturers. The average air temperature, air inlet temperature, air return temperature, evaporating temperature, electrical data of fans, heating wires, defrost heaters and light, coil volume, diameter of tubes and refrigeration load at 22°C – 65% RH and at 25°C – 60% RH of every cabinets have been put into a database. The refrigeration loads in display cases are dependent on indoor conditions in the supermarket [11]. Higher indoor temperature and relative humidity increase the cooling demand and the energy requirement. An energy balance of an open vertical cabinet is shown in figure 3.6 where heat losses from infiltration, radiation, conduction, lighting, fan, heating wires and defrost are presented. The figure shows also the interaction between the ambient condition in the supermarket and the interior condition in vertical cabinet. The heat losses dependent on the ambient condition in supermarkets are infiltration, radiation, conduction and defrost. The losses from infiltration are about 66 % of the total refrigeration load at 25°C.

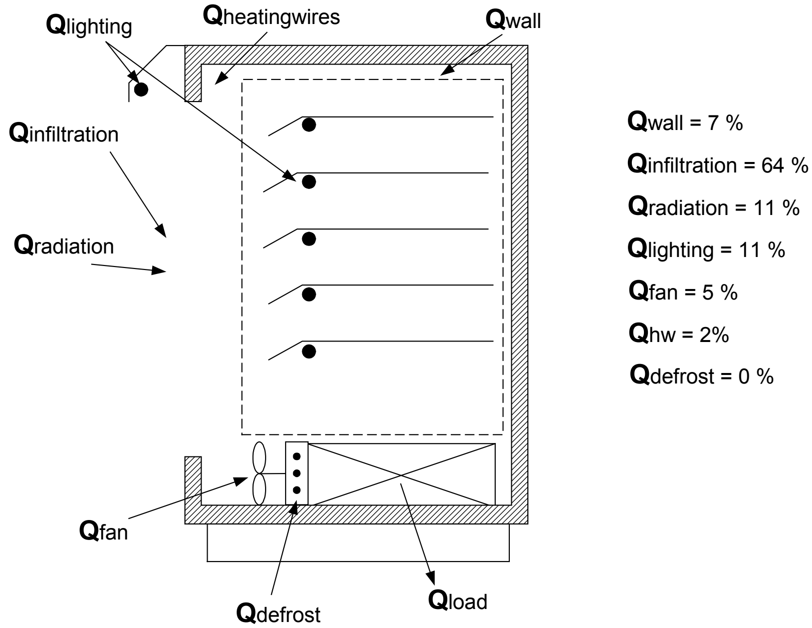

Figure 3.6: Energy balance of an open vertical display case

The effect of the indoor temperature on refrigeration load in vertical display cases is presented in figure 3.7. The values in the diagram have been calculated according to the heat balance in figure 3.6 at different indoor temperatures [11]. The diagram shows that at 14°C the refrigeration load will be halved and at the same temperature in the cabinet and in the surrounding the refrigeration load is equivalent to heat from the illumination, heating wires and fan that is about 18% of the total refrigeration load at 25°C.

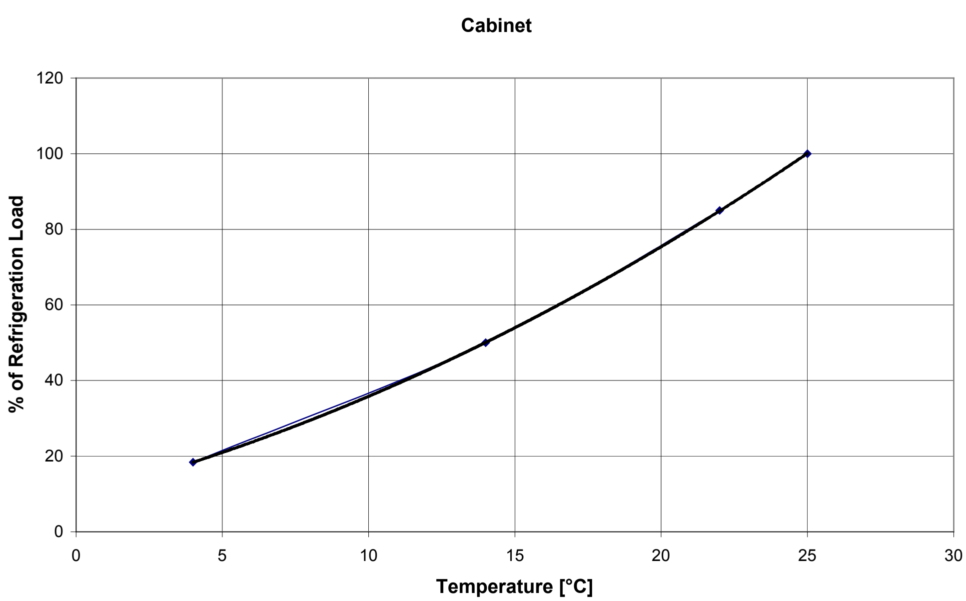

Figure 3.7: Effect of indoor temperature on refrigeration load in vertical display cases

The effect of the indoor temperature on the refrigeration load in a horizontal deep-freeze cabinet has been calculated according to the same reasoning as for vertical display cases. An energy balance of a horizontal cabinet is shown in figure 3.8 where heat losses from infiltration, radiation, conduction, lighting, fan, heating wires and defrost are presented. The heat losses dependent on the ambient condition in supermarkets are infiltration, radiation, conduction and defrost. The losses from radiation are about 46 % of the total refrigeration load at 25°C.

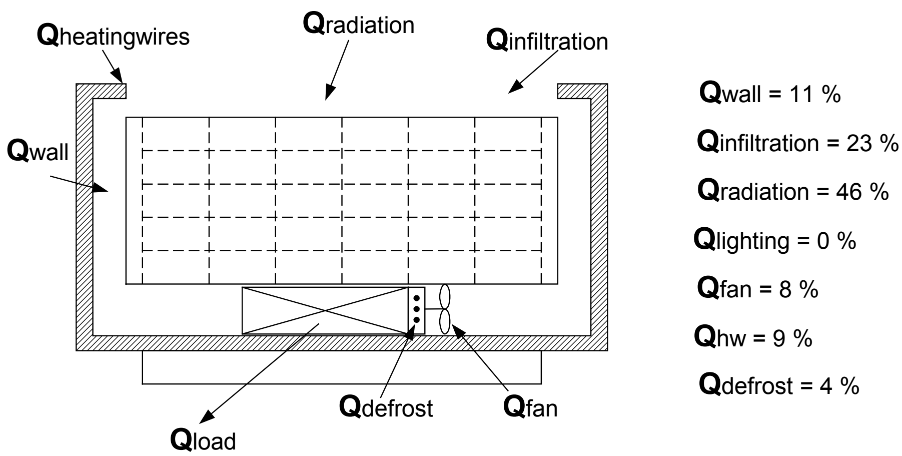

Figure 3.8: Energy balance of a horizontal display case

At an indoor temperature of 25°C the refrigeration load is equivalent to 100% and at the same temperature in the cabinet and in the surrounding the refrigeration load is equivalent to the heat from heating wires, defrost and fan that is about 20% of the total refrigeration load at 25°C.

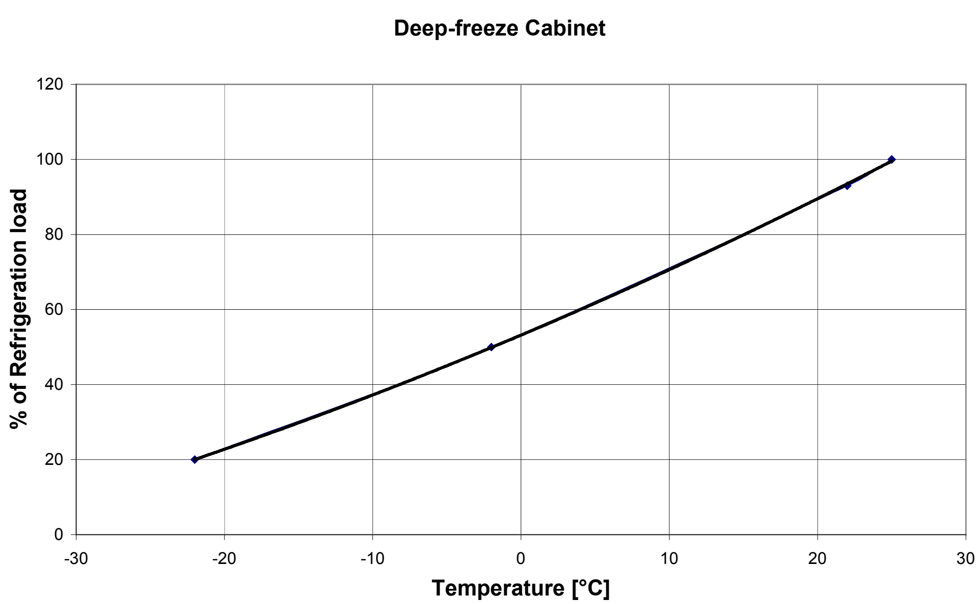

Figure 3.9: Effect of indoor temperature on refrigeration load in horizontal display cases

The influence of indoor moisture on the refrigeration load of display cases has been calculated according to [17]. The correction factor TP for the total heat transfer of the cabinets, when the display case temperature TC is known, has been defined as:

$$TP = 1 - (55 - RH) \cdot [D + E \cdot TC + F \cdot TC^2 + G \cdot TC^3] \tag{3.1}$$

The coefficients D, E, F and G for vertical display cases have been defined as:

D = 7.3867 E-3
E = 6.510 E-5
F = -4.6164 E-7
G = 7.2405 E-8

And the coefficients D, E, F and G for horizontal display cases are:

D = 6.577 E-3
E = 4.8668 E-5
F = 5.3562 E-7
G = 3.7393 E-9

3.2.2. Cold Storage

The capacity demand of cold storage is due to four factors, heat transmission, exchange of air, cooling or freezing of products and internal heat generation [15].

Heat transmission through walls, floor and ceiling is dependent on the overall heat transfer coefficient and the temperature difference between the room and the surroundings. The heat transmission has been defined as:

$$\dot{Q}_{Tr} = \Sigma\left(U_n \cdot A_n \cdot \left(t_{sur} - t_{room}\right)\right) \tag{3.2}$$

where
U_n: is the overall heat transfer coefficient of the walls, floor and ceiling
A_n: is the equivalent area of envelope element
t_{sur}: is the surrounding temperature
t_{room}: is the room temperature

The exchange of air in cold rooms depends on the frequency of door openings and the size of the room. The exchange of air increases the refrigeration load of the room. The influence of incoming air in the room can be calculated as:

$$\dot{Q}_{airex} = \dot{V}_{airex} \cdot \rho \cdot \left(h_{sur} - h_{room}\right) \tag{3.3}$$

Where
ρ is the air density at room temperature
h_{sur} is the enthalpy at ambient condition

h_{room} is the enthalpy at room condition

$\dot{V}_{airex}$ is an average volume flow of incoming air that is defined in [15] as:

$$\dot{V}_{airex} = V_{room} \cdot \frac{n_{24}}{24 \cdot 3600} \tag{3.4}$$

Where
V_{room} : is the volume of the room [m^3]
n_{24} is the number of air exchanges by 24 hours in the room. Temperatures and the frequency of door openings influence the number of air exchanges. Results from experiments are presented in the following table [14]

Room Volume	V_{room}	7	10	20	40	100	500	1000	3000
Room temp. > 0°C	n_{24}	38	31,5	21,5	14,5	9	3,5	2,5	1,35
Room temp. < 0°C	n_{24}	30	24,5	17	11,5	7	2,7	2,7	1,05

Table 3.1: Number of air exchanges from experiments.

The refrigerant load required to cool or freeze products is dependent on the mass flow rate and the enthalpy difference of the products.

$$\dot{Q}_p = \dot{m}_p \cdot \left(h_{pin} - h_{proom}\right) \tag{3.5}$$

Where
h_{pin} is the enthalpy of products before the cooling or freezing process.
h_{pout} is the enthalpy of products after the cooling or freezing process.
$\dot{m}_p$ is mass flow rate of product.

The enthalpy difference for freezer rooms has been assumed at 45 [kJ/kg] that is the average between the enthalpies of different products at temperatures -7°C and –18°C. In the same manner the enthalpy difference for the cold room has been assumed as 55 [kJ/kg] that is the average between the enthalpies of different products at the temperature 17°C and 1°C. The mass flow has been assumed as 20 kg by m^3 in 24 hours [6] for cold rooms and 15 kg by m^3 in 24 hours for freezer rooms.

Internal heat generation from lighting and people also affect the refrigeration load of the cold room. The heat generated by lighting has been assumed at 15 [W/m^2] and the heat from people at 200 W.

3.2.3. Dry cooler and other heat exchangers

The dry cooler and other heat exchangers have been modelled as counter flow heat exchangers. The effectiveness of the counter flow heat exchanger is defined in [20] as:

$$\varepsilon = \frac{1 - e^{\left(-NTU \cdot (1 - C_r\right)}}{1 - C_r \cdot e^{\left(-NTU \cdot (1 - C_r\right)}} \tag{3.6}$$

Where NTU is the number of transfer units defined for the counter flow heat exchanger as:

$$NTU_{HEX} = \frac{UA_{HEX}}{C_{min}} \tag{3.7}$$

C_{min} is defined as the minimum between the hot and cold fluid heat capacity rates. The cold heat capacity rate is the product of the mass flow and the heat capacity of the cold fluid

$$C_c = \dot{m}_c \cdot cp_c \tag{3.8}$$

The hot heat capacity rate is the product of the mass flow and the heat capacity of the hot fluid

$$C_h = \dot{m}_h \cdot cp_h \tag{3.9}$$

The Heat Capacity ratio C_r in equation (3.6) is defined as the ratio between the minimum and maximum fluid heat capacity rates between the hot and cold fluid heat capacity rates.

$$C_r = \frac{C_{min}}{C_{max}} \tag{3.10}$$

3.2.4. Pressure drop and pumps

The pressure drops in the secondary refrigerant loop and in the dry cooler fluid loop for both the medium and low temperature systems are dependent on the type of fluid used and on the thermophysical properties of the fluid. The Reynolds number Re is the decisive parameter to decide the type of fluid. The Reynolds number is defined [24] as:

$$Re = \frac{w \cdot d}{\nu} \tag{3.11}$$

Values of the Reynolds number above 2000 - 2300 generally means turbulent flow. Re lower than this value means laminar flow. The pressure drop can be calculated with the following relation [24]:

$$\Delta pf = \frac{f_1 \cdot \rho \cdot w^2 \cdot L}{d} \tag{3.12}$$

Where
w: is the velocity of the fluid in [m/s]
L: is the length of tube in [m]
ρ: is the density of fluid in [kg/m^3]
d: is the tube diameter in [m]
f_l: is the friction factor

The friction factor is defined for turbulent fluid as

$$f_l = \frac{0.092}{Re^{0.2}} \tag{3.13}$$

And for laminar flow the friction factor is defined as:

$$f_l = \frac{32}{Re} \tag{3.14}$$

The condensers and evaporators in indirect systems are plate heat exchangers. The pressure drops in the brine side of the evaporator and the dry cooler fluid side of the condenser have been estimated according to the following relations [16]:

$$\Delta p_{hex} = \frac{2 \cdot f \cdot N_p \cdot \dot{m}^2 \cdot L}{\rho \cdot d_e} \tag{3.15}$$

where
f is the friction factor calculated from performance data
N_p is the number of passes
$\dot{m}$ is the mass velocity of fluid [kg/m^2s]
L is the effective lenght [m]
ρ is the density of fluid [kg/m^3]
d_e is the effective diameter [m].

The pump power has been calculated by

$$P_p = \frac{\Delta p_{tot} \cdot \dot{V}}{\eta_{pump}} \tag{3.16}$$

where
Δp_{tot} : is the sum total of Δp_f, Δp_{hex} and $\Delta p_{cabinet}$
$\Delta p_{cabinet}$ pressure drop in cabinet that has been calculated with equation (3.12) and data from manufacturers.
$\dot{V}$: is the volume flow of fluid.
η_{pump}: is the pump efficiency

3.2.5 Condenser and evaporator model

The condenser and the evaporator in indirect systems are plate heat exchangers that have been modelled as classical heat exchangers. The refrigeration side of the evaporator and condenser has been assumed to having a constant temperature. The properties of the secondary refrigerant and the dry cooler fluid have been taken from Melinder [24] and data from manufacturers. The effectiveness of the condenser and evaporator is defined [20] as:

$$\varepsilon = 1 - e^{(-NTU)} \tag{3.17}$$

NTU is the number of transfer units defined for the evaporator as:

$$NTU_{evaporator} = \frac{UA_{evaporator}}{\dot{m}_{brine} \cdot cp_{brine}} \tag{3.18}$$

Where
$UA_{evaporator}$: heat transfer coefficient of the evaporator [W/K]
$\dot{m}_{brine}$: mass flow of the brine [kg/s]
cp_{brine} : specific heat of the brine [J/kgK]

And for the condenser as:

$$NTU_{condenser} = \frac{UA_{condenser}}{\dot{m}_{drycoolerfluid} \cdot cp_{drycoolerfluid}} \tag{3.19}$$

Where
$UA_{condenser}$: heat transfer coefficient of the condenser [W/K]
$\dot{m}_{drycoolerfluid}$: mass flow of the dry cooler fluid [kg/s]
$cp_{drycoolerfluid}$: specific heat of the dry cooler fluid [J/kgK]
The values of $UA_{condenser}$ and $UA_{evaporator}$ have been also taken from manufacture performance data and assumed to be constant during the simulation. The values of $UA_{condenser}$ and $UA_{evaporator}$ in the database correspond to the refrigerant R404A and Propylene Glycol as brine or dry cooler fluid in the evaporator or in the condenser respectively. For other secondary refrigerants or dry cooler fluids than Propylene Glycol the relation in (3.20), which calculate the new UA value, have been used.

$$\frac{1}{U_{evaporator}} = \frac{1}{\alpha_{ref}} + \frac{t}{\lambda_{met}} + \frac{1}{\alpha_{brine}} \tag{3.20}$$

Where
$U_{evaporator}$: heat transfer coefficient of the evaporator
α_{ref} : heat transfer coefficient on the refrigerant side of the evaporator assumed to be constant.
t : thickness of the plate.
λ_{met} : thermal conductivity of the metal.
α_{brine} : heat transfer coefficient on the brine side of the evaporator.

The heat transfer coefficient α_{brine} of the new brine has been calculated with the following relation defined in [16] as:

$$\alpha_{brine} = \frac{Nu \cdot \lambda_{brine}}{de} \tag{3.21}$$

where
λ_{brine} is the thermal conductivity of the brine
de is the effective diameter of the heat exchanger
Nu is the Nusel number defined for the plate heat exchanger and turbulent flow as

$$Nu = 0.2 \cdot Re^{n} \cdot Pr^{0.4} \cdot \left(\frac{\mu}{\mu_w}\right)^{0.17} \tag{3.22}$$

and for laminar flow as

$$Nu = 0.29 \cdot (Re \cdot Pr)^{0.4} \cdot \left(\frac{\mu}{\mu_w}\right)^{0.1} \tag{3.23}$$

where
n : exponent calculated from the performance data
μ : bulk dynamic viscosity
μ_w : wall dynamic viscosity
Pr : Prandtl number defined as

$$Pr = \frac{\mu \cdot c_p}{\lambda_{brine}} \tag{3.24}$$

c_p is specific heat of the brine
Re : is the Reynolds number defined as

$$Re = \frac{w \cdot d}{\nu} \tag{3.25}$$

w: is the velocity of the fluid.
d: is the tube diameter.
ν: is the cinematic viscosity.

3.2.6. Compressor Model

The compressor model has been developed from the performance data. The compressor manufacturers have developed different software where compressor power, refrigeration capacity and mass flow are given for different evaporating and condensing temperatures. The data of compressor power and refrigeration capacity has been introduced in a database as two matrices of three lows and three columns equivalent to condensing temperatures of 30°C, 40°C and 50°C and evaporating temperatures of -15°C, -10°C and -5°C. The matrix of compressor 2N-5.2Y is shown in figure 3.5.

The variation of the compressor power is dependent on the variation of the indoor and outdoor climate that can influence the refrigeration load of the evaporator and the capacity of the compressor. The simulation of the compressor power occurs by interpolation and/or extrapolation between the condensing and evaporating temperatures.

Compressor 2N-5.2Y 13,8 kW

$$\begin{array}{cc} & \begin{array}{ccc} 30 & 40 & 50 \end{array} \\ \begin{array}{c} -15 \\ -10 \\ -5 \end{array} & \begin{bmatrix} 14{,}8 & 12{,}5 & 9{,}9 \\ 18{,}5 & 15{,}7 & 12{,}7 \\ 22{,}7 & 19{,}5 & 15{,}9 \end{bmatrix} \end{array}$$

Figure 3.10: Matrix with performance data of refrigeration capacity of Compressor 2N-5.2Y

3.3. Direct System Modelling

The traditional refrigeration system design in supermarkets is the direct system. The characteristics of the system are long lines of refrigerant between the compressor, evaporator and condenser that affect the total refrigerant charge in the system and the potential of refrigerant charge. Figure 3.4 presents the components of the direct system that are simulated in CyberMart. The most important device in the model is the compressor that has been modelled as the compressor in the indirect system. The values of compressor powers and refrigeration capacities have been introduced in a database for different condensing and evaporating temperatures and the simulation of the compressor power has been made by interpolation and/or extrapolation between the condensing and evaporating temperatures.

3.3.1. Condenser and evaporator model

The condenser and the evaporator have been modelled as classical heat exchangers. The condensers are an air-cooled condenser type and the evaporators are the fincoils in cabinets and in the air coolers. The refrigeration side of the evaporator and condenser has been assumed with constant temperature.

The effectiveness of the condenser and evaporator have been defined in [20] as:

$$\varepsilon = 1 - e^{(-NTU)} \tag{3.26}$$

Where NTU is the number of transfer units defined for the evaporator as:

$$NTU_{evaporator} = \frac{UA_{evaporator}}{\dot{m}_{air} \cdot cp_{air}} \tag{3.27}$$

And for the condenser as:

$$NTU_{condenser} = \frac{UA_{condenser}}{\dot{m}_{air} \cdot cp_{air}} \tag{3.28}$$

The values of $UA_{condenser}$ and $UA_{evaporator}$ have also been taken from manufacturer performance data and assumed to be constant during the simulation.

3.3.2. Pressure drop in gas and liquid lines

The pressure drops in vapour and liquid lines in the refrigeration system can be calculated from the following equation:

$$\Delta p = \frac{f_1 \cdot \rho \cdot w^2 \cdot L}{d} \qquad (3.29)$$

The friction factor can be found as a function of the Reynolds number in equation (3.13) and (3.14). Often it is convenient to know the pressure drop in an equivalent temperature drop for the refrigerant in order to see the effect of the pressure drop in the evaporating temperature.

Granryd has used the Clapeyron equation to convert the pressure drop from Pa to an equivalent change in saturation temperature [14].

$$\Delta t'' = T \cdot \frac{(v''-v')}{r} \cdot \Delta p \qquad (3.30)$$

Where
T: is the absolute temperature of the fluid in [°K]
v’’: is the specific volume for saturated vapour in [m^3/kg]
v’: is the specific volume for saturated liquid in [m^3/kg]
r: is the latent heat of vaporization [KJ/kg]

3.4. Life Cycle Cost (LCC)

The Life Cycle Cost is an economic method that evaluates present and future costs from the investing, operating and maintaining of a project over its life cycle. To calculate the LCC it is necessary to compute the present value of all costs during the period of study (usually related to the life of the project). The costs during the life cycle are of two types: single costs and annually recurring costs. The single costs occur one or more times during the study period as investment cost, repair cost, etc. The annually recurring costs occur regularly every year as energy cost, annual maintenance cost, etc.

The present value of single costs should be calculated by the Single Present Value formula

$$SCp_n = \frac{SC_n}{[1+(i-p)]^n} \qquad (3.31)$$

Where
SCp_n : is the present value of a single cost SC_n after n years.
SC_n : single cost after n years.
n : is the amount of year after year 0
i : is the discount rate
p : is the inflation or the price escalation

The present value of annually recurrent costs should be calculated by Uniform Present Value formula

$$RCp_n = RC_n \cdot \frac{1-\lfloor 1+(i-p)^{-n} \rfloor}{(i-p)} \qquad (3.32)$$

Where
RCp_n : is the present value of an annually recurrent cost RC_n
SC_n : annually recurrent cost.
n : is the period of study
i : is the discount rate
p : is the inflation or the price escalation

The LCC formula [25] used in CyberMart is

$$LCC_{TOTAL} = Inv + LCC_{ENERGY} + LCC_{OM\&R} + LCC_{Environment} + LCC_{others} \qquad (3.33)$$

where
Inv: Present value of investment costs that include, for instance, costs of products, installation, administration, etc.
LCC_{ENERGY}: Present value of annually energy costs calculated in CyberMart.
$LCC_{OM\&R}$: Present value of non-fuel operating, maintenance and repair cost.
$LCC_{Environment}$: Present value of environmental costs.
LCC_{Others}: Present value of other costs calculated.

3.5. Total Equivalent Warming Impact (TEWI)

Global environmental impacts as ozone depletion and greenhouse gas emission to the atmosphere have been associated with refrigeration systems during the last years. The influence on the ozone from refrigerants CFC and HCFC is well known and stronger legislation has been implemented to decrease the use of these refrigerants. Refrigerants CFC and HCFC are also recognised as greenhouses gases since these primarily substitute the refrigerant HFC. The concept of TEWI is a useful tool when comparing the influence of a refrigeration system in global warming. The TEWI combines the direct emissions of CO_2 due to refrigerant leakage and the indirect emissions of CO_2 associated with energy consumption and this generation. The TEWI calculation of a refrigeration system is based on the following relation:

$$\mathrm{TEWI} = \mathrm{M}_{\mathrm{losses}} \cdot \mathrm{N} \cdot \mathrm{GWP}_{\mathrm{ref}} + \mathrm{RC} \cdot \mathrm{E} \cdot \mathrm{N} \qquad (3.34)$$

Where
M_{losses} : Refrigerant leak in [kg/year]
N : equipment operation time [year]
GWP_{ref} : Global Warming Potential of Refrigerant in [kg CO_2/kg refrigerant]
RC : Regional Conversion Factor is the emission of CO_2 per unit of energy delivered in [kg CO_2/kWh]. The best value of RC [Fischer] for Sweden is about 0.04[kg CO_2/kWh], for Denmark it is about 0.84[kg CO_2/kWh], for Norway about 0.00[kg CO_2/kWh] and for Finland about 0.24[kg CO_2/kWh]
E : Annual energy consumption of the equipment [kWh/year]

3.6. Results from model

CyberMart is a day-to-day simulation program that allows the user to see the variation of different variables such as compressor power, refrigeration capacity and temperatures in the supermarket during one year. The users interface of CyberMart is shown in figure 3.11.

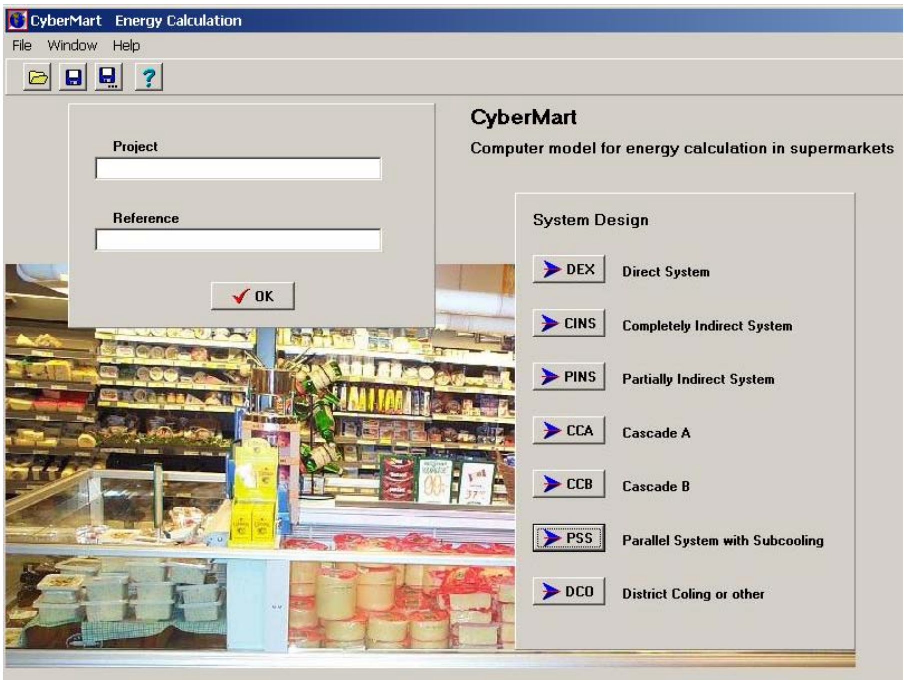

Figure 3.11: User interface for CyberMart with current system solutions.

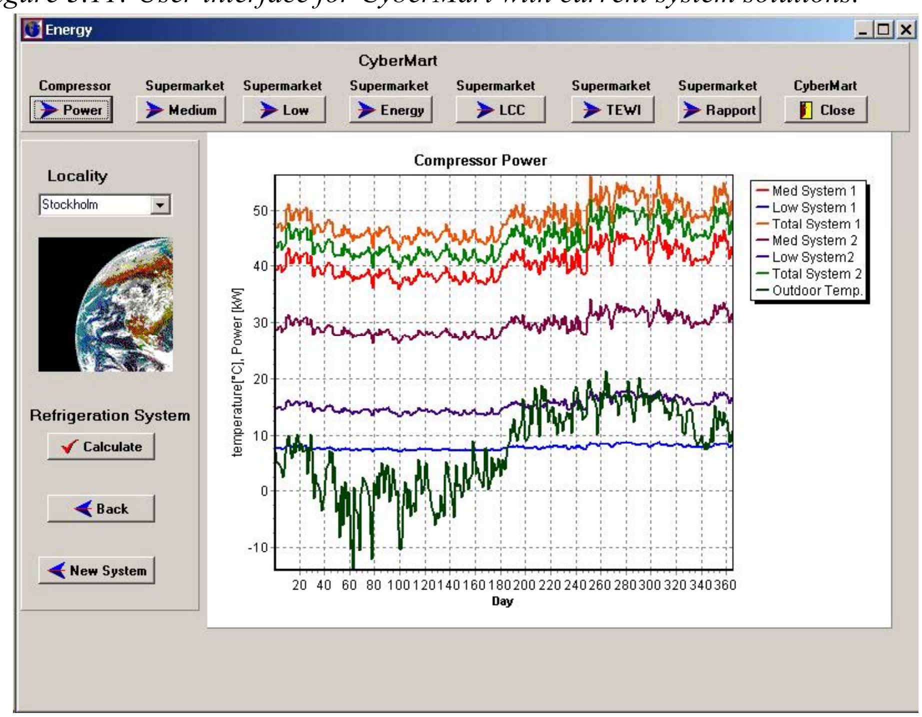

Figure 3.12: Results from CyberMart.

The window for calculation of energy consumption is shown in figure 3.12. Here the user will select the locality of the supermarket. The program starts the calculation of the energy consumption of the refrigeration system design from the button Calculate When the calculation has finished a window with eight buttons, Power, Medium, Low, Energy, LCC, TEWI, Rapport and Close and a diagram will appear. The different buttons open new windows for start new calculation, as buttons LCC and TEWI, or to show results as the buttons Power, Medium, Low, Energy and Rapport. The diagram shows the results from the simulation of the compressor power of the intermediate and low temperature systems during one year.

Figure 3.13 illustrates results obtained from simulation and field measurements from one of the supermarkets included in the study. The time in this particular study is one year. The difference in calculated compressor power between simulation and model is depicted in the figure (the bottom curve) and the yearly energy balance differs with 6.8%. It is clear that the general characteristics are relatively well recreated in the software.

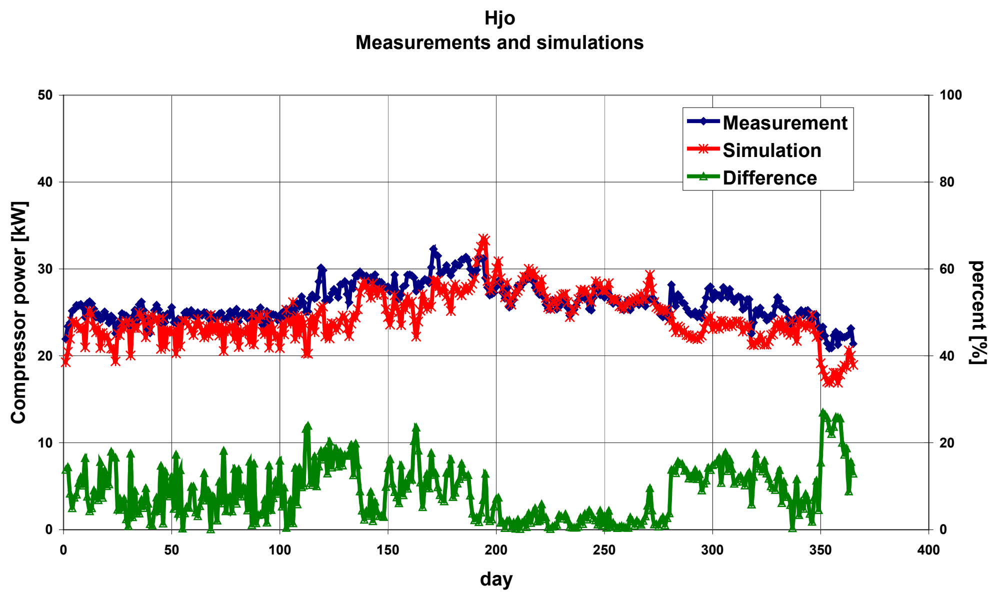

Figure 3.13: Comparison of measurements and simulations for one supermarket included in our study.

One idea behind the development of the software Cybermart is to facilitate comparisons of different solutions for the same supermarket. Results from simulations are presented in figure 3.14 where two different systems are compared. The systems are a cascade system and a parallel system with mechanical sub-cooling. The results indicate that a parallel system with mechanical sub cooling is more energy efficient than the cascade system. Mechanical sub-cooling means that the secondary refrigerant on the medium temperature side is used to sub-cool the refrigeration system of the low temperature circuit. The effect of this is dependent on the chosen refrigerant in this case, R404A, almost 30% better capacity is achieved on the freezer side. The COP_2 of the high temperature system is also more energy efficient than the low temperature system

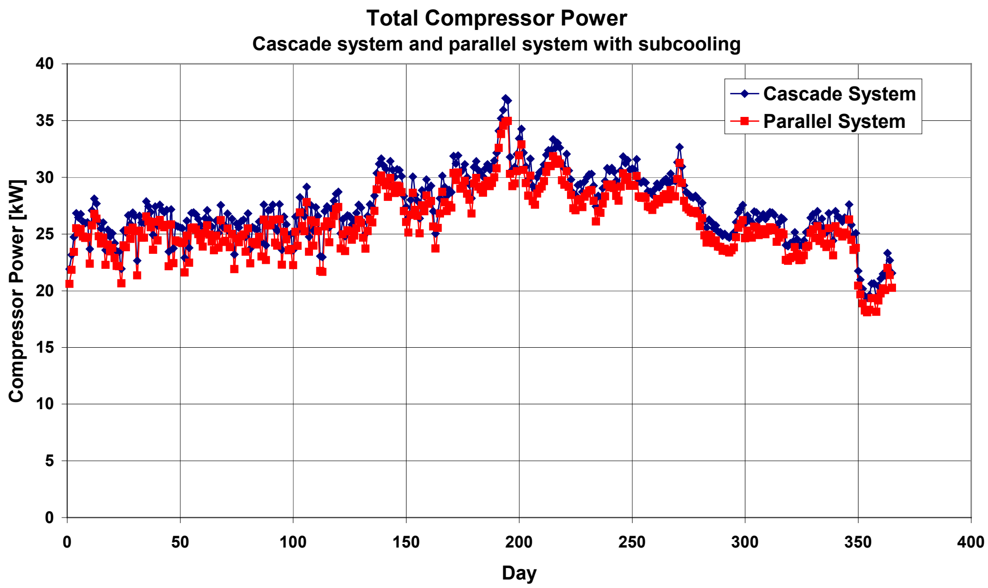

Figure 3.14: A comparison of the solutions Cascade system and Parallel system with mechanical subcooling.

4. Measurements

4.1 Overview

Field measurements in four supermarkets in Sweden have been carried out to validate the computer model CyberMart. The measurements have been divided into three different periods to reduce the amount of measurements and to cover the most important parameters in supermarkets [2].

In Period 1 (during one year), measurements have been done on the outdoor temperature, indoor temperature, indoor air relative humidity, brine temperature before and after the chiller and the compressor power of the medium temperature system. In Period 2 (during one week) measurements have been done on the air temperature in the inlet, middle and outlet of a display case and the compressor power of the low temperature system. In Period 3 (during one hour) measurements have been done on the air temperature in the inlet, middle and outlet of a display case.

The refrigeration system design of the supermarket in the city of Sala is a cascade system (figure 4.1). The brine in the intermediate temperature system cools the display cases and the condenser of the chiller for the low temperature system. In this supermarket there is a heat recovery system.

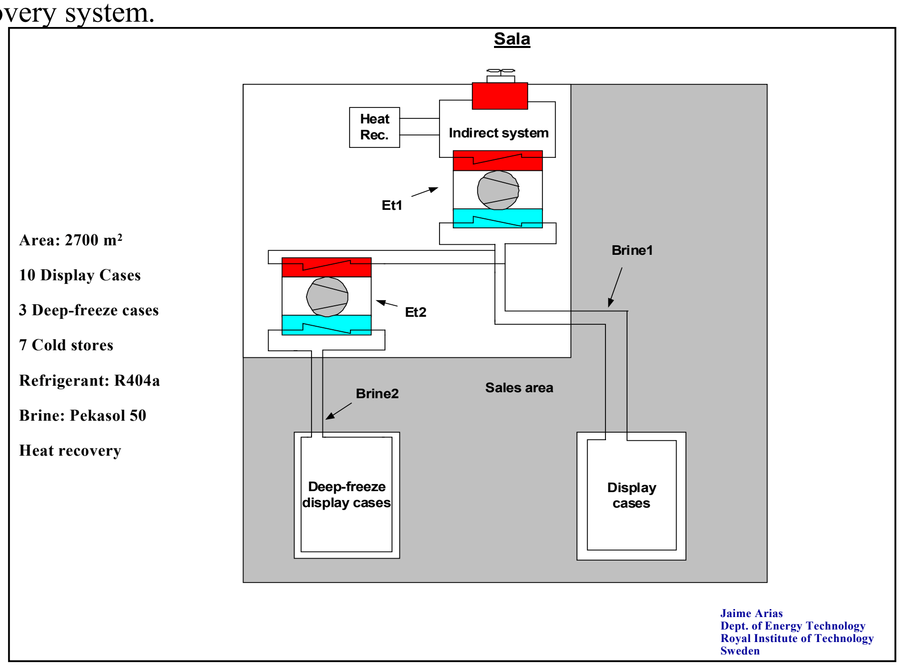

Figure 4.1: Refrigeration system design in Sala

The system design in the supermarket Gröna Konsum in the city of Hjo is also a cascade system (figure 4.2). In this case there is a refrigeration system in every deep-freeze cabinet. The condensers for these machines and the display cases are cooled with a chiller situated in the machine room. In this supermarket there is also a heat recovery system.

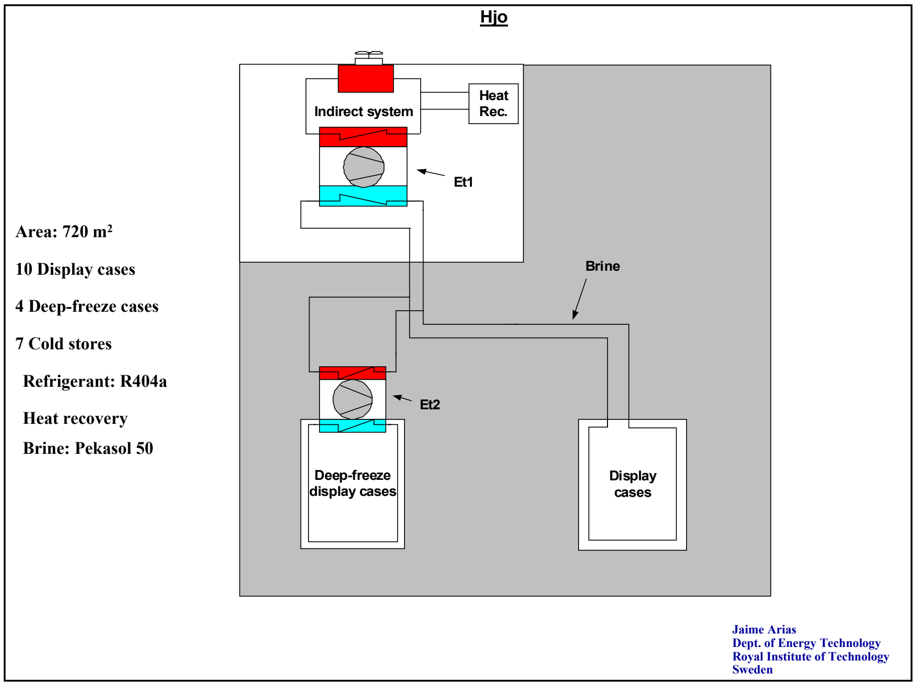

Figure 4. 2: Refrigeration system in Hjo

The system design in supermarket Gröna Konsum in Farsta Centrum has one chiller that cools the condensers of the refrigeration equipment in every deep-freeze cabinet, display case and cool storage via a brine loop. The supermarket is divided into two different zones, a cold zone for products requiring refrigeration and where all the display cases are located, and a warm zone for non-refrigerated products. The refrigeration system has 46 compressors distributed in the cold zone and the supermarket has no heat recovery system.

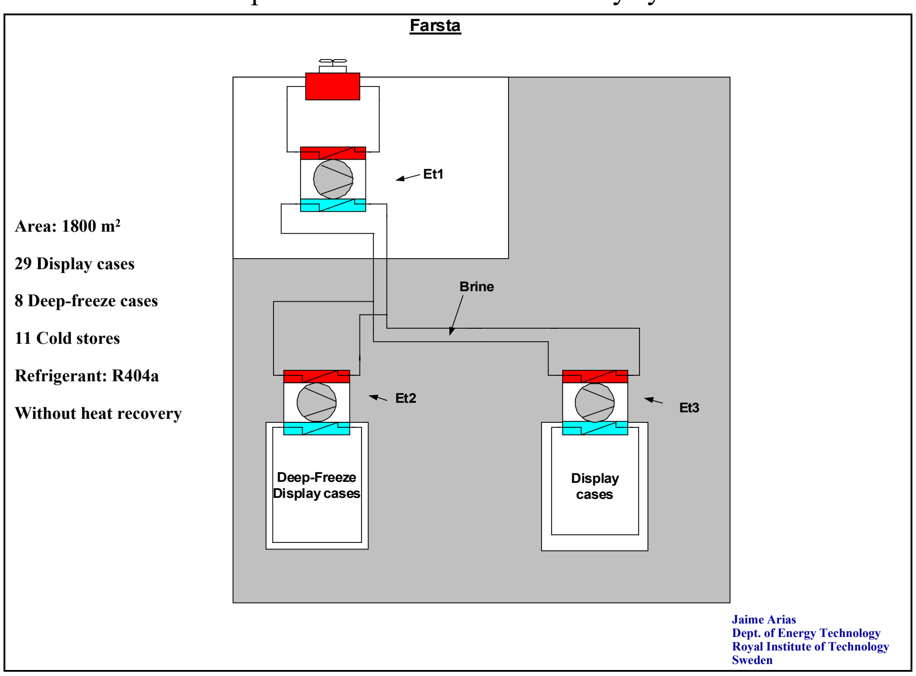

Figure 4.3. Refrigeration system design in Farsta Centrum

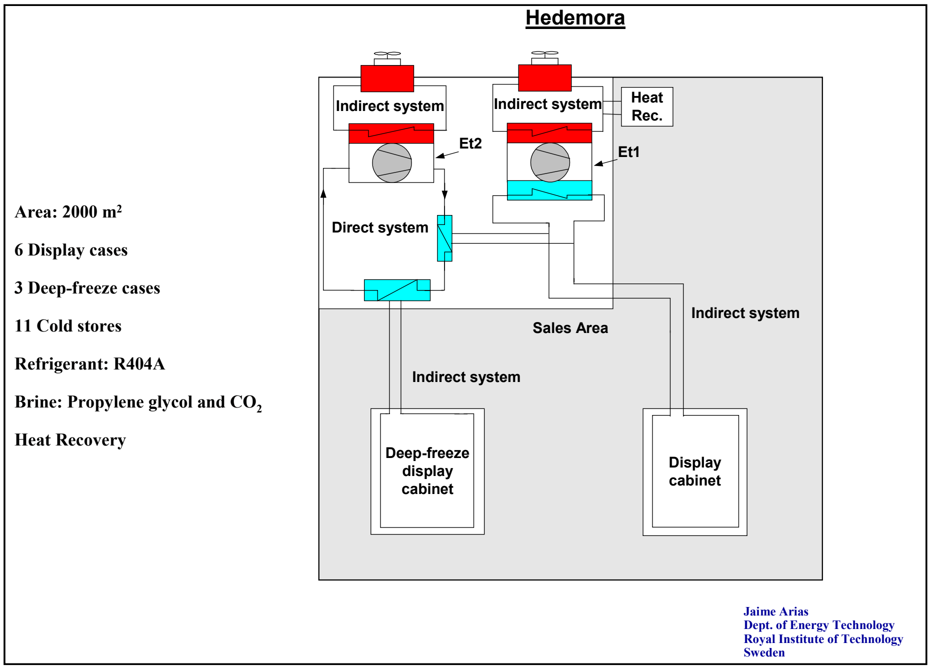

Figure 4.4. Refrigeration system design in Hedemora

The supermarket in the city of Hedemora has a refrigeration system design with parallel system and subcooling. The secondary refrigerant in the medium temperature system is propylene glycol and in the low temperature system, the secondary refrigerant is CO_2. The refrigeration capacity in the intermediate temperature level is 75 kW and 29 kW in the low temperature level.

The temperatures and air humidity have been measured with Tinytag-loggers from Intab, and the compressor power with Energy-logger Elite 4 from Pacific Science & Technology. The temperatures and relative humidity have been measured momentary and stored every hour. The compressor power is also stored every hour but in this case the value represents the mean power during the last hour.

4.2. Results

The results from the measurements confirm the importance and the influence of the outdoor temperature, indoor temperature and relative humidity of air on the compressor power. Figure 4.5 shows the average values during one day for compressor power, indoor temperature and outdoor temperature in the course of one year at Gröna Konsum in Hjo. The variations of the compressor power follow the variations of outdoor temperature for the whole year including the heating season that starts in the middle of October and ends in the middle of March. In figure 4.5 it is also possible to see the influence of the indoor temperature on the compressor power. During the period 15-Dec. to 4-Jan. a malfunctioning fan resulted in low air temperatures in the supermarket, which in turn strongly influenced the compressor power.

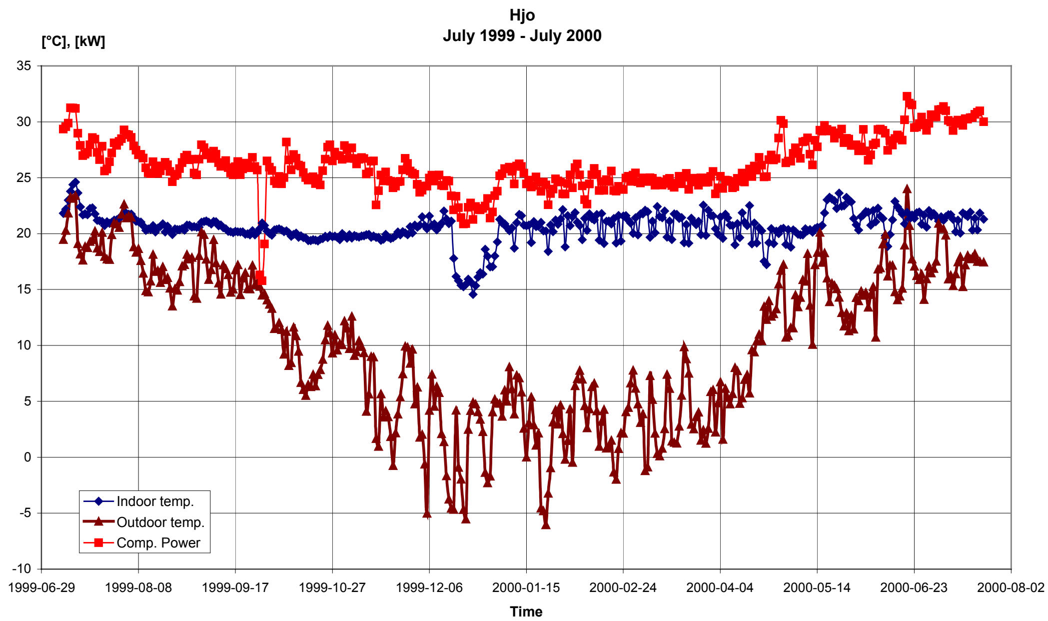

Figure 4.5: Indoor-temperature, outdoor-temperature and compressor power in Hjo

Figure 4.6, which presents the measurements in Farsta Centrum of indoor temperature in the cold and warm zones, outdoor temperature and compressor power for the chiller, corroborates the assertion about the influence of those temperatures on the compressor power. The difference between the temperatures in the cold and warm zones is about 2°C.

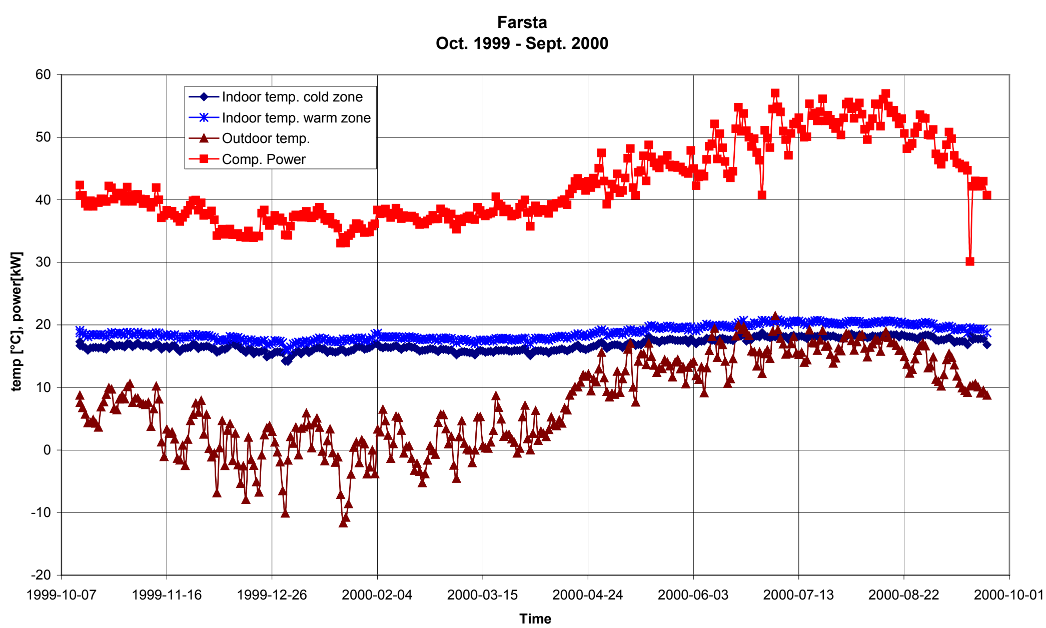

Figure 4.6: Indoor temperature, outdoor temperature and compressor power in Farsta

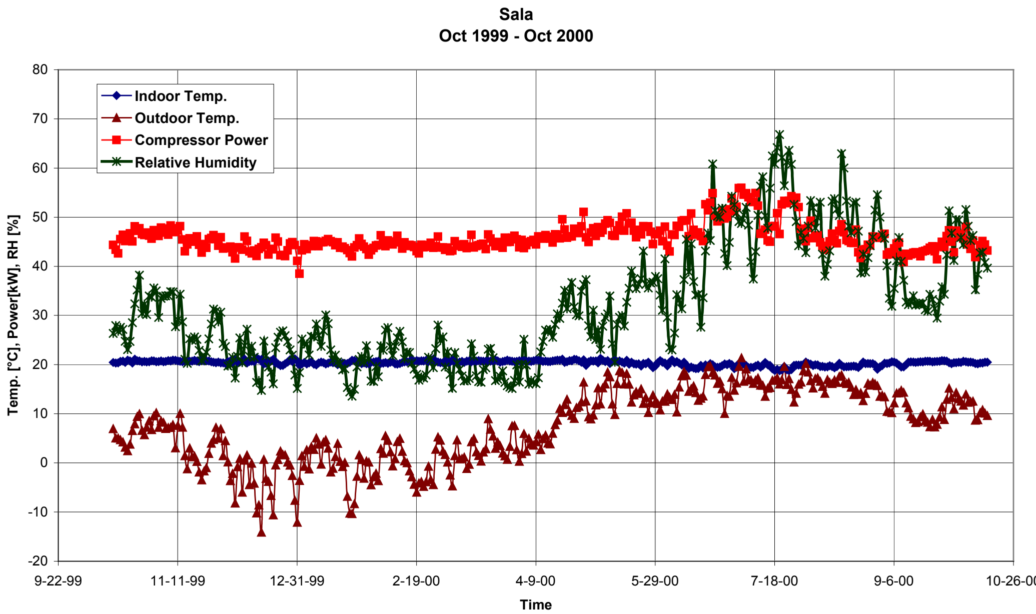

Figure 4.7: Indoor temperature, outdoor temperature, indoor relative humidity and compressor power in Sala

Results from the measurement of indoor and outdoor temperatures, relative humidity and compressor power from Sala and Hedemora are presented in figure 4.7 and 4.8. The diagrams show the influence of moisture on compressor power, and the relation between relative humidity and outdoor temperature.

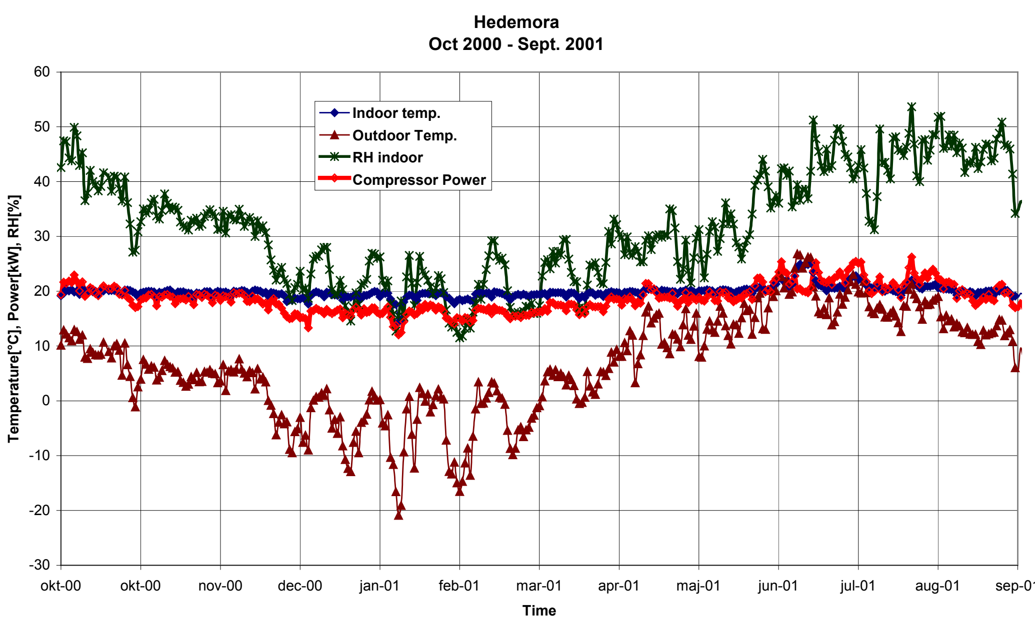

Figure 4.8: Indoor temperature, outdoor temperature, indoor relative humidity and compressor power in Hedemora

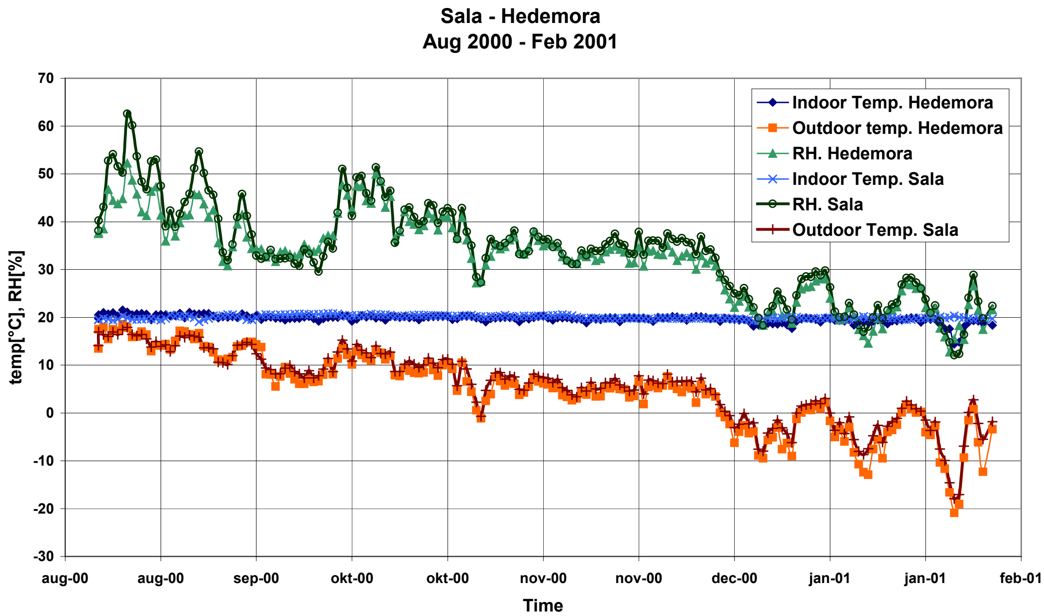

Figure 4.9: Indoor temperature, outdoor temperature and indoor relative humidity in Hedemora and Sala

Measurement of indoor temperatures, outdoor temperatures and indoor relative humidity from Hedemora and Sala are shown in figure 4.9. The distance between the cities is about 60 km. The results confirm the dependence of relative humidity on outdoor temperature. The indoor relative humidity in both supermarkets follows the variation of outdoor temperature during the period of study.

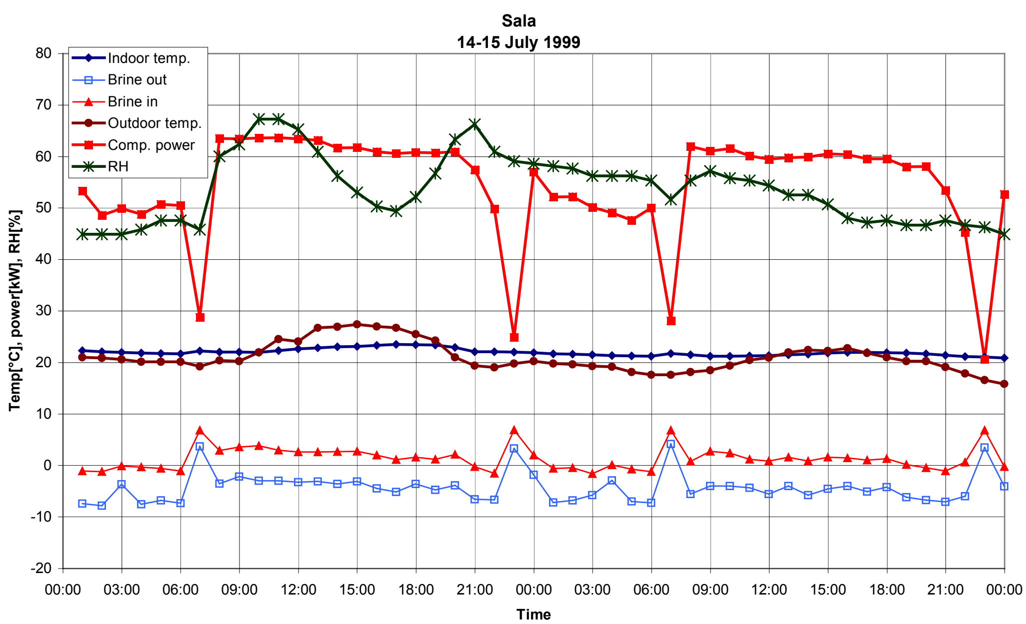

Figure 4.10: Temperatures, Moisture and Compressor Power in Sala

The influence of relative humidity on compressor power is confirmed in figure 4.10, which presents the outdoor temperature, indoor temperature, relative humidity, brine temperatures before and after chiller and the compressor power during two days. The 14th of July was a rainy and warm day in Sala that affected the moisture in the supermarket and caused the highest compressor power during the summer of 1999.

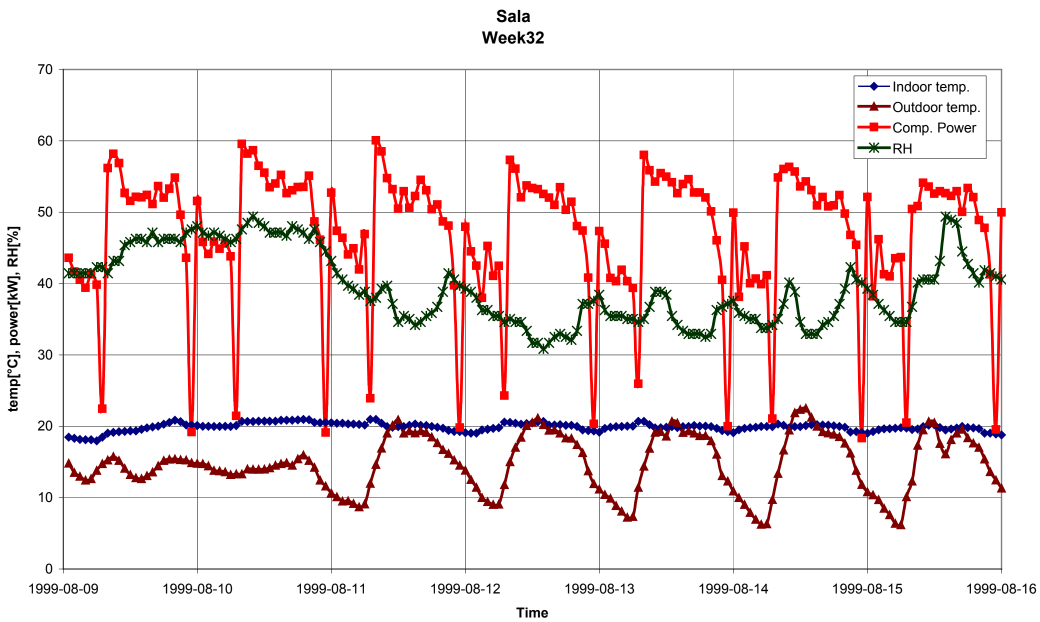

Figure 4.11: Temperatures, relative humidity and compressor power in Sala

The variations of temperatures, moisture and compressor power during week 32 in August 1999 in Sala are shown in figure 4.11. In both figures 4.10 and 4.11 it is also possible to see the influence of night covering of the display cases and defrosting on the compressor Power. The cabinets are defrosted with electrical heaters that decrease the compressor power during the defrosting, but increase the compressor power after defrosting.

According to figures 4.10 and 4.1 the compressor power is reduced between 10 and 20 % by the night covering of the cabinets. The covering occurs automatically when the supermarket closes at 21.00 and ends at 8.00 in the morning. The positive effect on energy saving of night covering is dependent on the quality of the curtains.

The measurements in the cabinet in Sala confirm the influence of night covering on temperatures in the display cases and on the compressor power. In figure 4.12, the air inlet temperature, average temperature and air return temperature of a display case, are presented. The heat extraction rate, according to the manufacturers technical data for climate class 25°C/60%RH, is 9,0 kW, the storage temperature is +1°C, the air inlet is –2°C and the air return is +5°C. During the period when the supermarket is open, the variation of temperatures agrees with the cabinets technical data. When the cabinet is covered, the air inlet temperature and the air return temperature converge toward the storage temperature.

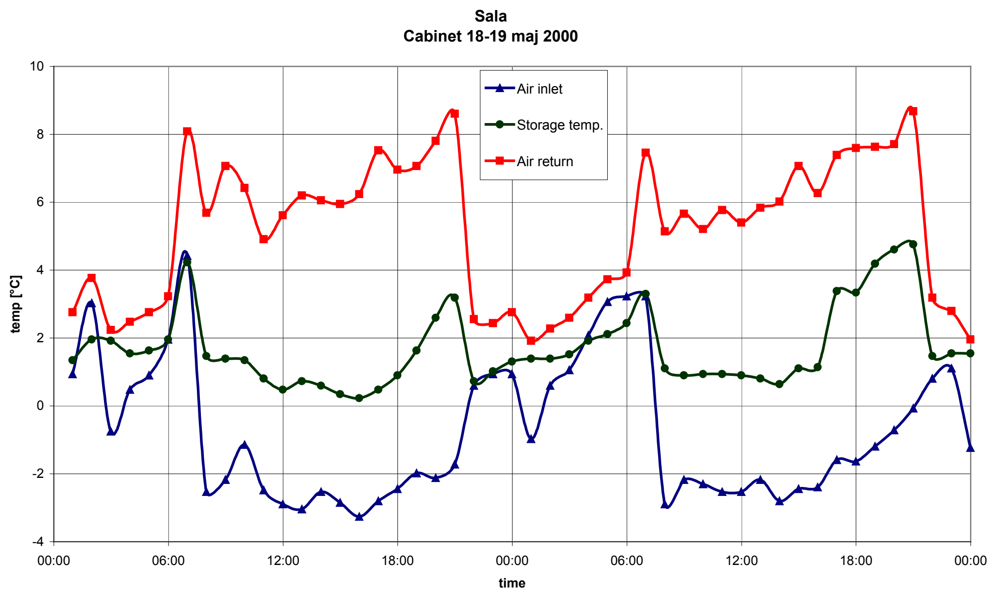

Figure 4.12: Temperatures in the cabinet in Sala

The measurements in the cabinet during Period 3 (one hour), on the 17th of May 2000 are presented in figure 4.13 and corroborate the technical data of the display case. The temperatures have been measured momentary and stored every 15 seconds. In the diagram it is also possible to see the influence of customers and employees on the storage temperature and air return temperature when they are shopping, the influence in air inlet temperature is insignificant.

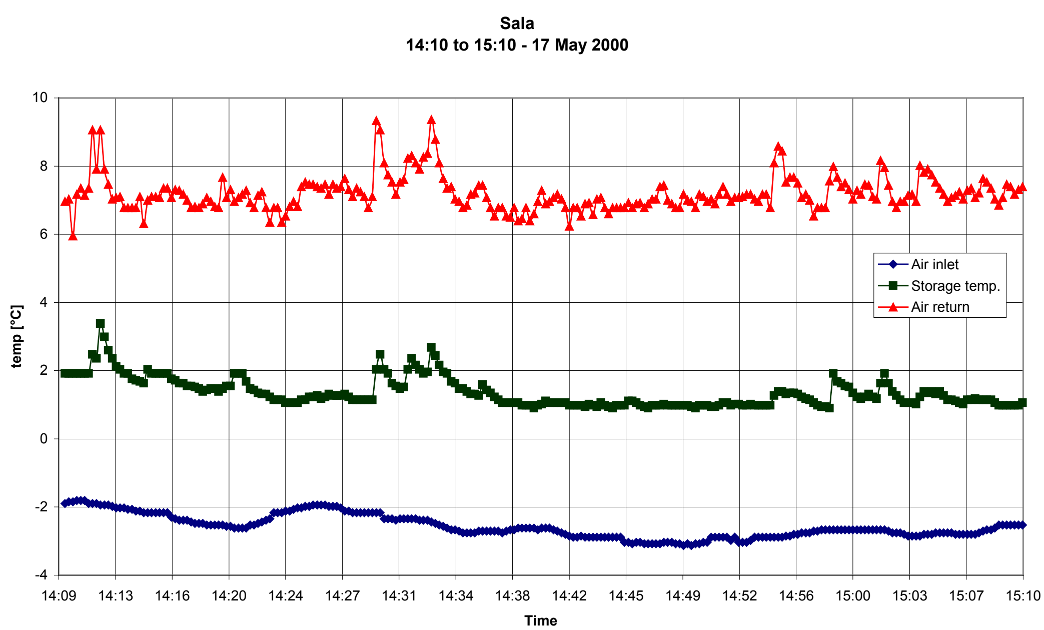

Figure 4.13: Measurements during period three in the cabinet in Sala.

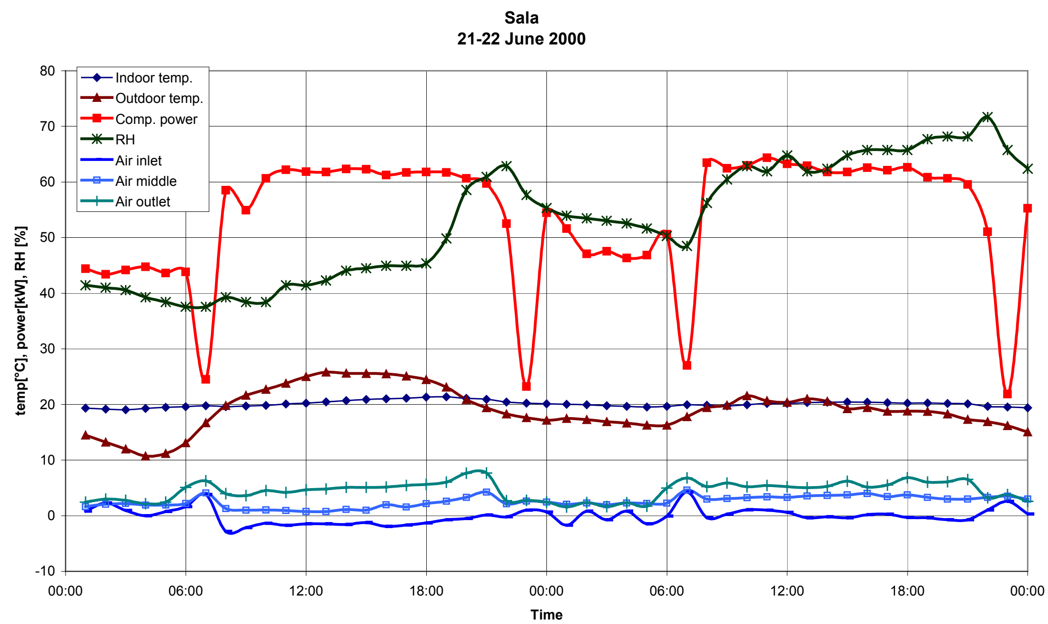

Figure 4.14: Temperatures, moisture and compressor power during two days in Sala.

Both high outdoor temperature and high indoor relative humidity increase the compressor power and the refrigeration load. The influence of high compressor power in air temperatures in the cabinet can be seen in figure 4.14 when temperatures, moisture and compressor power are presented during two days in June. The compressor operates at maximum capacity and cannot maintain the required air inlet temperature in the display cases (below –2°C). The temperature in the cabinet consequently increases to a temperature around 3°C.

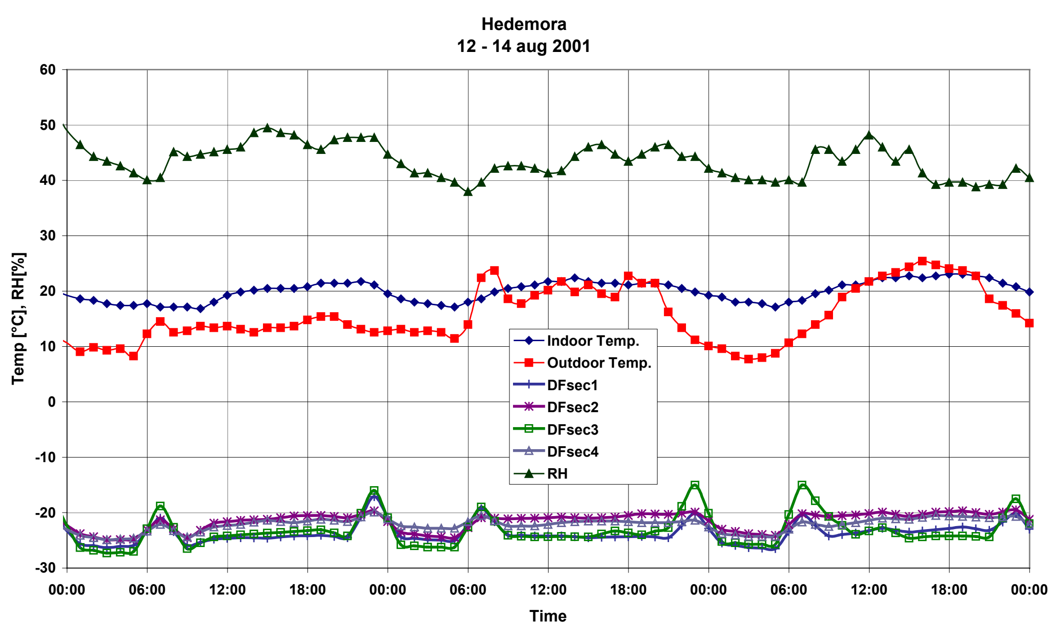

Figure 4.15: Temperatures in a deep freeze cabinet and indoor and outdoor temperatures during three days in August in Hedemora.

The influence of high indoor temperatures on compressor power has been studied in Hedemora. The high investment cost an the air conditioning system and a short period of higher outdoor temperatures during the summer have affected the decision of install AC in many supermarkets in Sweden. Figure 4.15 presents indoor temperatures, outdoor temperatures and air return temperatures in four different sections of a deep-freeze cabinet during three days in August 2001. The air temperatures in the sections are below –20°C with indoor temperature around 20°C.

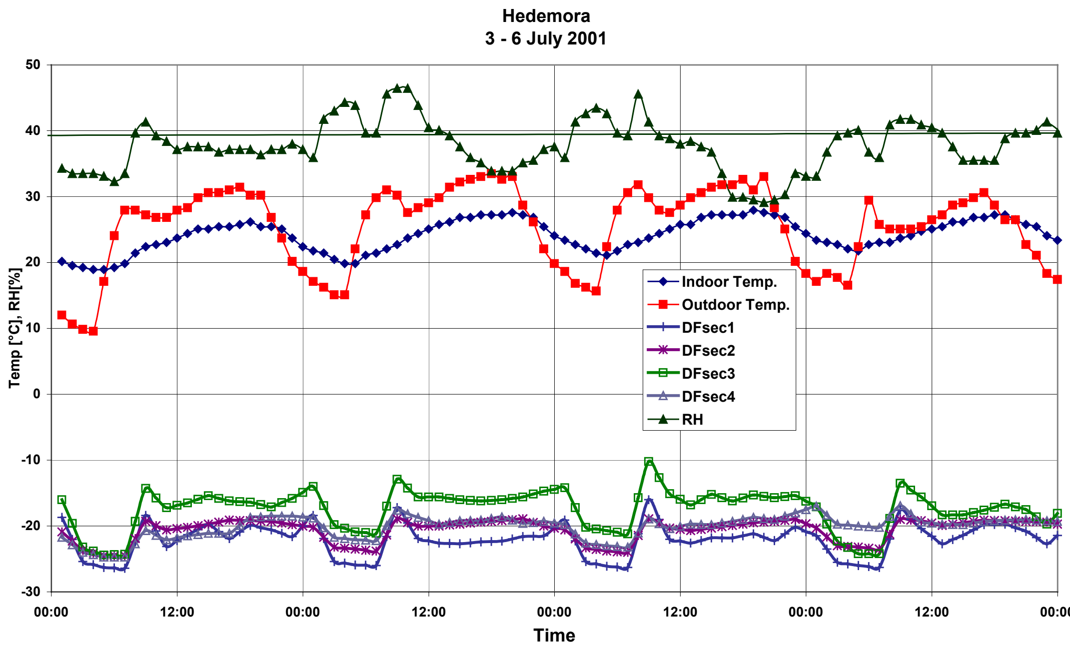

Figure 4.16: Temperatures in a deep freeze cabinet and indoor and outdoor temperatures during four days in July in Hedemora.

Figure 4.16 presents indoor temperatures, outdoor temperatures and air return temperatures in the same four sections but during four days in July 2001. The outdoor temperatures are above 30°C during a long part of the day and this affects the indoor temperatures. The values of indoor temperatures during the day are above 25°C, which is the dimensioning indoor temperature of the display cases. The maximum indoor temperature is 28°C, which occurs on the 5th of July at around 17:00 o'clock. The air return temperatures in three sections are above –20°C and one of them has air return temperatures higher than -15°C during the day. If the temperature of the products is assumed 1°C higher than the air return temperature, which is a reasonable supposition, then the temperature of the product should be about –14°C at an air return temperature of -15°C. According to the National Food Administration, which regulates and supervises the food area in Sweden, the maximum temperature of freeze products is –18°C. The date for the minimum durability, or use-by date, of frozen food is calculated at a temperature of –18°C. Higher product temperature implicates a risk of the apparition of bacteria in the frozen food that might cause illness to the consumers. The results from figures 4.14 and 4.15 confirm the necessity of air conditioning during the warmer days in the summer.

Another parameter to take into consideration is the humidity ratio. The variation of humidity ratio in Sala between July 1999 and July 2000, is shown in figure 4.17. The low values during the winter period occur in the four supermarkets and it can produce a reduction of weight in fruits, vegetables and other products. The magnitude of the weight loss depends on the time that the products are in the supermarket.

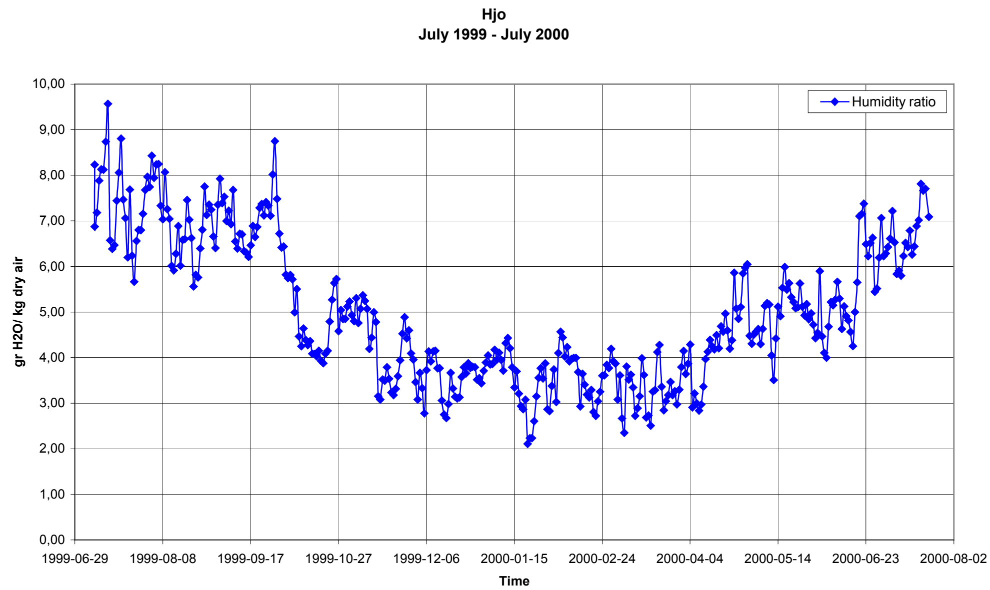

Figure 4.17: Humidity ratio in Hjo.

The humidity ratio difference between cool and warm zones in Farsta Centrum is shown in figure 4.18. The difference is minimal. The average of the humidity ratio during the winter are about 3,5 [gram H_2O/ kg. dry air]. During summer there are some days with values of humidity ratio over 9 [gram H_2O/ kg. dry air].

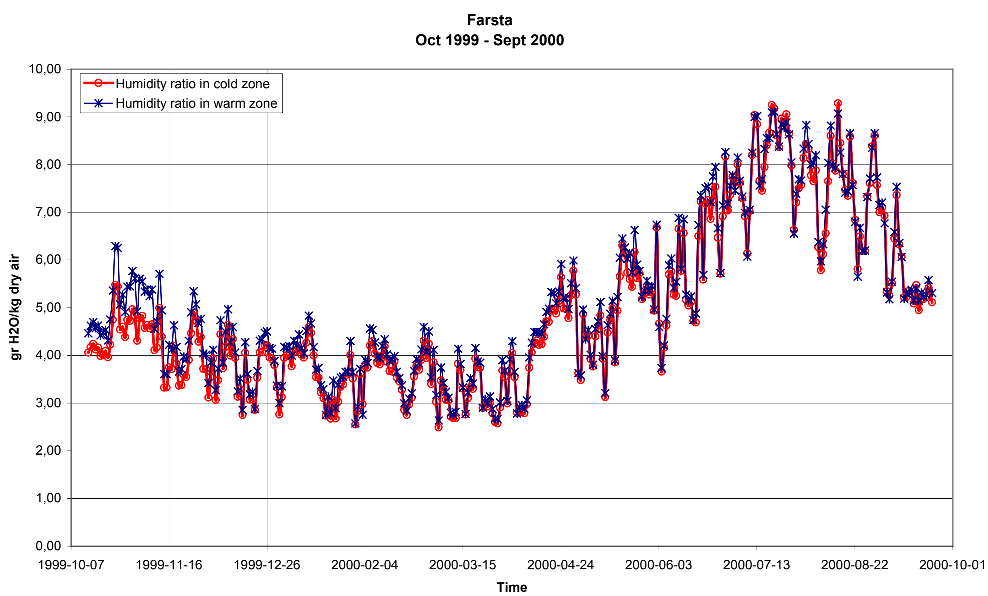

Figure 4.18: Humidity ratio in cold and warm zones in Farsta Centrum.

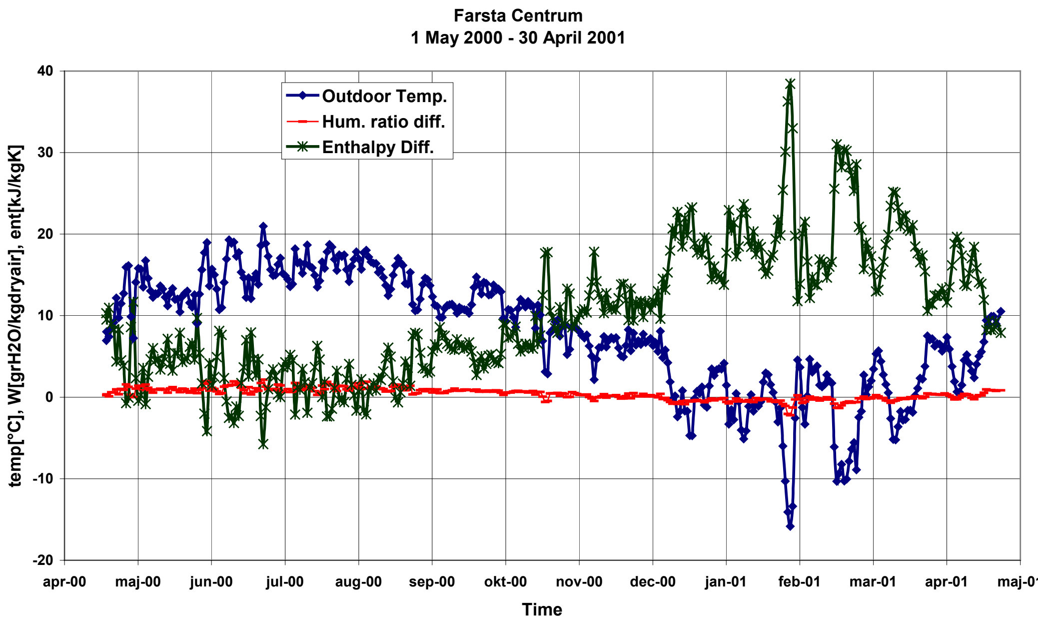

Figure 4.19: Outdoor temperatures, humidity ratio difference and enthalpy difference between indoor and outdoor climate in Farsta Centrum.

Outdoor temperatures, humidity ratio differences and enthalpy differences between indoor and outdoor climate in Farsta Centrum are shown in figure 4.19. The air enthalpy difference between indoor and outdoor climate should be calculated as a function only of the air temperature difference. The difference of humidity ratios is quite small, which means a low production of water vapour from machines, products and people in the supermarket. The indoor and outdoor humidity ratios are shown in figure 4.20.

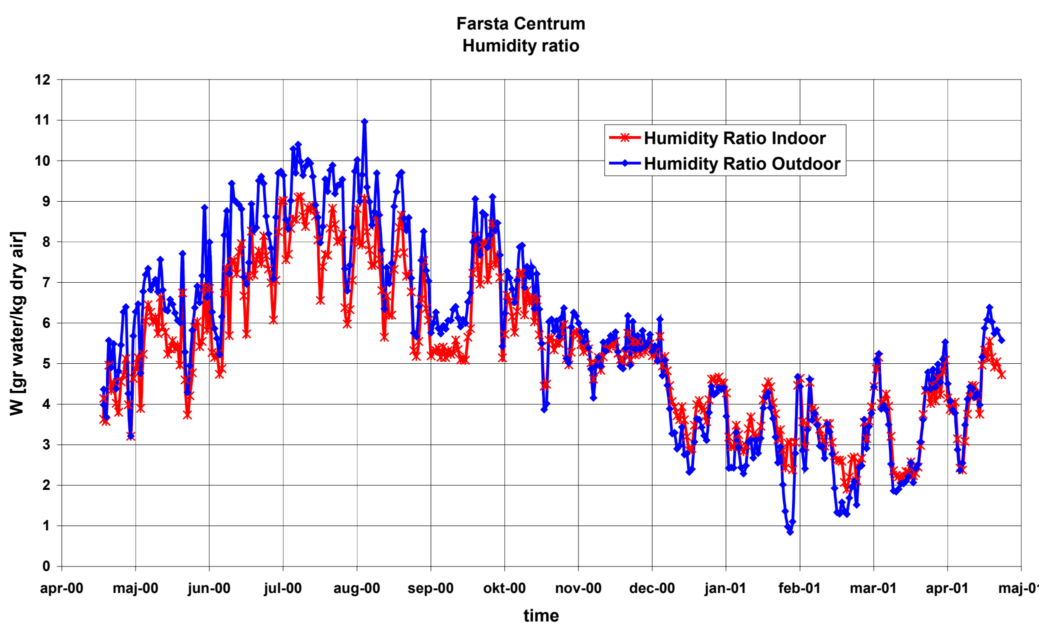

Figure 4.20: Indoor and outdoor humidity ratios in Farsta Centrum.

5. Heat recovery

Many supermarkets in Sweden utilize heat recovery from condensers as one way to increase the overall energy efficiency of the system. One interesting question is how much energy can be utilized from the condensers in a real system. There are many viewpoints here and practical experiences indicate that 40 – 70% of the necessary heat can be recovered. One reason for this is that the refrigeration systems do not operate continuously. Another issue here is that the design and operation of the refrigeration system and the HVAC systems not are done by the same people/organization and the communication between them is not always the best [4].

5.1. Heat recovery system design

The refrigeration system in supermarkets always rejects a large amount of heat from the condenser to the environment and a new efficient refrigeration system design takes advantage of the rejected heat by using it for air heating. The two biggest supermarket chains in Sweden use two different types of heat recovery systems. The first heat recovery system design, presented in figure 5.1, has an approach temperature of around 38°C to the dry cooler and a return temperature of around 32°C.

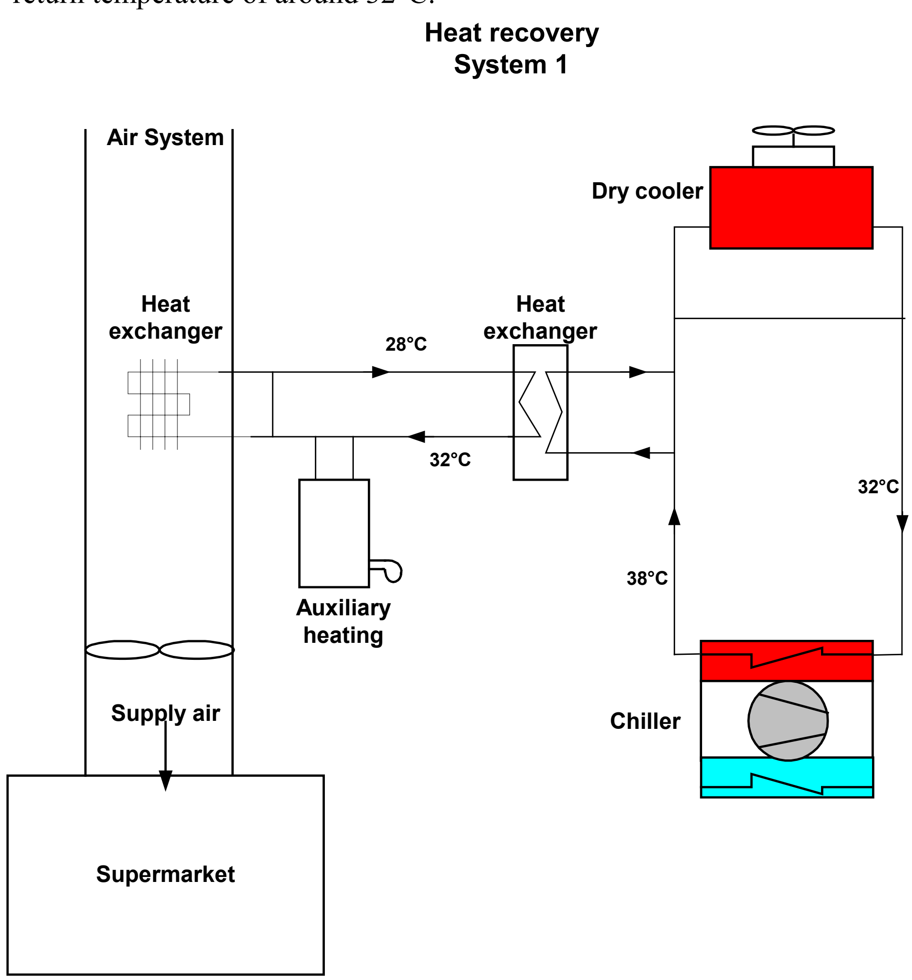

Figure 5.3: Heat recovery system design 2

The heat rejected from the refrigeration system is recycled to the supermarket by the air system during the winter. The circuit between the condenser and the dry cooler is connected to the air system by an extra heat exchanger. An auxiliary heater has been connected after the heat exchanger to insure the desired hot water temperature before the finned heat exchanger.

The reason for the extra heat exchanger is the possibility to use another fluid, as water, with better thermodynamics properties than the secondary coolant (glycol/water mixture). The system design has two disadvantages: the first is that the heat exchanger reduces the approach temperature to the coil in the air system and the second is when the air supply temperature is reached the control system by-passes the heat exchanger in the air system and, as often the case, the heat from the auxiliary heating is rejected to ambient in the dry cooler!

The second heat recovery system design presented in figure 5.2 avoids these problems. The heat from condensers is rejected directly to the air system via a heat exchanger. The approach temperature to the heat exchanger is around 38°C, and the return temperature is around 32°C. The auxiliary heating is connected to the air system after the heat exchanger that recovers the heat from the refrigeration system as shown in the figure. The auxiliary heating is supplied in a separate system.

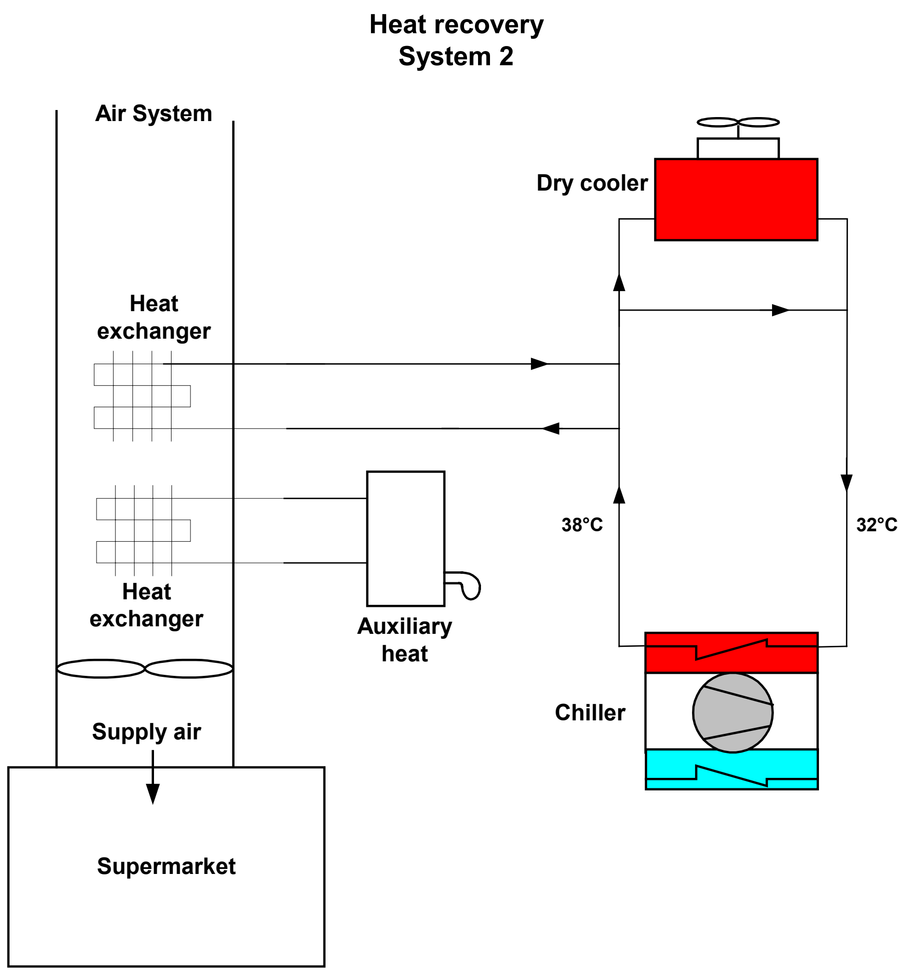

Figure 5.2: Heat recovery system design 2

5.2. Heat recovery in practice

Field measurements, in the supermarket of Sala, have been carried out during the month of Mars 2001 to see the variation of the compressor power and the influence of this on the condenser coolant temperatures. The temperatures and relative humidity have been measured momentary and stored every fives minutes. The compressor power is also stored every five minutes but in this case the value represents the mean power during the last five minutes.

Figures 5.1 and 5.2 show results from the measurement of outdoor temperature, indoor temperature, relative humidity, compressor power, dry cooler fluid approach temperature and return temperature during the third and the eleventh of March 2001.

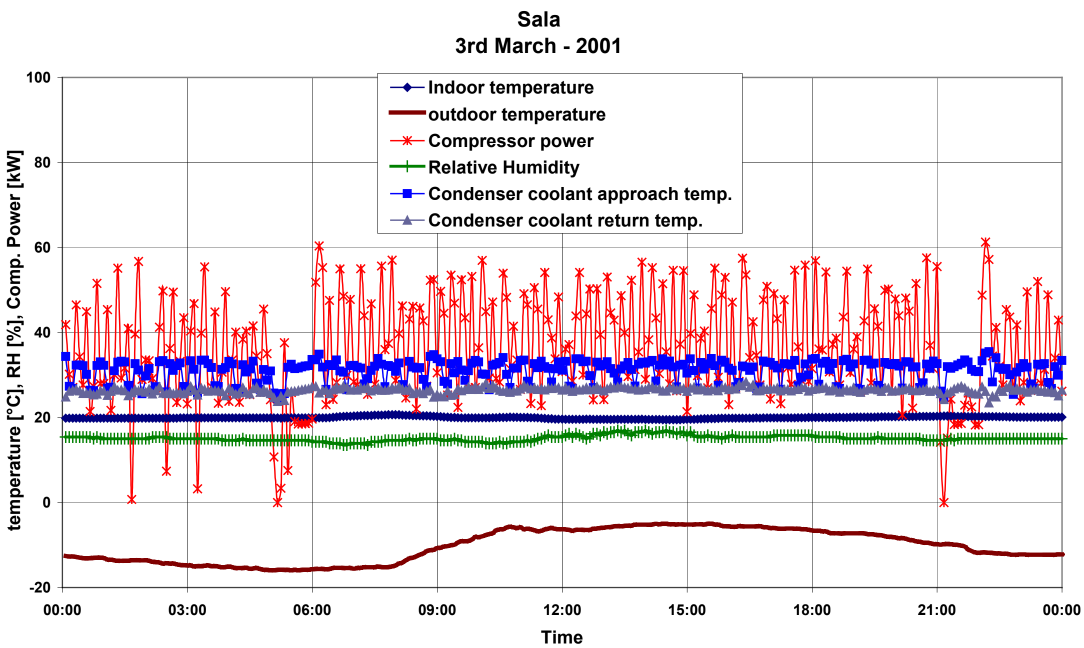

Figure 5.1: Detailed measurement from the supermarket in Sala the third of March 2001

During the third of March 2001 the average of the compressor power was 36,4 kW and during the eleventh of March 42,6 kW. Those values are equivalent to 64 % and 77 % of the nominal compressor power. The over dimension of the refrigeration system during cold days make the running time of the compressor very short, which influences the condenser coolant approach temperature. In figures 5.1 and 5.2 show the influences of the outdoor temperature and moisture in the compressor running time: The low average outdoor temperature, about –10°C, and relative humidity, about 18%, during the third of March caused a lower compressor running time and a larger variation of condenser coolant approach temperature than the eleventh of March when the average outdoor temperature was about 5°C, and the relative humidity was about 33%.

The percentage of time when the dry cooler fluid approach temperature was higher than 33°C has been calculated. The results show that on the 11th of March the percentage of time was 74% while on the 3rd of March the percentage of time was 48%.

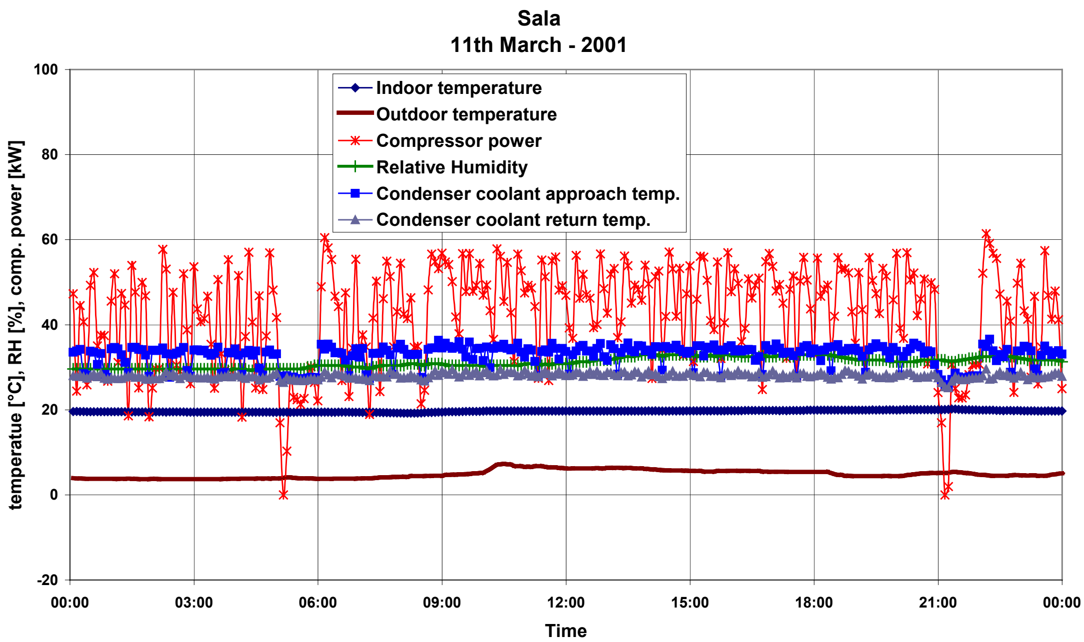

Figure 5.2: Detailed measurement from the supermarket in Sala on the eleventh of March 2001.

Measurements carried out in a supermarket in the city of Västerås by two students of the Mälardalen University show that one of the reasons for the low dry cooler fluid approach temperature is the existence of some mix points in the system that affect the dry cooler temperature and also the brine temperature on the evaporator side.

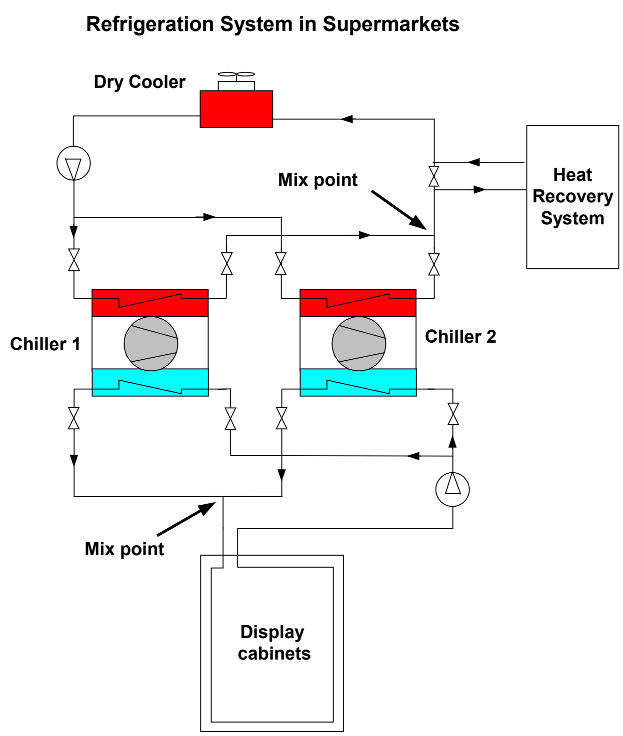

Figure 5.3: Refrigeration system in supermarkets

When one of the compressors in figure 5.3 is off, the dry cooler fluid from the heat recovery system circulates through both condensers. The temperature of the dry cooler fluid after one of the condensers increases in about 4°C while the temperature of the fluid after the other condenser has not changed. The fluids from both condensers will be mixed in the mix point with a decreasing of the total dry cooler fluid approach temperature. This phenomenon occurs also on the brine side with an increase of the brine approach temperature.

5.3. Heat recovery: Results from CyberMart

The utilization of heat recovery from condensers increases the overall energy efficiency of the system. The question is how much energy can be recovered. An energy balance in Sala has been realized to estimate heat requirements during the heating season, which have been compared with the consumption of district heating and available condenser heat from simulation in CyberMart. The results, presented in table 5.1, show that the available condenser heat is more than sufficient from an energy balance point of view to cover the heat requirement.

	Heat requirement	District heating	Condenser heat	Utilized	Percent
1999 - 2000	kWh	kWh	kWh	kWh	%
Oct	16750	4691	95915	12059	13
Nov	28333	7402	85223	20931	25
Dec	36707	8516	88527	28191	32
Jan	44403	7572	88210	36831	42
Feb	40099	10176	80637	29923	37
Mar	35773	7807	88008	27966	32
April	20404	5529	89820	14875	17

Table 5.1: Estimated need for heating, district heating (by meter), available and utilized condenser heat for a supermarket in the city of Sala.

An obvious drawback with the heat recovery system is that the condensation temperature must be kept at a level where heat can be transferred to the heating system of the supermarket. This is normally an air coil in the supply air ventilation system. Another solution is floor-heating coils. A common experience among practitioners indicates that the typical required temperature level for the condenser coolant is 38°C. This leads to increased energy consumption for the compressors. Several different control strategies may be applied and results from four different cases simulated in CyberMart are reported here.

The four cases considered here are:

- Case 1, Floating condensation year round + aux heating – no heat recovery
- Case 2, Floating condensation and fixed “bottom” temperature level of condenser coolant set to 16°C + aux heating – no heat recovery
- Case 3, Condenser coolant set to 38°C all year round – heat recovery
- Case 4, Condenser coolant set to 38°C during the heating season and floating condensation during the warm season - heat recovery

The choice of Case 2 emanates from the fact that it is difficult to fully utilize floating condensation at low outdoor temperatures due to hardware and control problems (the expansion devise for example). Case 1 and 2 does not utilize heat recovery at all.

The energy for the refrigeration systems is calculated using CyberMart software for all four cases. The results are shown in figure 5.4 and table 5.2

aug 1999 - jul 2000	No heat recovery	Heat recovery	Heat recovery floating during summer
District heating[kWh]	222468	51693	51693
District heating price[kr/kWh]	0,5	0,5	0,5
District heating cost [kr]	111234	25847	25847
Compressor power [kWh]	304162	425589	377753
Electricity price [kr/kWh]	0,5	0,5	0,5
Electricity cost [kr]	152081	212795	188877
Total cost [kr]	263315	238641	214723

Table 5.2: Energy and total cost for Cases 2, 3 and 4.

The results in table 5.2 show that Case 4 with condenser coolant set to 38°C during the heating season and floating condensation during the warm season is the most effective of the four cases analyzed.

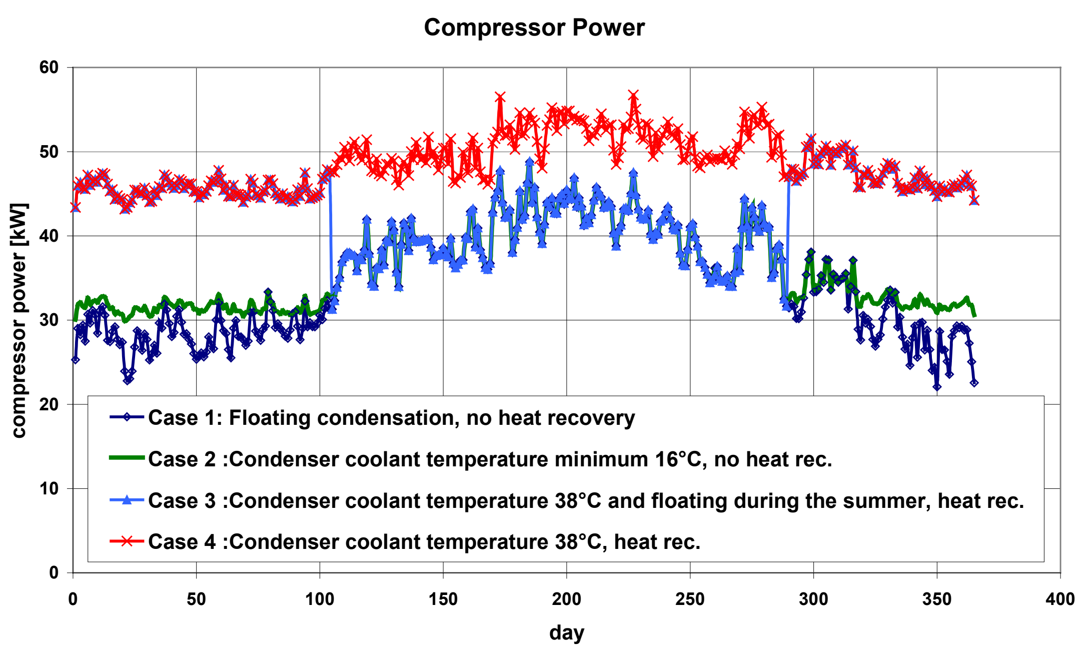

Figure 5.4 Results from comparison of Cases 1-4.

6. Conclusions

Simulations show that the completely indirect system with sub-cooling is more energy effective than the cascade system. The results and the experiences of this system are more positive than the cascade system

The influence of the outdoor temperature on the indoor relative humidity of air is an important factor to take in consideration when dimensioning refrigeration and heat recovery systems in supermarkets. Lower outdoor temperature and moisture affect the compressor power and the dry cooler fluid approach temperature

.
Night covering of display cases and deep-freeze display cases reduce the energy consumption in supermarkets with about 10% to 20%. Night covering is an efficient method to reduce infiltration and radiation loss in cabinets.

The economy and overall energy efficiency of supermarkets, in cold climates, benefit from heat recovery. In theory, the necessary heat can always be supplied from the condensers. Practical experiences show that in real systems only 40 % – 70 % of the necessary heat is recovered. There are many reasons for this but the most important are on/off regulation, low cooling load during cold days, poorly designed heat recovery systems and non-communication control systems for refrigeration and HVAC.

Indoor temperatures above 25°C increase the temperature of products in display cases and deep-freeze cabinets to levels that affect the date for the minimum durability of the foods. An air conditioning system is necessary to avoid indoor temperatures higher than 25°C even if the warm season is short.

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FABER MAUNSELL

IEA Annex 26:

Advanced Supermarket Refrigeration and HVAC Systems

Final UK report on Phase 2 Studies

Prepared by:	and	....
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Contents

EXECUTIVE SUMMARY

Executive Summary

This report is the final report on the Phase 2 UK contribution to the Annex objectives. Phase 1 was the production of a 'Technology Review' and Phase 2 consisted of detailed field tests, analysis and modelling of a number of key Phase 1 subjects. The main body of this report deals with the detailed results of the Phase 2 studies and the overall analysis of the current position regarding the available technologies. A summary of the findings of the Phase 1 studies is included.

The second phase of the UK contribution covered four research areas:

- secondary refrigerant systems
- alternative defrost approaches
- interactions between the store environment and the cabinet case
- combined heat and power and combined heating, cooling and power applications in supermarkets.

This executive summary records the main points that were concluded from each of the four studies carried out. Each area of research was carried out by an organisation specialising in refrigeration technology. The areas where further research may be useful have also been identified.

Secondary Systems

Secondary refrigerant systems offer the potential of reduced total equivalent warming impact (TEWI) as a consequence of reduced refrigerant leakage of high ozone depleting potential (ODP) refrigerants. However, it is recognised that there may be additional energy requirements to operate this type of system because of greater temperature gradients and increased pumping power.

Field monitoring data from a typical supermarket was used to validate a refrigeration simulation model, and a detailed investigation of the thermal and fluid properties of secondary refrigerant fluids was made. This enabled simulations to be carried out to assess the feasibility of a wide range of secondary systems and refrigerants in a typical supermarket. A key finding of the simulation was that the energy use of the secondary systems simulated would typically be greater than for other systems and as a consequence they would deliver higher operational CO_2 emissions. In terms of overall TEWI, the secondary system using R290 (propane) would reduce the TEWI by 8% over a 15 year life compared to an R22 direct system with 15% refrigerant leakage. The benefit of a reduced TEWI of a secondary system is lessened as the leakage rate of the equivalent direct system decreases.

The project has shown that there is sufficient knowledge of secondary fluid properties to be able to design an effective system. However, careful optimisation will be needed for a secondary system to become more attractive both financially and in terms of TEWI, so further research is recommended into the optimisation of secondary systems.

Secondary systems - further research

Secondary systems using pumpable ice have been tried in the UK but further work is needed to establish if it is a viable option. Research is also needed to address the issues of store layout and pipework arrangements with the specific aim of reducing the capital cost of the secondary system.

The use of CO_2 as the secondary refrigerant needs to be researched in the UK context. The current situation is not clear with respect to the health and safety issues and the performance of the systems in operation. Cascade systems have been used in the USA and may have some energy benefits and the possibility of using these in the UK should be investigated.

If changes in legislation force the adoption of lower Global Warming Potential (GWP) refrigerants then retailers may wish to investigate secondary systems further. The options for improvement include: system design (store layout to reduce pipework capital costs); pumpable ice; CO_2; and cascade systems. Reference should be made to experience in North America and Scandinavia where some of these approaches have been adopted.

Defrost

The energy used for defrosting evaporator coils in supermarkets can amount to 30% of the operating energy. Alternative means of reducing the defrost energy have been researched by laboratory studies of different defrosting techniques and control strategies. Studies were carried out in both high and low temperature applications and the following defrost techniques were considered:

- off-cycle
- hot/cool gas
- electric.

However, not all techniques can be used in all circumstances, for example, off-cycle is not appropriate in low temperature applications, leaving a choice of gas or electric defrost.

The findings showed that, for high temperature applications, the cheapest and least energy consuming means of defrost is ‘off-cycle’. This method of defrost is widely used in the industry. For low temperature applications, the use of gas defrosting is more energy efficient than electric defrost but the considerable extra capital costs lead to long payback periods, up to 28 years. Consequently, electric defrost is most commonly used for low temperature applications in the UK.

The length and frequency of the defrost cycle has a significant effect on the energy use of a refrigerated cabinet. The normal procedure of a fixed number of cycles per day is wasteful of energy because the time period between defrosts may be either too long - thereby reducing the frosted evaporator efficiency before defrost - or too short which provides unnecessary defrost events. In addition, the length of defrost can be excessive and this wastes further energy, and risks adverse effects on the product quality and life. Optimised defrost cycles could reduce energy use by between 25 and 50%, where electric defrost is employed. Work is under way to develop controls to achieve this.

The optimum defrosting cycle will depend on the cabinet type and store conditions and a means of control that relates to these parameters would lead to significant energy savings.

Defrost - further research – feedback from UK partners

A number of issues for further research were revealed by these studies. Liquid defrosting for low temperature applications should be further researched as it is potentially a more effective and energy efficient technique.

Methods of determining the need for defrosting should be investigated and these should take into account the avoidance of simultaneous defrosting of a number of units leading to high electrical loads. Evaporator design has an influence on the formation of ice and should also be researched in order to avoid or reduce the incidence of frosting.

CHP/CCHP

Combined heat and power and combined cooling, heat and power are potentially efficient methods of providing the energy for supermarkets. A simulation model has been developed that allows the energy and heat flows in a supermarket to be modelled and will predict energy and cost savings arising from a range of measures taken to improve efficiency of supply or use of energy. The model was tested against a store using CHP and then used to evaluate a range of options.

The studies showed that currently available equipment can be used satisfactorily but needs to be carefully optimised for the specific store in which it is being installed. The cost effectiveness of CHP/CCHP is sensitive to the fuel costs and in particular the ratio of the gas and electricity costs. Additionally the number of hours for which the plant operates in a year is a key determinant of the cost effectiveness.

CHP/CCHP - further research
Determining the heat and power load profiles of a wider range of stores would assist in developing a clearer view of the potential for CHP in supermarkets.

The design of the refrigeration system with the use of CHP and CCHP needs to be investigated as novel approaches may provide a better match of heat and power usage. For example, gas engine prime movers for vapour compression machines can be attractive and the potential for their use should be examined.

The simulation model that has been developed in the context of this research could be made available for retailers and other interested parties to carry out feasibility studies on future stores.

Case and Store Environment Interaction
This project was based on the premise that the internal environment of the store, in which the refrigerated display cabinet is located, can have a significant bearing on the energy use and performance. The temperature and humidity of the air and its movement around the cabinet are all likely to influence the energy use. A store environment was measured and the conditions were recreated in the laboratory. Tests were then carried out with these different internal environmental parameters to investigate their impact on energy use.

The results showed that for the freezer tested varying either of the store temperature, relative humidity and ventilation mode and rate had little impact on the energy consumption of the unit. It may be surmised that although the rate of frost formation on the evaporator was dependent on the ambient relative humidity, whilst the defrost initiation and duration were preset, no significant impact on the energy use for defrost was observed.

For the chilled cabinet the energy consumption was more dependent on the store conditions. A change of store temperature from 19 to 22^{o}C increased the energy use by up to 20%. At the higher temperature of 22^{o}C when the relative humidity of the store environment was increased from 35%rh to 50% rh the energy consumption raised by 15%. Increased ventilation around the chiller could also lead to some increase in energy use.

Overall, relative humidity is not as much of a significant factor in energy use as was initially expected from the literature review. This is probably because the rh values in the UK are typically much lower than in the USA, where the other research had been carried out.

An additional finding was that reduction of the voltage supply to the cabinet had a large effect on the electricity use. Reducing the voltage from 240 to 220 volts resulted in a saving in energy of approximately 10% without any loss in performance. This effect has also been observed in stores, and current practice is to reduce voltage supply to the whole store in order to make savings.

Case and Store Environment Interaction - further research
The use of high velocity air supply in aisles in order to mix the stagnant cool air needs to be investigated particularly to ascertain the impact on customer comfort. This needs to be studied in the context of customers' expectations of comfort levels in stores and particularly adjacent to refrigerated display areas.

Research is required into controlling anti-sweat heaters by methods other than time control. The control of the anti-sweat heaters is an area in which there is a potentially large energy saving to be made.

The design and manufacture of cabinets is a major issue that warrants further examination. The tendency is to provide standard solutions that are not being developed in line with the aims of improving energy efficiency. To overcome this needs the active participation of manufacturers in the realistic testing and measurement of in-store operation in future trials.

Overall Conclusions

These projects have confirmed that many of the practices the retailers currently have in place are the most cost effective e.g. the use of DX systems at low leakage rates, and electric and off-cycle defrosting. However, some of the more expensive and/or innovative technologies could further reduce the TEWI of supermarket refrigeration.

There also appears to be great scope for considering the whole system – from power supply to refrigerated goods display in a store environment – as a single system to be optimised; rather than a collection of individual components each acting in isolation.

1 INTRODUCTION

1 Introduction

The objective of Annex 26 of the IEA Heat Pumps Implementing Agreement was to demonstrate and document the benefits of advanced supermarket refrigeration and HVAC systems. One of the specific goals was to provide information that would result in a reduction the total equivalent warming impact (TEWI) of supermarket refrigeration systems.

The impact on the TEWI of refrigeration in supermarkets involves a complex set of interactions between the plant installed and the environment into which it has been placed. This Annex takes advantage of research and development across all of the participating countries to accelerate adoption of technologies that will decrease the environmental impact of supermarkets. The Annex was comprised of five participating countries:

- United States of America – Operating Agent
- Sweden
- Denmark
- Canada
- United Kingdom

The Annex was divided into two phases that are best characterised as:

Phase 1, a state-of-the-art review considering all developing technologies
Phase 2, field trials and tests of candidate systems and components.

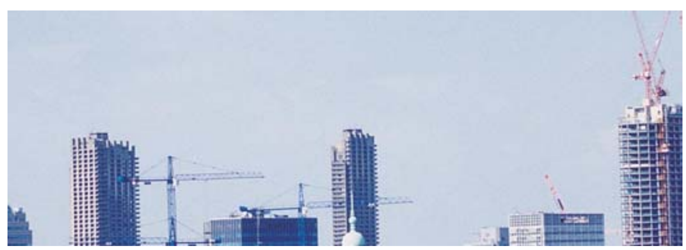
This report is the final report on the UK contribution to Phase 2 of the Annex objectives. The UK contribution has been split into two Phases as in the overall management of the Annex. Phase 1 was the production of a 'Technology Review' and Phase 2 consists of detailed field tests, analysis and modelling of a number of key Phase 1 subjects. The main body of this report deals with the detailed results of the Phase 2 studies and the overall analysis of the current position regarding the available technologies. A summary of the findings of the Phase 1 report is included in this report.

The UK participation in the Annex has been funded by the following organisations:

- ACDP (Integrated Building Services) Ltd
- Carbon Trust (AEA Technology & Environment)
- ASDA
- Calor Gas
- FaberMaunsell
- Honeywell
- Marks & Spencer
- Safeway
- Sainsbury's Supermarkets Ltd
- Somerfield
- Weatherite Ltd

FaberMaunsell represented UK within the IEA Annex and managed the UK input to the Task.

2 OBJECTIVES OF IEA ANNEX 26

2 Objectives of IEA Annex 26

The impact on the TEWI of refrigeration in supermarkets involves a complex set of interactions between the plant installed and the environment into which it has been placed.

To quote from the definition of the Annex 26 of the IEA Heat Pump Implementing Agreement:

> *"The objective of this Task is to demonstrate and document the benefits of advanced systems design for food refrigeration and space heating and cooling for retail foodstores (supermarkets). A specific goal is to identify advanced systems design options to reduce total equivalent warming impact (TEWI) of supermarkets."*

The system concepts that were envisaged as important at the outset of the Annex were:

- Secondary loop systems – a central chiller and secondary coolant loop to the display cases
- Distributed compressor systems – small compressor racks in close proximity to the display cases
- Self-contained display cases – individual compressor and condensing unit for each display case.

Additionally, it was anticipated that integrating the refrigeration system with the space conditioning systems would be investigated, as would water-source heat pumps for condenser heat reclaim. It was also recognised that it may be possible to use a desiccant system for humidity control of the store and regenerate this with rejected heat.

In meeting these aims the Annex investigated candidate advanced system design approaches to determine the potential to reduce refrigerant usage and energy consumption for both refrigeration and heating/air-conditioning in supermarkets. To achieve this the work of the Annex was divided into two main task areas.

2.1 TASK 1

a) Analyse candidate advanced supermarket refrigeration systems and compare them to state-of the-art systems (each participating country to specify the systems for which they would contribute). The systems would include:

Secondary loop systems

- Brine as secondary coolant with condenser heat rejection to air
- Brine as secondary coolant with condenser heat rejection to water loop
- Ice slurry as secondary coolant with condenser heat rejection to air
- Ice slurry as secondary coolant with condenser heat rejection to water loop
- CO_2 as secondary coolant with condenser heat rejection to air
- CO_2 as secondary coolant with condenser heat rejection to water loop.

Distributed compressor systems

- Distributed compressor racks with condenser heat rejection to air
- Distributed compressor racks with condenser heat rejection to water loop.

Self-contained display cases

- Self-contained display case with condenser heat rejection to indoor air
- Self-contained display case with condenser heat rejection to outdoor air
- Self-contained display case with condenser water loop heat rejection

Capacity control options for self-contained cases including:
- Compressor cycling
- Modulating compressor (variable speed, cylinder unloading, etc.)

b) Determine 'best' or 'most appropriate' method(s) to integrate HVAC systems with refrigeration to maximise waste heat recovery.

2.2 TASK 2

c) Conduct one or more field demonstrations of advanced systems in actual store (in collaboration with system manufacturers, participating stores, and local utilities) to compare with conventional store systems currently in common use, or alternatively,

d) Prepare case studies documenting ongoing or previous demonstrations being conducted by various agencies within each country.

Task 1 analysis work would be directed toward comparison of the advanced system options with a baseline system in terms of first cost, energy consumption for all system components - compressors, heat rejection subsystem (fans and pumps), secondary coolant pumps, display cases (fans, lights, etc.) – including defrost energy and impact of control strategies, possible heat load to the shopping area from the refrigeration system (increases both the space cooling and refrigeration demand), TEWI, maintenance requirements/costs as well as product quality (i.e. to minimise product weight loss and temperature fluctuations in the products).

HVAC options would be evaluated to identify which option(s) can provide maximum utilisation of refrigeration reject heat with respect to reducing HVAC energy usage and TEWI, and providing improved indoor temperature and humidity control. Cost and maintenance issues would be evaluated for each HVAC option as well. A variety of alternative refrigerants (fluorocarbons and natural refrigerants) and secondary coolants would be considered in the analysis. Selection of a field test site(s) or identification of ongoing field tests/demonstrations for case study documentation would also be considered.

The preferred approach for Task 2 would be to conduct field tests of one or more advanced systems. Store(s) and system(s) would be instrumented to allow full characterisation of their performance in terms of operating parameters such as indoor and outdoor ambient conditions, control set points, etc. At a minimum the instrumentation would measure sufficient parameters to fully document the refrigeration and HVAC loads and energy usage and the store and display cabinet operating conditions. For comparison purposes a baseline direct expansion system would be monitored.

The aim was to obtain as much data from equivalent size stores in each of the participating countries under similar conditions of loads and climate. One of the studies was to include some form of heat recovery from the refrigeration system into the store HVAC.

The Annex was scheduled to last for a period of 30 months and would provide comparative data of field tests or case studies to document the performance of advanced and cost efficient supermarket refrigeration/heat recovery options and the potential energy and TEWI reductions.

2.3 THE UK TASKS

The UK research has met these objectives through the choice of projects, and the approach taken. This included inviting the retail partners to contribute store examples and provide information on typical store conditions e.g. refrigerant leakage rates; using in-store measurements to create representative laboratory conditions where in-store testing would be inappropriate; and through using actual store performance to validate and develop mathematic models. These models were then used to investigate alternative methods of reducing TEWI.

2.4 OVERALL SCHEDULE

Work on the Annex 26 began on 1January 1999 and it will be completed by 31 March 2003. The UK research was completed in August 2002.

3 SUMMARY OF PHASE 1 - THE TECHNOLOGY REVIEW

3 Summary of Phase 1 - The Technology Review

The Phase 1 work undertaken by the UK partners was a 'Technology Review', summarised below. The review addressed the following aspects that characterise the overall TEWI of the refrigeration plant:

- Refrigeration system type and configuration
- Individual system components
- Interaction of the systems and components with the supermarket environment.

Within each of these major categories a number of sub-categories have been studied. The table below shows the specific topics and which organisation carried out the work.

Topic	UK Partner
Centralised, Distributed and Integral Systems	EA Technology
Refrigerants	EA Technology & Calor Gas
Compressors	Brunel University
CHP / CCHP	South Bank University
Gas Engine Driven RAC Equipment	Calor Gas
Air Cycle Systems	FRPERC, University of Bristol
Refrigeration Controls	Honeywell & EA Technology
Evaporator Coils and Frosting/Defrosting Issues	Brunel University
Optimisation of Secondary Systems	EA Technology
Case/ Store Interaction	Food Refrigeration and Process Engineering Research Centre, University of Bristol
Heat Recovery for Supermarkets	Weatherite

3.1 SYSTEM TYPES

The supermarket refrigeration system cools a product that is on display and then rejects the energy extracted to another medium. This can be achieved in a number of ways and from a range of technology. Issues involved in the design and configuration of these systems are:

- Degree of centralisation or distribution of the compressors
- The use of secondary loops
- CHP and CCHP
- Gas engines
- Air cycle machines.

3.1.1 Centralisation/distribution of the compressors

One area where significant development has taken place in recent years is the move from a central refrigeration plant to more distributed systems. There are three main configurations for compressors in a supermarket:

a) A large compressor at a central point with refrigerant pipework to each of the display cases – central
b) Several smaller compressors at a central point with refrigerant pipework to each of the display cases – distributed
c) A compressor as part of the case itself – integral.

Central systems using the HFC refrigerants are now rare, as the industry has moved away from large compressors to multiple smaller compressors – the distributed system. The distributed system allows better load control of the system and in the event of equipment failure it has the advantage that the entire system is not affected.

The integral or mobile system has the compressor as part of the display case. The heat is rejected to air or to a water loop. These are used in smaller cases such as chest freezers and ice cream displays. These units need to be carefully integrated with the HVAC of the store to avoid increasing its cooling load when using locally rejected heat. The use of secondary systems opens up the potential for greater flexibility in design for distributed systems.

Both the distributed and integral systems provide energy and cost benefits over the central plant systems. However, with the more recent moves to consider alternatives to HFC refrigerants, there may be a move back to centralised systems due to health and safety considerations related to the properties of alternative non-HFC refrigerants. For example, an ammonia plant needs to be located away from occupied areas is generally in open air or its own plant room.

3.1.2 The use of secondary loops

A secondary refrigeration system uses a primary refrigerant to cool a secondary refrigerant – the secondary refrigerant is then circulated around the display cases to cool the product. Concerns over the effects of refrigerants on ozone depletion have led to the need to reduce refrigerant leakage and to consider the use hydrocarbon and ammonia as refrigerants. Using a secondary refrigerant circuit has two significant advantages in achieving this:

- In a secondary system the charge of the primary refrigerant is much reduced. This allows the use of hydrocarbon or ammonia as a refrigerant – both of which cannot be circulated around the display cases due to toxicity/flammability. Ammonia is both toxic and explosive within certain limits.

- It reduces the amount of HFC/HCFC refrigerant per system if this is chosen as the primary refrigerant.

There are numerous secondary refrigerants in use including: propylene, glycol, Tyfoxit and CO_2. The liquid based refrigerants transport sensible heat whereas the phase change systems (such as CO_2 and binary ice) move latent heat. There are examples of all these types of secondary systems in both the UK and the partners' countries.

The performance of the secondary systems depends on the operating conditions and is difficult to compare with direct expansion (DX) systems. Temperature lifts across heat exchangers are significant and should be kept as small as possible. The pumping energy of low temperature (LT) brine solutions is high and phase change materials become more attractive. Phase change secondary techniques can overcome some of these barriers, but if CO_2 is used, a buffer storage system is required to prevent excess pressure from developing within the system.

The capital cost of a secondary system is some 10% greater than for a DX system, but this is seen to reduce as pre-insulated plastic pipe-work becomes available. There is a move in Scandinavia to install mainly secondary systems because of environmental legislation and high taxes on HFC/HCFC refrigerants.

3.1.3 Combined Heat and Power / Combined Cooling Heat and Power

Combined Heat and Power (CHP) and Combined Cooling Heat and Power (CCHP) are both technologies that have a role in the provision of energy for supermarkets. The exemption of 'Good Quality' CHP schemes from the Climate Change Levy gives a considerable boost to the potential cost benefits of using this technology. Limited experience in the UK shows that CHP in supermarkets can provide reasonable payback periods.

Considering the summertime refrigeration load, studies show that the heat to power characteristics of supermarkets should be suitable for either CHP or CCHP. This is especially true if the refrigeration can be provided by absorption plant running to provide cooling to the display cases from the heat output of the CHP plant.

3.1.4 Gas engine driven equipment

In addition to CHP as a system energy provider, the technology exists to drive compressors directly by using a gas engine.

This technology is well proven but is generally restricted to mobile applications such as transport refrigeration and vehicle air conditioning. It is used in other parts of the world to reduce peak electricity demand, and, for example, 5-10% of residential and commercial split air conditioning systems in Japan use gas engines.

The capital costs of the engine will be greater than the equivalent electric motor, but running costs are more advantageous when the price of electricity is more than 2.5 to 3 times the cost of gas (per kJ). The technology has not been applied to supermarket refrigeration but if the cost benefits can be proved the technology can be adopted for central systems, either with or without secondary systems.

3.1.5 Air cycle machines

A further system that could be used in supermarket refrigeration is the air cycle. The air cycle uses air as the working fluid and as such has the least environmental impact, either locally or globally, of any of the refrigerants. Compressed and cooled air is expanded through a turbine and the cool low-pressure air removes the heat from the refrigerated space. The cycle can be either open or closed and given that air has no capital cost the open system is very attractive. Equally the construction of the plant is easier and less costly. An example of a closed system in a commercial application is the cooling of railway carriages, and the open cycle is used in aircraft cabin air-conditioning systems.

In terms of supermarket refrigeration, the most likely choice would be an open system, where air can be expanded directly into the cabinet. This has an additional benefit in that heat exchangers can be eliminated. Such a system has been demonstrated in the laboratory, and it consists of the cases being fed with compressed air that is expanded and discharged within the case to cool the product. The system produced air temperatures between $-0.5^{o}C$ and $4.6^{o}C$ for a chilled cabinet and $-27.4^{o}C$ to $-37.6^{o}C$ for a frozen well cabinet.

Scaling up of the concept from a single case to multiple cases will cause problems due to the volume of compressed air required, but with multiple compressor locations the dimensions of the ring main will be reduced, and the capital cost reduced.

3.2 SYSTEM COMPONENTS

The following system components have been examined:

- Refrigerant
- Variable speed compressors
- Evaporator coil defrost – various alternative methods
- Controls

3.2.1 Refrigerant

From the environmental perspective, the major system component is the refrigerant used. Over the last ten years there have been major changes in the legislation concerning refrigerant usage and leakage due to the ozone depleting and green house warming potential of refrigerant gases. Consequently, there has been a large shift in the choice of refrigerant for supermarkets. A typical refrigerant currently in use is HFC404a - which has zero ozone depleting potential but a considerable global warming impact - but this is now being challenged by more ‘natural’ refrigerants such as ammonia and hydrocarbons.

Hydrocarbons have been compared with CFCs and HFC based refrigerants and performance enhancements in excess of 25% have been recorded (HC1270 in place of HFC404a in a high temperature (HT) scroll compressor). However, ammonia and hydrocarbon refrigerants may need to be used in secondary circuits, with a consequent energy efficiency penalty, which will reduce the overall benefit. Predicting the overall benefit by means of the TEWI is consequently more complicated because of the combined effects of ozone depletion, system efficiency, and global warming potential.

There are examples of both ammonia and hydrocarbons being used in supermarkets in the UK, and in partner countries. On occasion these systems are used in conjunction with a secondary circuit and local compressors for the low temperature (LT) applications depending on the fluid used for the secondary loop.

The various types of refrigerant available and a review of current refrigerant legislation in the UK are included in the full Phase 1 Report – see Volume 2.

3.2.2 Variable speed compressors

Variable speed capacity control in refrigeration compressors, for small to medium capacity refrigeration applications, has been investigated over the last 30 years. This technology can now be found on a wide variety of small capacity packaged air conditioning and heating systems from Japanese manufacturers but has not yet gained wide acceptance in the commercial refrigeration sector.

In variable speed refrigeration, the refrigeration capacity at part load conditions is matched to the load by regulating the speed of the compressor motor. The speed can be changed in two ways:

- discreet steps (stepwise)
- varied continuously to give an infinitely variable speed control.

Either of these approaches can be achieved by two means:

- indirect coupling of the load to the motor (a constant speed motor and a speed control device between the motor and the load)
- direct coupling of the load to the motor (using a variable speed motor).

The most common approach is to provide speed control to a motor directly coupled to the compressor. In this case stepwise speed control can be achieved by using multi-pole electric motors. The required compressor capacity is obtained by switching a finite number of poles to achieve the desired speed. Infinitely variable capacity control can be achieved by using electronic variable speed drives to regulate the speed of the compressor motor. The speed variation is determined by the frequency output of the variable speed drive inverter.

Stepwise control is less costly than continuous speed control but step controlled motors have lower efficiency than constant speed motors and the fixed number of speeds restricts compressor capacity control compared to continuous stepless capacity control. In all cases there is a minimum speed of operation approximately 25-30% of maximum speed because of lubrication limitations in the compressors.

A study was carried out at Brunel University, on a 25kW semi-hermetic reciprocating compressor, operated over a 2:1 speed range and with head pressures between 11 bar and 18 bar. The results showed that the COP rises significantly as the speed decreases and consequently there are significant energy benefits. Savings of up to 24% can be achieved, but there is still a need to consider the other implications of VSDs, such as harmonics on the electrical supply and lubrication and cooling issues for the compressor.

The main reasons for the reluctance of the refrigeration industry to adopt inverter based variable speed drives (VSDs) are the relatively high capital cost and the fact that initial demonstration installations have failed to produce the expected energy savings. More studies, as above, confirming the energy benefits could well change this attitude.

3.2.3 Evaporator coil defrost

Frosting up, and subsequent defrosting, of the evaporator coil results in a significant use of energy. In the first case, frost formation reduces the efficiency of the evaporator and it must be removed to maintain the system performance, and secondly the removal of the frost and ice adds heat to the cabinet. It has been estimated that in supermarket applications the combination of defrost and anti-sweat heaters to avoid misting of the cabinet doors may account for more than 30% of the total refrigeration system electrical energy requirements. Consequently, improvements in the efficiency of defrosting evaporator coils can have a significant benefit to the overall operating energy consumption.

The two main considerations for defrosting are:

- defrost method
- defrost initiation/termination control.

A variety of methods of evaporator defrost are available and each has its own benefits and disadvantages.

Defrost Methods

Off-cycle defrost

The refrigerant flow is ceased and the ice is melted by air circulating over the evaporator. This is slow and only applicable to temperatures above about 1^{o}C. The initiation can be controlled by a number of methods including a combination of: simple time clocks, suction pressure, and temperature, to both initiate and terminate the cycle. It is a low cost system and mostly employed with relatively higher temperature fixtures.

Electric defrost

For low temperature refrigerators (i.e. below 0^{o}C), or when rapid defrost is needed, electric resistance heating is used for defrost. This can be either within the evaporator body, or more commonly, externally mounted. The use of electrical defrost systems increases the capital cost of the units, and the operating energy to melt the ice and frost means that operating energy costs are higher.

Electric defrost is normally initiated by a timer and terminated by either: time from initiation; pressure rise in the suction gas; temperature rise of the evaporator surface. Electric defrosting can cause steaming and the consequent frosting of other surfaces.

Latent heat defrost – Hot or cool refrigerant gas

Latent heat defrost is often called ‘hot gas’ defrost as it uses high temperature refrigerant gas from the compressor to melt the ice and frost. The evaporator is in effect made to operate as the condenser. This technique is most appropriate for a multiple evaporator system so as to provide a balance within the system between the cooling load and the defrost load. Hot gas defrost is a fast defrost method but because of the high differences in temperature the thermal shocks can cause deterioration of the plant. Consequently, ‘cool gas’ defrost has been developed to reduce the temperature variation in the pipe-work and evaporator.

Typical defrost strategies in stores

Hot gas defrost is the most popular method for industrial applications, whereas for smaller and intermediate sizes electric defrost is employed. Each system has its own advantages and disadvantages, with hot gas being considered more efficient but more complex in design terms. Electric defrost has lower capital cost and is more simple to design. However, the overall energy impact of adopting these different systems is not fully understood

Defrost control

Defrost control is possibly as important as the means of defrosting. However the industry has in general adopted a timing system for defrost initiation even though this is not the optimum method. A more effective method would be based on the need for defrosting – “demand defrost”.

A wide range of techniques for detecting the correct time to initiate the defrost cycle has been researched. These include:

adaptive timing defrost – varying the time of onset to the prevailing conditions
air pressure sensor – measuring the fall of air pressure across the coil
combined air temperature difference with air flow – measuring both flow and temperature difference across the coil
evaporator temperature and pressure – measuring refrigerant conditions in the coil
thermal conductivity – the temperature difference between the coil surface and the air off the coils
capacitance - the capacitance of the ice layer as a measure of thickness
acoustic resonant frequency – determining the resonant frequency of the ice mass
infra-red detection – detects reflected infra-red radiation from the frost
fibre-optic – transmitted or reflected light to measure thickness
fuzzy logic control – combination of a range of variables to initiate defrost

Research is under way to establish new methods for controlling 'demand defrost' initiation with particular reference to supermarkets and the varying demands in response to seasonal changes in humidity.

3.2.4 Controls

Traditional control systems have separated the control of the cabinet from the compressor/condenser unit. Either an expansion valve or pressure regulator controls the cabinet temperature, and the refrigeration pack is controlled by evaporator pressure. The condensing (or head) pressure may be allowed to float in order to provide more efficient operation.

Developments in communication and control technology are opening the path to integrated systems based on building control communication protocols such as Lonworks. This and the ability of the modern systems to 'self learn' and thus control the system at its optimum is a benefit that may come as the systems become more widely available and costs are reduced.

The integration of the HVAC and refrigeration system controls will enable savings to be made in running costs although a higher capital cost will be encountered.

3.3 THE STORE ENVIRONMENT

The energy use of a refrigerated display cabinet should not be considered in isolation from its operating environment, and the interaction with customers. Both of these have a significant impact on the operating characteristics and performance of the cabinet.

3.3.1 Local environmental conditions

Understandably retailers insist that cabinets displaying refrigerated goods do not impede the customers' view of, or access to, the chilled products. Consequently, the cabinets tend to be either open to the store, with an air curtain providing separation, or fitted with heated glass doors.

The interaction between the store and the cabinets takes place in both directions, i.e. the store affects the cabinet efficiency and the cabinet affects the store conditions:

- The open cabinets lead to cold feet from the cold air spillage, and the heat removal from the cabinets can cause an increase in the heating requirement for the store.

- High relative humidities (>55%) can significantly increase energy use because of increased defrost energy for the evaporators and the requirement for increased use of anti-sweat heaters on glazed doors.

- The air-flows around the cases, the humidity, and temperature all have an effect but optimising these to achieve good energy efficiency has to be balanced against achieving the comfort needs of the customers and staff.

One study on the dehumidification of the store air from 55% to 35% RH showed savings of 20 to 30% in compressor energy, 40 to 60% in defrost energy, and 19 to 73% in anti-sweat heater operation.

Trials are ongoing into the use of special raised floors from which spilled cold air in the refrigeration zone aisles is extracted and returned to the main HVAC plant (but not as part of the Phase 2 works).

Computational fluid dynamics (CFD) modelling is a useful tool that may enable the designer to avoid some of the problems associated with very localised air flow patterns. However, CFD as a technique requires a considerable skill in use and not a little experimental validation.

3.3.2 Heat recovery

The energy flows within a supermarket store (heat and coolth) are large enough to make the investigation of heat recovery techniques a worthwhile exercise. However, there are many diverse sources of heat - from lighting gains to extract air from bakeries - and the integrated recovery of these needs a case by case examination. Some of the opportunities to recover energy from the store's systems are looked at in more detail in the full Phase 1 Report – see Volume 2.

3.4 CONCLUSIONS

The state of the art review of advanced supermarket refrigeration has shown a number of areas in which significant energy savings may be made and environmental impact reduced. From the overall design of the energy supply, from CHP and CCHP, through to the interaction of the store environment and the cabinets, savings can be made. These factors offer great scope to bring about reductions in the energy use and TEWI of supermarket refrigeration systems.

Following on from these individual reviews, all three aspects – system types, system components and the store environment were investigated further, through four specific research projects:

- Secondary systems and fluids –evaluation of alternatives evaluated and comparison to DX systems
- Defrost – a study of different techniques and means of control
- CHP/CCHP – development of a system simulation tool for their application in supermarkets
- Store and cabinet interaction – a study in laboratory conditions simulating store environments.

4 SCOPE OF THE PHASE 2 RESEARCH


4 Scope of the Phase 2 Research

In Phase 2, the UK undertook to provide a detailed examination of four of the technologies which had featured in the Phase 1 research. The aim was to come to a definitive result as to the current performance of the technology involved. However, it must be borne in mind that circumstances change and technology continues to develop more effective measures and so these results relate to the current situation. Future developments may well change the findings here, as the financial and legislative climate changes the relative values of the materials in use – gas electricity, refrigerant etc.

Each of the studies was carried out by an appropriate research organisation with experience in field studies and laboratory test procedures. When possible they used existing equipment and installations to develop their findings. This ensured that the results of the research are as relevant as is possible for the retailers who must put the findings into practice. The topics and research bodies for this second phase of investigations are as given below:

Topic	**Methods**	**UK Partner**
Secondary Systems	Laboratory determination of fluid properties and simulation of system performance.	Brunel University
Defrosting control	Environmental test chamber determination of defrost behaviour.	Brunel University
Combined Heat and Power (CHP) and Combined Cooling Heat and Power (CCHP)	Development of calibrated mathematical model and parametric examination of feasibility of CHP and CCHP.	South Bank University
Display Case and Store Environment Interaction	Field study and environmental chamber experiments on store environmental conditions and display cabinet energy use.	EA Technology

5 PHASE 2 RESEARCH RESULTS

5 Phase 2 Research Results

This section of the report deals with the individual research topics. It describes the aims, methods and results of the research. Each section is a summary of the work conducted and the results obtained. The topics researched are:

- Secondary systems
- Defrost control
- Combined heat and power and combined cooling heat and power
- Store environment and display cabinet interactions.

5.1 SECONDARY SYSTEMS

5.1.1 Aims and Methods

The aim of the project was to investigate the viability of secondary systems in the UK. The study was carried out by simulating a secondary system, using a specially developed model of energy use in a supermarket. The study into secondary systems had three key parts:

- a typical store was monitored to develop and calibrate a model of store refrigeration
- the fluid properties of secondary refrigerants were determined for simulation of secondary system performance
- the calibrated model would be used to determine the operational energy use and TEWI for the different refrigerants and leakage rates to identify the optimum system.

5.1.2 Introduction

The majority of large modern supermarkets in the UK employ centralised multi-compressor refrigeration systems that distribute refrigerant to the evaporator coils in refrigerated cabinets and cold rooms. These centralised systems offer a number of advantages:

- they allow for centralised heat rejection to ambient, thus reducing summer cooling loads, and noise in the retail area
- they enable the application of centralised controls for system optimisation
- they allow for heat recovery and integration of the refrigeration system with the space heating, ventilation, and air conditioning (HVAC) system.

However, a major drawback of the centralised vapour compression `direct' refrigeration system is the large quantity of refrigerant circulating in the heat exchangers and distribution piping. This can lead to high refrigerant leakage rates that contribute significantly to the TEWI. Centralised systems of this type are also constructed on site and reliant on on-site quality control for the leak tightness of the installed system.

Secondary refrigeration systems have a centralised refrigeration system to cool a secondary fluid which is circulated to the refrigerated cabinets and other cooling coils in the store. This arrangement reduces the amount of refrigerant used in large supermarket refrigeration systems and therefore potentially reduces the environmental impact. They are also usually factory assembled, which helps to reduce refrigerant leakage in use.

In the UK, although a number of supermarket chains have experimented with secondary systems, they have not yet become common place due to uncertainties about their owning and operating costs as well as their environmental impacts compared to conventional `primary' systems.

5.1.3 Feasibility Study Test Case

The feasibility study was carried out using data from a medium size supermarket in Airdrie, Scotland, see Figure 1, The total sales floor area was approximately 2400 m^2 and the annual electrical energy consumption was of the order of 2000 x 10^3 kWh of which around 1000 x 10^3 kWh was consumed by the refrigeration equipment. The refrigeration system consisted of 3 high temperature packs serving medium and high temperature cabinets and 2 low temperature packs serving low temperature cabinets. The packs used semi-hermetic reciprocating compressors. Heat rejection was achieved with multi-fan air-cooled condensers located on the roof of the supermarket, Figure 2.

The packs operated with fixed suction and head pressures and heat recovery was employed from the high temperature packs. The recovered heat was used to heat the ventilation air.

Figure 1: Store used for feasibility study

Figure 2: Air cooled compressors on the roof of the store

The supermarket was equipped with a central monitoring and control system and data from the monitoring programme were used to validate the supermarket simulation model. A schematic of the secondary system is shown in Figure 3.

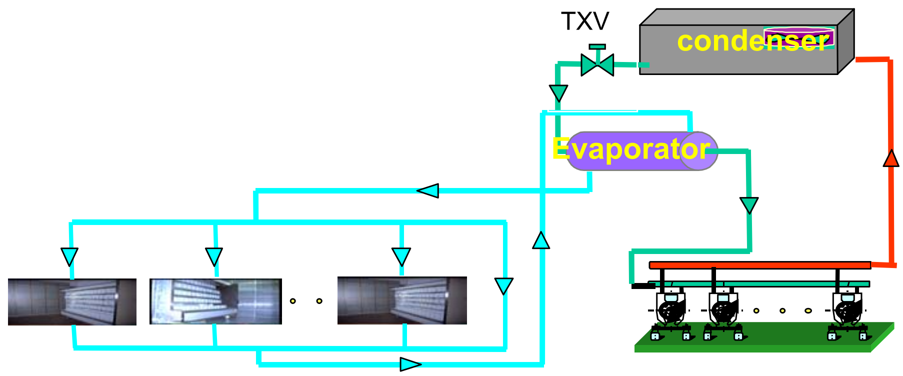


Figure 3: Schematic of secondary refrigeration system

5.1.4 Modelling Methodology

The model (SuperSIM) was developed within the TRNSYS simulation environment as this offered the greatest flexibility in terms of combining the various components of the systems to be modelled. The main elements of the model included:

- indoor store environment – from standard multi-zone TRNSYS building model
- store HVAC and heat recovery system
- primary high temperature refrigeration system
- primary low temperature refrigeration system
- compressor
- condenser
- display cabinet and cold room evaporator coils
- expansion valves
- control system.

For modelling the secondary refrigeration systems, the primary subsystems were linked to the secondary subsystems by a plate heat exchanger. The refrigeration load of the display cabinets was met by varying the flow of secondary fluid.

5.1.5 Secondary Fluid Properties

The main characteristics and the impact on the performance of the secondary refrigerants are:

- freezing point – operating temperature
- density – pumping power
- viscosity – pump power and heat transfer
- specific heat – heat transfer coefficient and volume flow rate
- thermal conductivity – heat transfer and temperature difference.

For a total of 15 secondary liquid refrigerants (see below) these properties were used to rank their performance and also for implementation in the model.

Type of solution	Symbol
Ethylene glycol/water	EG
Propylene glycol/water	PG
Ethyl alcohol/water	EA
Methyl alcohol/water	MA
Glycerol/water	Glyc
Ammonia/water	NH_3
Potassium carbonate/water	K_2CO_3
Calcium chloride/water	$CaCl_2$
Magnesium chloride/water	$MgCl_2$
Sodium chloride/water	NaCl
Potassium acetate/water (Tyfoxit 1.x)	KAc
Saturated Ammonia water	$NH_3.H_2O$
Potassium Formate (Tyfoxit F20)	Kfo
Potassium Formate (Tyfoxit F40)	Kfo
HFEL-13938	

Figure 4, shows a typical example of investigations into the different secondary refrigerants.

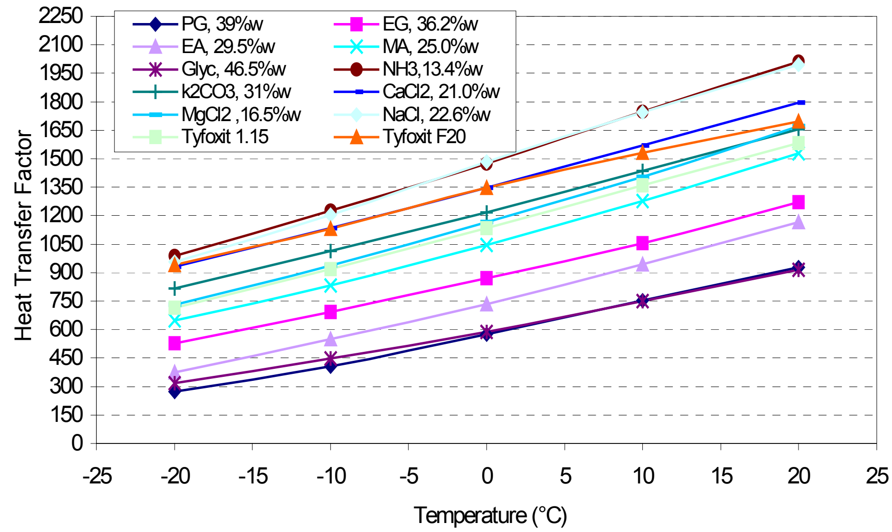


Figure 4: Variation of heat transfer factor with temperature for various secondary refrigerant fluids for medium and high temperature applications

A simulation program called '2ndREFProp' was used to determine these properties when mixed with various quantities of water. On the basis of these properties, and the store refrigeration model, the system performance could be calculated and compared with the direct system.

5.1.6 SuperSIM Model Validation

The data from the monitored supermarket were used to validate the model and a number of parameters were used to establish the accuracy of simulation. The model predictions matched the measured performance reasonably well, given the constraints of the modelling time step for annual simulations. The model was considered accurate enough to be used in the comparison of the primary and secondary system performance.

5.1.7 Comparison between Direct and Indirect Systems

Using the calibrated model a number of alternative secondary refrigerants and systems were compared with a direct system using R22.

System No.	Direct System		Indirect System			
	HT Pack	LT Pack	HT Pack	HT Pack Cabinets	LT Pack	LT Pack Cabinets
1	R22	R22	-	-	-	-
2	-	-	R290	PG (40%)	R290	Kac(40%)
3	-	-	R22	PG (40%)	R22	Kac(40%)
4	-	-	R290	EG (36.2%)	R290	EG (56.2%)
PG= Propylene glycol/water ;EG= Ethylene glycol/water; Kac= Potassium acetate/water						

Table1: Modelled options for secondary system refrigerants

The overall energy use, including compressor power, condenser fan power, pumping power, and defrost energy, was 30% more with the secondary system. Most of this could be attributed to the pumping energy.

The TEWI was calculated for the four systems simulated over a 15 year life with annual charge leakage rates of 5%, 10%, and 15%, see Table 2.

This showed that the TEWI of an R22 direct system, at 5% leakage rate, was less than all the secondary systems simulated. At a leakage rate of more than 15% the TEWI of the direct system exceeds that of the secondary systems considered.

Table 2 Comparison of systems on the basis on TEWI

	System Number			
	System 1 Direct R22	**System 2 Indirect Primary-R290 HT-PG 40% LT-Kac 40%**	**System 3 Indirect Primary-R22 HT-PG 40% LT-Kac 40%**	**System 3 Indirect Primary-R290 HT-EG 36.2% HT-EG 36.2%**
Refrigerant Charge (Kg)	1295	130	293	130
Electrical Energy Consumption (kWh)	1217060	1690421	1694996	1726617
TEWI - Direct Effect (Kg CO_2)	5144344	326	416974	326
TEWI - Indirect Effect (Kg CO_2)	9675626	13726608	13475220	13726608
Total TEWI 5% Leakage (tonnes CO2)	11519	13439	13892	13727
Total TEWI 10% Leakage (tonnes CO_2)	13169	13439	14266	13727
Total TEWI 15% Leakage (tonnes CO_2)	14820	13440	14639	13728
PG= Propylene glycol/water ;EG= Ethylene glycol/water; Kac= Potassium acetate/water GWP for R290 = 3 GWP for R22 = 1700 Generation factor = 0.53 kg CO_2/kWh System Lifetime = 15 years				

5.1.8 Cost Comparison

Capital costs of systems were obtained for the supply and installation of the various designs based on the case study supermarket above. This showed secondary systems to be approximately 28% more expensive than a conventional direct system. This confirms the UK experience of increased costs of between 15% and 30% for secondary systems.

In terms of operating costs the results showed that a well engineered direct system with a leakage rate of 5% per annum of R22 would have a lower capital cost, lower running cost and lower TEWI than an indirect system.

Table 3 Comparison of the systems on the basis of cost.

	System No.			
	System 1 Direct R22	**System 2 Indirect Primary-R290 HT-PG 40% LT-Kac 40%**	**System 3 Indirect Primary-R22 HT-PG 40% LT-Kac 40%**	**System 4 Indirect Primary-R290 HT-EG 36.2% HT-EG 56.2%**
Compressor Load (kW)	304	322	325	348
Cabinet Length (m)	266	266	266	266
(A) Total Cabinet Cost (£)	506407	506407	506407	506407
(B) Cost of Packs (£)	116860	123838	124721	133589
(C) Cost of pipework (including insulation and installation) (£)	28693	145154	145154	143943
(D) Total Cost (A+B+C)	651960	775399	776282	783939
Installed Cost Direct System (D x 1.4) Indirect System (D x 1.5) (£)	912744	1163098	1164423	1175909

Table 4 Comparison between direct and indirect systems.

	System No			
	System 1 Direct R22	**System 2 Indirect Primary-R290 HT-PG 40% LT-Kac 40%**	**System 3 Indirect Primary-R22 HT-PG 40% LT-Kac 40%**	**System 4 Indirect Primary-R290 HT-EG 36.2% HT-EG 56.2%**
Capital (installed) Cost (£)	912744	1163098	1164423	1175909
Annual Electrical Energy Consumption kWh	1217060	1690421	1694996	1726617
Annual Running Cost at 0.05 £/kWh (£)	60853	84521	84749	86330
Total TEWI 5% Leakage (tonnes CO2)	11519	13439	13892	13727

5.1.9 Conclusions

Over the last few years a number of indirect systems have been installed in the UK and many of these installations have experienced problems, particularly during commissioning and the initial stages of operation. Many of the problems have occurred due to lack of sufficient knowledge in the refrigeration industry on the design and installation of indirect systems.

Although no data from actual installations are available for the running costs of equivalent direct and indirect systems, the running costs of indirect systems are likely to be higher than the running costs of direct systems due:

- to operation at slightly lower evaporating temperatures because of the additional primary-secondary refrigerant heat exchange
- the secondary fluid pumping power which can be significant.

Simulation results and capital cost analyses have shown that for indirect systems to become attractive alternatives to direct systems in UK supermarkets, design improvements need to be implemented to reduce both the capital and running costs of these systems. More stringent legislation and incentives may also contribute to a wider application of secondary systems.

5.2 DEFROST CONTROL

5.2.1 Aims and Methods

The aim of this project was to investigate and compare the three alternative defrost methods with respect to defrost efficiency and energy consumption and quantify the benefits that could arise from the use of demand defrost. Both in store monitoring and laboratory tests were employed.

5.2.2 Introduction

Defrosting of the evaporator coils, which are normally located in the base of the cabinets, is necessary because they operate at temperatures below the freezing point of water. With continuous operation, frost will accumulate on the coil surface leading to a decrease both in the air-flow rate and in the overall heat transfer coefficient. To maintain satisfactory performance, evaporator coils require defrosting periodically. It is claimed that the defrost cycle can use as much as 30% of the operating energy of a refrigeration system. Therefore this topic has been investigated as part of this project.

5.2.3 Typical UK Supermarket Defrost Methods

In the UK three methods of defrost are employed in supermarkets:

- electric defrost
- saturated gas or, as it is more commonly known, `hot or cool' gas defrost
- 'off-cycle' defrost.

Typically, electric defrost is used in small and intermediate size plants, while hot and cool gas defrost are used in larger plants. However, the design of supermarket refrigeration systems has changed substantially in recent years. The use of fixed head pressure control combined with heat recovery and cool gas defrost is now being replaced by floating head pressure control and electric defrost. There is also an increasing trend in the use of off-cycle defrost for medium temperature systems.

5.2.4 Energy Use in Defrost

Electric defrost

In electric defrost, the thermal energy to melt the ice is provided by an electric strip heater which is situated across the face of the coil. During defrost, the refrigerant supply to the evaporator coil is switched off, the electric heater is switched on, and the evaporator fans blow warm air, heated by the strip heater, through the coil, melting the ice from the coil surface.

Gas defrost

Hot and cool gas defrost techniques are implemented in remote systems. The former involves the circulation of hot gas from the compressor discharge manifold directly to the display cabinets, whereas the latter utilises cooler gas from the liquid receiver. The cool or hot gas condenses in the evaporator, releasing heat that melts the ice from the coil. In integral systems, hot gas defrost is effected by reversing the flow of refrigerant during the defrosting process, with the evaporator becoming the condenser and the condenser the evaporator during the short defrosting period. This method of defrost, which is widely applied to small heat pumps and air conditioning units, is also known as reverse cycle defrost.

Off-cycle defrost

In off-cycle defrost, no external heat is supplied to the evaporator to melt the ice. For this reason this method of defrost can only be applied to medium or high temperature cabinets in which the temperature of the air in the cabinet is above 0^oC. During defrost, the refrigerant supply to the coil is switched off and air is circulated over the coil to melt the ice. In open display cabinets this air is a mixture of cabinet and ambient store air entrained with the cabinet air along the air curtain.

5.2.5 Defrost control

The defrost controls are very important for successful defrosting because they dictate the exact moment at which defrosting starts and the moment at which it will be terminated. Normally, irrespective of the defrost method employed, defrost is initiated on a fixed elapsed time from the previous defrost cycle. It is then terminated when the evaporator coil air off temperature has reached a pre-set value, normally 12^oC, or after 30 minutes from defrost initiation, whichever is sooner.

It is widely acknowledged, however, that time defrost may cause a number of unnecessary defrost cycles and this reduces the energy efficiency of refrigeration systems, as well as the accuracy of temperature control of refrigerated display cabinets.

Defrost Initiation

Normally, electro-mechanically or electronically operated timing devices are used to initiate the defrost cycle at predetermined time intervals which depend upon the size and application of the refrigeration system. Time defrost initiation is very popular because it has low cost, it is simple to install and easy to maintain. The drawback with this method of defrost initiation, however, is high energy consumption during operation under low frost accumulation conditions.

Defrost Termination

Traditionally, defrost termination has been performed on a fixed time basis or by sensing a number of parameters such as the temperature rise of the air at the coil outlet, the temperature rise of the coil surface or the rise in the evaporator pressure.

A number of authors have considered the use of variable defrost time but it is difficult to predict the required time accurately due to the non-homogeneous nature and complexity of frost formation on the coil and the transient nature of the defrost cycle.

Sensing the air temperature near the evaporator could provide adequate control for defrost termination. The optimum position for the evaporator temperature sensing points, however, can vary depending on the geometry of the coil and it is difficult to find a suitable location for the sensor to ensure that all parts of the evaporator were free from ice when the defrost cycle is terminated. An appropriate solution to this problem could be the use of a number of temperature sensors located at different places over the evaporator coil.

5.2.6 Demand Defrost

Implementing defrost when it is required, or on demand, should lead to a reduction in the number of defrost cycles and offers the potential for substantial energy savings and improved product quality and life. Many techniques have been investigated, and these were reported as part of the Phase 1 review. For this second phase the research concentrated on examining the defrost energy use as a function of the defrost timing and duration and the temperatures of the working fluids – refrigerant and air.

5.2.7 In Store Measurements and Laboratory Tests

To better understand the energy impact of defrosting display cabinets, the characteristics of commercial equipment were studied under actual supermarket conditions and also under laboratory controlled conditions.

Store Monitoring

A supermarket in Scotland was selected for the monitoring of in-service performance. The store had 2400m^2 of sales area and 266m length of display cases. It also had cold rooms and holding areas. The refrigeration plant comprised high and low temperature systems served by a total of 17 compressors. The defrost method was cool gas with time termination. The store was heated by an underfloor heating system, with the ventilation air preheated to 22^{o}C from heat recovered from the refrigeration packs supplemented by a gas boiler. The store environment was monitored and a sampling system set up to collect condensate from the evaporators.

Laboratory Measurements

A 2.4m remote cabinet rated at 3.7kW of cooling was installed in an environmental test chamber. The cabinet was loaded with plastic containers filled with water and food substitute material. The refrigerant used was R22, fed from a mini-compressor pack designed to emulate the operation of standard supermarket packs. The refrigerant flow rate was measured using a Coriolis mass flow meter. Air and refrigerant temperatures were measured at various points in the cabinet using thermocouples. Measuring points included:

- Refrigerant temperature at inlet and outlet of the evaporator;
- Surface temperature at a number of points on the evaporator coil;
- Air temperature entering the evaporator coil (air on temperature);
- Air temperature leaving the evaporator coil (air off temperature);
- Air temperature in the back flow tunnel;
- Air velocity at the evaporator outlet;
- Product temperature at various points in the cabinet.

The operation of the system, which was controlled by a standard supermarket controller board, was monitored using a computer based data acquisition system.

5.2.8 Store Monitoring Results

The average internal temperature remained fairly constant throughout the year. In the winter months the temperature was just below 20^{o}C whereas in the summer months the average internal temperature rose to 21^{o}C. The average external temperature increased from around 5^{o}C in winter to around 18^{o}C in the summer. The temperature in the aisles of the chilled food area varied between 18^{o}C and 20^{o}C in the winter and between 17^{o}C and 20^{o}C in the summer. The temperature in the dry goods area in the summer rose to 23^{o}C.

The parameter that most influences the rate of frost formation on the evaporator coil is the moisture content of the air in the store. The monthly average external relative humidity remained within the range between 60% and 80%. The indoor monthly average relative humidity increased from about 28% in the winter months to a maximum of 50% in the summer months. Figure 5 below, shows the variation of the monthly average external and internal moisture content (absolute humidity). It can be seen that in the winter months the average internal moisture content is almost the same as the external moisture content.

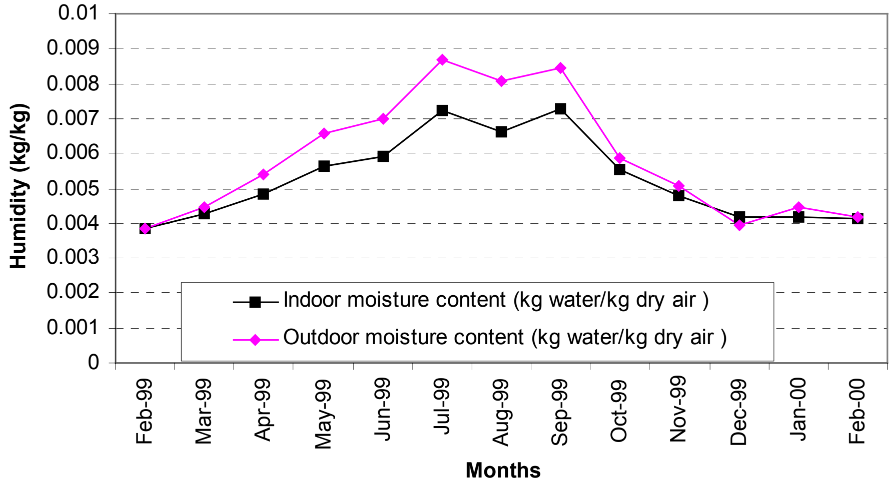


Figure 5: Variation of internal and external moisture content over a twelve month period

Figure 6 shows the variation of the average amount of condensate collected during each defrost for different values of store relative humidity and an ambient temperature of 19^{o}C. It can be seen that the condensate per defrost increased from around 4.0 litres of water per defrost cycle for a 30%rh, to around 7.5 litres per defrost cycle for 50%rh.

These field results indicate that a dairy case would ideally require up to 50% fewer defrost cycles in winter than the number of defrost cycles required in the summer. This would represent a considerable amount of energy saved where electric defrost is employed. However, in the majority of cases, chilled cabinets employ off-cycle defrost, so there would be no such saving.

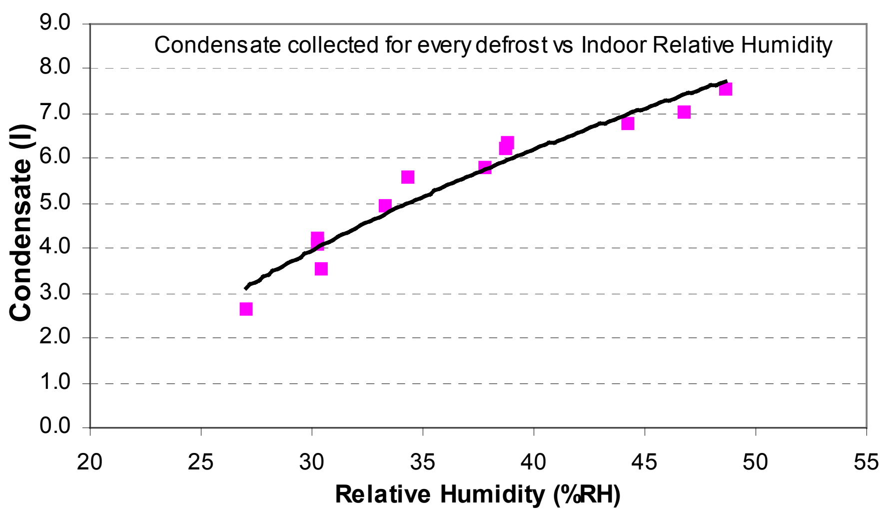


Figure 6: Variation of condensate per defrost cycle with relative humidity

5.2.9 Laboratory Test Results

Frost Formation

Figure 7, below shows the condensate collected from the coil of the test cabinet during defrost, after a cooling cycle of 440 minutes between defrost cycles, for different combinations of temperature and relative humidity. It can be seen that the condensate increases with both ambient temperature and relative humidity and so in environments where both temperature and relative humidity can vary, the effect of both parameters should be considered either directly or indirectly in the design of a demand defrost control system.

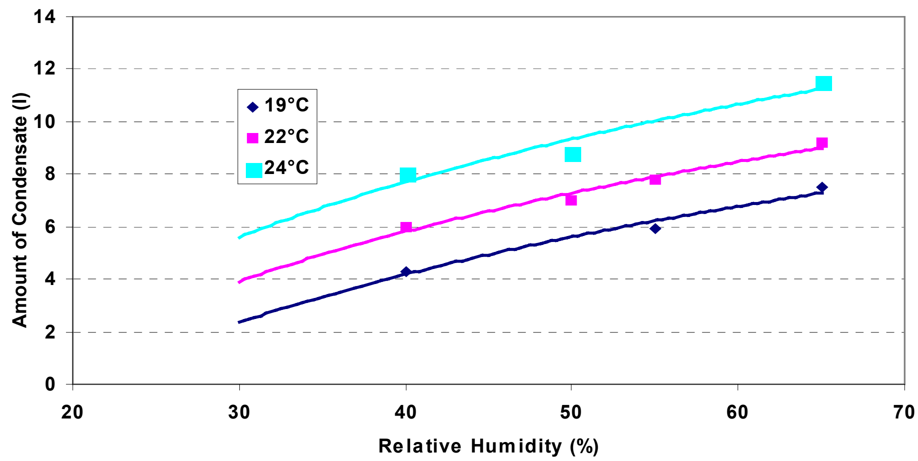


Figure 7: Influence of relative humidity and temperature on frost accumulation on the coil for a 440 minute operating cycle.

Studying the air velocity from the evaporator after a defrost cycle showed a significant increase in fall-off of velocity with increasing %rh of the room air. This shows a slower rate of frost formation with lower % rh and the possibility of longer periods between defrost cycles without adverse consequences. A similar indicator of the need for defrosting is a change in the temperature immediately after the evaporator coil. This parameter again responded to varying environmental conditions and could be used to optimise defrost periods.

5.2.10 Defrost Techniques

To compare the effectiveness of different defrost methods tests were carried out in the environmental test chamber in the laboratory on the multi-deck chilled food cabinet, using electric, cool gas and off cycle defrost. The defrost cycle tests were initiated after the cabinet was operated at constant ambient conditions of 19^{o}C and 50%rh for 6 hours to ensure that sufficient frost accumulated on the coil prior to defrost. The ambient conditions chosen are representative of conditions in a UK supermarket in the summer months.

Electric Defrost

The time taken for defrost was taken as from the end of refrigerant pump-down until the air-off temperature reached the target value of 12^{o}C. Experiments were carried out with two ratings of electric heaters - 1.6kW and 3.2kW. The defrost times were 19 and 15 minutes respectively, showing a shortening of defrost period of 4 minutes with the doubling of heater size.

Cool Gas Defrost
Using the same conditions as for the electric defrost, the time for cool gas defrost was 8 minutes. The major saving in time for the gas defrosting was in heating the coil to 0°C after the ice had melted.

Off-cycle Defrost
Within the period of 30 minutes from defrost initiation, the coil air-off temperature had not reached the 12°C set-point, consequently the defrost cycle was terminated on time rather than temperature. Also it was observed that the coil surface and coil air-off temperature equalised approximately 25 minutes from defrost initiation, indicating that the coil had cleared completely of frost and condensate, even though the air-off temperature was as low as 7°C.

5.2.11 Energy Requirements of Defrost Techniques

Using the findings from the laboratory/test room experiments, as described above, the energy requirement for defrosting in the case study store can be calculated for the different defrost techniques. This shows a considerable reduction in energy use for the hot and cool gas defrost methods when compared with the electric defrost system. For example, the overall electricity use for hot gas defrost of a low temperature cabinet can be less than 20% of that required with direct electrical heating.

However, the capital cost of the hot gas defrost system is very much greater than the electric defrost and consequently payback periods are approximately 28 years for low temperature cabinets and 9 years for medium /high temperature cabinets.

Demand Defrost
Optimising the defrost regime will save considerable energy by avoiding either too frequent or too long defrost periods. The laboratory tests on a medium temperature cabinet can be seen in the Figure 8 below. Here it can be seen that at 19°C and 50%rh the optimum time between defrosts will be 9 hours rather than the 6 hours currently used in most supermarkets.

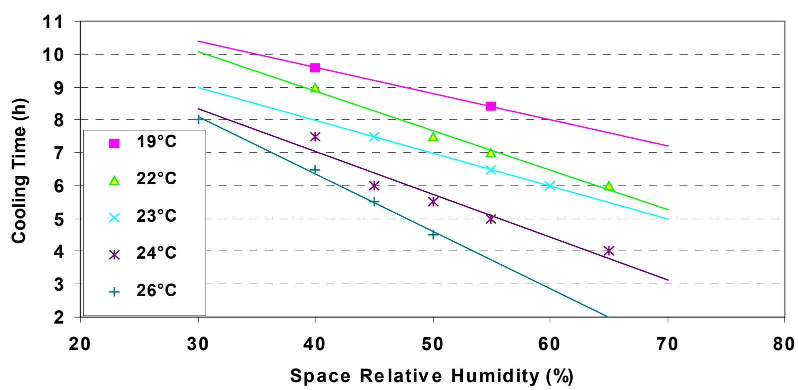


Figure 8: Optimum time between defrosts for a medium temperature multi-deck cabinet.

Another parameter currently being used by a commercial refrigeration controls manufacturer, as the basis for a demand defrost controller for optimum defrost initiation, is the duration of the previous defrost cycle. The duration of the defrost cycle being the time elapsed from initiation of the defrost cycle to the time the air off coil temperature rose to 12°C. A short defrost cycle indicates a small frost quantity on the coil and slow frost growth operating conditions for the display cabinet. A long defrost cycle would indicate heavy frost built-up on the coil and operating conditions promoting high rates of frost growth. The length of the previous defrost cycle can thus provide an indirect indication of the optimum frequency of defrost for a given cabinet and operating conditions. For the cabinet tested which employed electric defrost, the variation of the optimum number of defrost cycles, based on the length of the previous defrost cycle is shown in Figure 9.

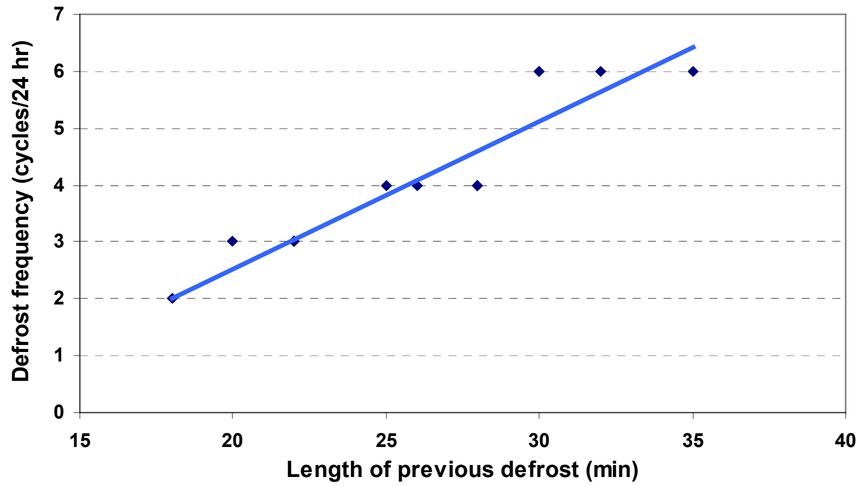


Figure 9: Variation of defrost frequency with length of previous defrost cycle for electric defrost.

From these experiments it can be shown that if defrost on demand was implemented the number of defrost cycles could be reduced by between 25% and 50% in the monitored supermarket. This would lead to savings of between 20,000 and 40,000 kWh per year. These savings could increase further if better defrost termination controls are employed to reduce the load imposed on the cabinets by the defrost cycle.

5.3 COMBINED HEAT AND POWER AND COMBINED COOLING, HEAT AND POWER (CHP AND CCHP)

5.3.1 Aims and Methods

The objective of the project was to investigate the viability of CHP and CCHP systems in supermarkets based on the critical parameters of energy utilization, overall efficiency, energy cost saving, reliability, capital cost and payback period, compared with a conventional energy-supply system. A specially developed and calibrated mathematical model was used for the investigations because real data were not available.

5.3.2 Introduction

The requirements of energy for a conventional supermarket are heating, cooling and power. In a conventional supermarket, the electrical demand for the requirements of lighting, refrigeration and various other appliances in the store is satisfied by mains electricity. A gas boiler normally provides the heat demand for the space heating and hot water. In the case of CHP scheme, the energy *demand* of the supermarket is the same as the conventional store but the CHP plant can be used to provide the energy by three distinct strategies. These control strategies are defined as ‘electrical led’, ‘heat led’ and ‘base load led’.

- *Electrical led* is normally selected to use on a site where there is no electricity supply from mains. The electrical demand of the store is wholly provided by CHP unit as well as part of heat demand of the store. A gas boiler is installed as well to meet the remainder of the heat demand.

- *Heat led* is normally used at sites where the generator runs in parallel with the mains. The heat demand of the store is wholly provided by the heat output of the CHP unit. In this case, the heat led strategy enables the engines to modulate to always satisfy the heat demand of the store so that a boiler is not needed. In this configuration the balance between electricity demand and CHP unit output can be maintained with the export or import of power.

- *Base load* led is used on a site where the engine is operated under full load to provide base electrical and heat demand of the store. The electrical supply is connected with the mains grid, and the balance between electricity demand and CHP unit output can be maintained with the import of power. A gas-fired boiler is installed to make up the extra heat demand of the store.

5.3.3 Mathematical Model of CHP and CCHP

The mathematical models for the conventional supermarket, CHP scheme, and CCHP scheme have been developed in the form of Excel spreadsheets. The model simulates both the building energy balance and the refrigeration loads. The input parameters to model the loads include:

- internal air temperatures
- average external dry bulb temperatures
- type and refrigeration load and the operating condition of the store
- occupancy loads
- electric lighting load
- global conductive heat loss coefficient
- ventilation rate
- infiltration rate
- fan power

In order to model the CHP scheme, the size, heat and electrical efficiency, control strategy and the capital cost of the CHP unit are added into the initial input data. For the CCHP model it is assumed that the cooling load is met by an absorption unit operating from the CHP heat supply. This technique is particularly useful to address low heat to power ratios. However, for low temperature applications cascaded systems are required.

The heat and electrical demand of the store can then be calculated based on these input data. The calculation procedure uses the BIN method to calculate the hourly steady state energy consumption and cost based upon input data, for every single day of the year, see Figure 10 below. In order to consider the non steady state effects such as energy storage in the building fabric during night setback and preheat, a time constant approach was used in analysing non-steady state heat transfer.

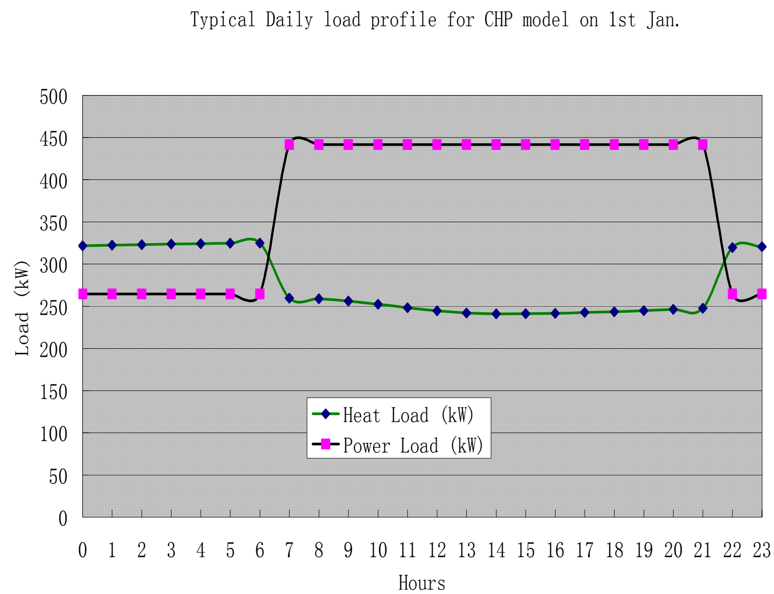


Figure 10: Output of model showing a daily profile of electrical load and heat demand.

The hourly energy consumption figures are summed to give annual usage. Annual energy costs were determined based on the tariffs of fuel and electricity.

The main outputs of the model are annual gas consumption and cost, annual electricity consumption and cost, and primary energy consumption and total energy cost. The calculation results such as the energy use (kWh/m^2/year) and energy cost (£/m^2/year) are also given. The criteria used to compare the viability of the different systems are energy costs, capital costs and primary energy usage. These are then related to the payback period compared with conventional energy-supply systems.

5.3.4 Conventional model validation

The model was validated against generic benchmark figures for conventional stores of 15,000 ft^2, 28,700 ft^2, and 40,000 ft^2 (1,394 m^2, 2,666 m^2 and 3,716 m^2 respectively). The annual energy consumptions and costs calculated for the three supermarkets are shown in Table 5. The results of the comparison showed that the model predicts, with reasonable accuracy, both the energy cost per unit area and the energy consumption.

	Model predictions typical supermarket			**Sainsbury's Data**	**BRE Typical**
	15,000 ft^2	**28,700 ft^2**	**40,000 ft^2**		
Energy use (kWh/m^2/year)	1252	1323	1341	1172	1254
Energy cost (£/ m^2/year)	42.2	44.4	44.7	47	-

Table 5: Comparison of predicted results with other supermarket energy data

5.3.5 CHP model validation

The CHP model has been validated by real data from a typical UK supermarket of 3,250 m^2 with an 'Air CHP' unit (further heat is extracted by passing ventilation air over the prime mover and generator which is sited inside the air-handing system). The air handing unit schematic is shown in Figure 11 below.

The total installed refrigeration cooling capacity was approximately 400-500 kW. It employed compressor packs operating on a common air cooled condenser to deliver cooling to distributed chillers (HT) and freezer (LT) circuits. The system was operated 24 hours every day, without night blinds, and the set point of the internal air temperature of the store was 20-22^{o}C, for all seasons without night setback.

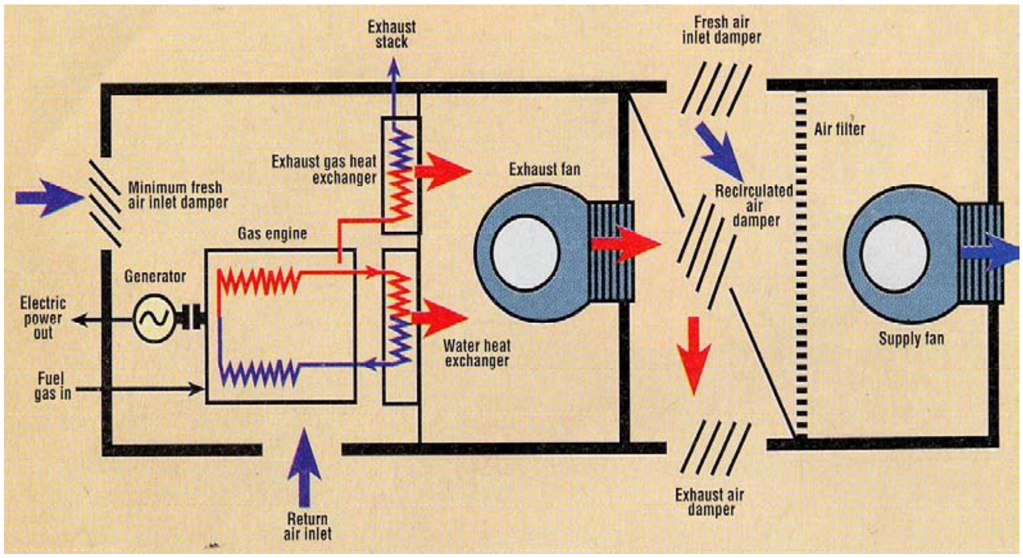


Figure 11: Schematic of Air CHP system.

The 'Air CHP' installation employed a 'base load' led control strategy. The CHP unit used a naturally aspirated gas engine as prime mover that provided a maximum electricity output of 147 kWe and thermal output of 336 kWt, from 508 kW of gas input (Gross Calorific Value), 29% electrical efficiency, and 95% CHP efficiency by design, to meet part of heat and power requirements of the store.

The energy utilization of the unit was monitored on-line which included data gathering and monthly reporting. Comprehensive monthly reports included details of days run, energy consumption, overall efficiency, energy cost saving, giving a short and long-term performance of the system based on the operating months.

The predicted results of the model are compared against real data in Table 6 and Figure 12. These show that the prediction is reasonably accurate.

Criteria and values	**Model predicted results**	**Data from store**
Peak Heat demand (kW)	416	500
Peak Electrical Demand (kW)	636	600
Cost saving (£/year)	35364	37778
Payback period (Years)	4.81	4.5
Note 1: In order to compare model predicted results with installed system, the same gas and electricity price were used. These are: gas (0.65p/kWh); Electricity Day (4.7p/kWh) Night (2.37p/kWh).		

Table 6: CHP model validation for CHP store.

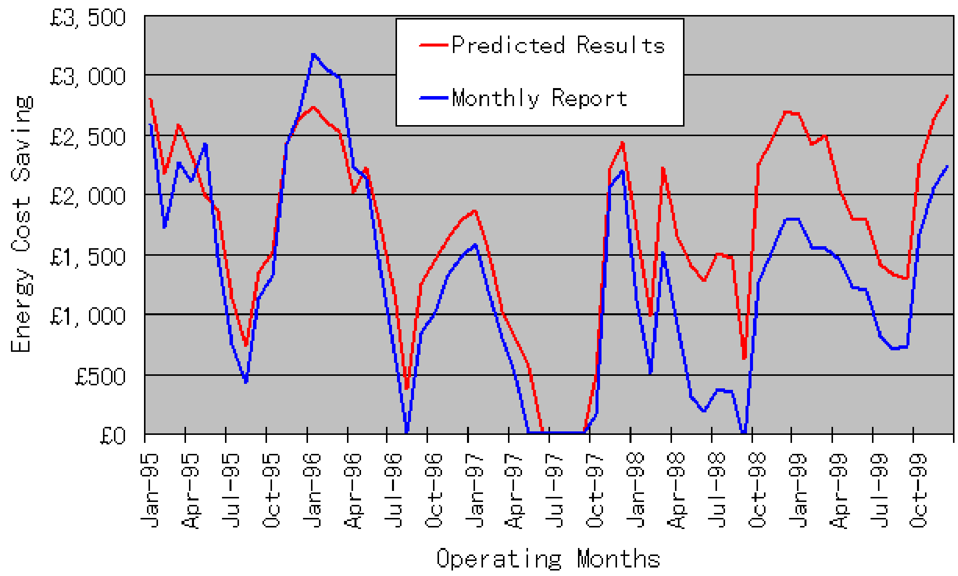


Figure 12. CHP model validation by energy cost savings

5.3.6 Investigation of CHP parameters using validated model

Based on the validated model, and using the validation store as a baseline, the effects of the following parameters were investigated:

- type of CHP prime mover
- control strategy of CHP unit
- fuel price.

5.3.6.1 Type of CHP Prime Mover

The results of the simulations showed that a gas engine would be more cost effective than a turbine, as seen in Table 7 below, largely as a consequence of its greater shaft efficiency and heat recovery potential, despite significantly higher annual maintenance costs.

	Traditional System	Gas Turbine	Gas Engine
Gas consumption (kWh/year)	2762860	5678600	4752106
Gas cost (£/year)	32891	51107	42769
Electricity used (kWh/year)	1744718	2039973	2039973
Electricity cost (£)	3327693	85181	85181
Primary energy used (kWh)	11519947	11046951	11049622
Additional CHP maintenance cost (£)	-	2575	7726
Total annual revenue cost (£)	182066	138864	135676
Energy Cost savings (£/year)	-	43202	46389
Primary energy savings	-	7%	14%
CHP Capital cost (£)	-	187000	170000
Payback period (years)	-	4.33	3.66

1. Safeway at Milton-Keynes is selected as the system investigated.
2. Electrical efficiency: gas turbine 23%; gas engine 28%.
3. Heat recovery efficiency: gas turbine 50%; gas engine 57%.
4. Maintenance cost (£/kWh): gas turbine 0.002 (£/kWh); gas engine 0.006 (£/kWh).
5. Capital cost (£): we assume that gas turbine has 110% of capital cost compared with that of gas engine.
6. Gas and electricity price for traditional system and CHP system were chosen to reflect the climate change levy concession that applies to Good Quality CHP.

Table 7: Investigation of effect of prime mover type on payback period.

5.3.6.2 Control Strategy

Figure 13 below, shows that for the given site, with an electrical load of approximately 400kW and a peak heat requirement of approximately 300kW, the payback period is insensitive to CHP control strategy up to the level of the site electrical demand. Above that the payback period will increase significantly with an electrical load lead control strategy.

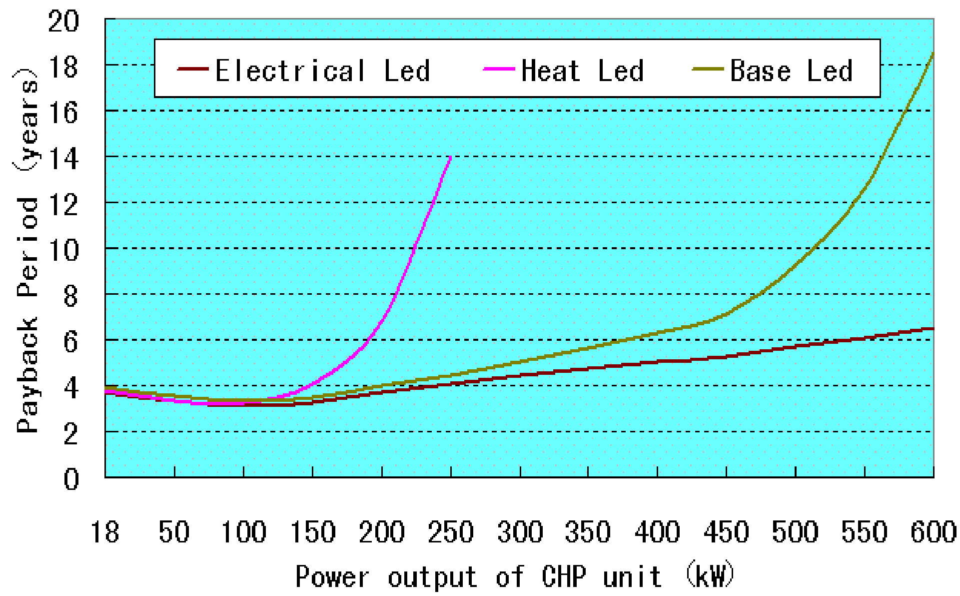


Figure 13: Effect of CHP control strategy on payback period.

5.3.6.3 Fuel price

Figure 14 below shows how changes in fuel price can effect the payback period of the CHP system. Note that the price variation is 10% in each case and real variations may be significantly more than this.

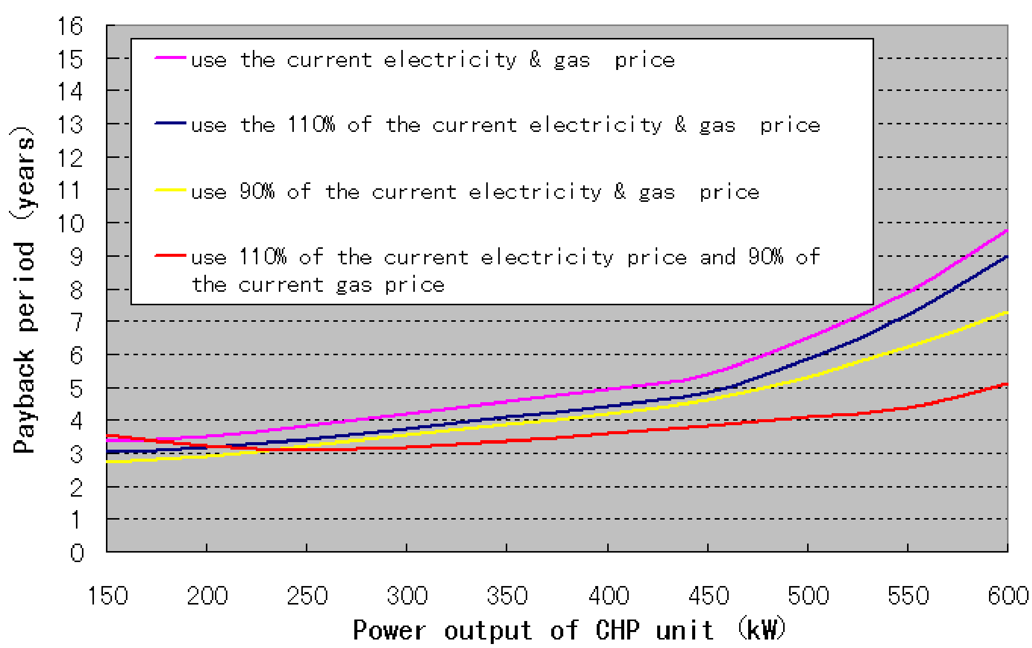


Figure 14: Effect of energy prices on payback period of CHP system.

These results indicate that, by selection of the correct size of CHP unit, payback periods of less than five years are achievable. However, this payback period is sensitive to control strategy, fuel price and CHP power output.

5.3.7 Investigation of CCHP using validated model

The simulation of the CCHP options required the model to include data on the size, type and operating regime of the absorption chiller. Table 8 below shows some of the characteristics of the CCHP schemes that were simulated.

Option	Absorption Chillers	Chillers COP	Medium Cooled	Medium Temp	Chilled food Refrigeration	Frozen food Refrigeration	Gas engine	Engine Efficiency	Water Temp
1	Single effect NH_3/Water	0.58	Propylene glycol	-8 to -4	Using cold glycol	Conventional Vapour compression	High Temperature	46%t/ 32%e	124
2	Double stage NH_3/Water	0.4	Propylene glycol	-8 to -4	Using cold glycol	Conventional Vapour compression	Conventional	57%t/ 33%e	90
3	Single effect LiBr/Water	0.71	Water	7 to 14	Cascade vapour Compression system In cabinet	Cascade vapour Compression system In cabinet	Conventional	57%t/ 33%e	90
4	Low Temp Single effect LiBr/Water	0.62	water	7 to 14	Cascade vapour Compression system In cabinet	Cascade vapour Compression system In cabinet	Conventional	59%t/ 33%e	70
5	Silica Gel/Water Adsorption	0.6	Water	7 to 14	Cascade vapour Compression system In cabinet	Cascade vapour Compression system In cabinet	Conventional	58%t/ 33%e	80

Table 8: Absorption chiller CCHP options.

5.3.7.1 Type of absorption chiller

The most cost effective of these was Option 3 (single effect lithium bromide) as can be seen in Figure 15 below.

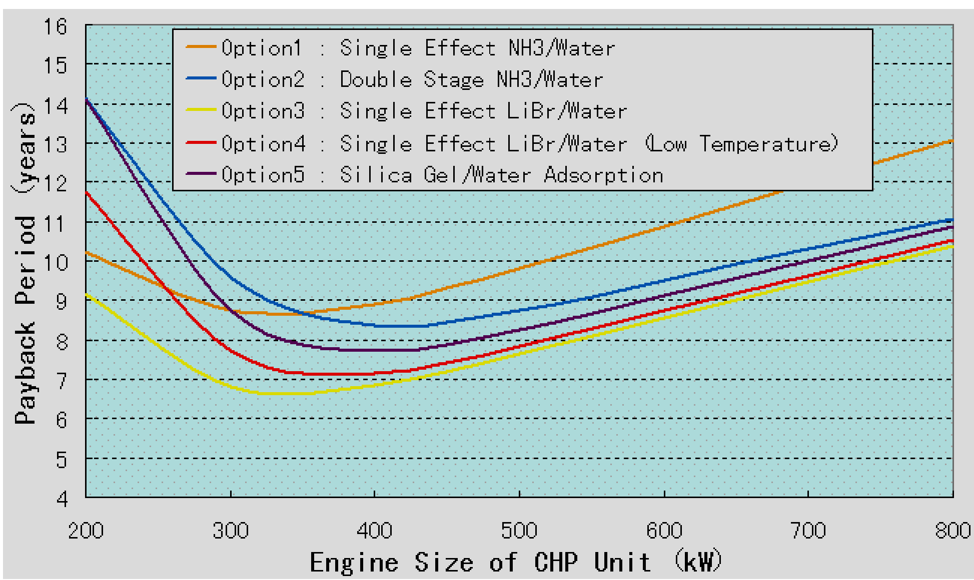


Figure 15: Variation of payback period with engine size and absorption refrigeration type

Further analysis showed that the payback period is dependent on operating hours of the plant expressed as Equivalent Full Load Hours (EFLH). Typically, the optimum operating regime would be approaching 8000 EFLH per year.

5.3.7.2 Effect of fuel prices

As for the CHP option, changes in fuel prices were simulated. The price sensitivity of the CCHP options is shown below in Figure 16.

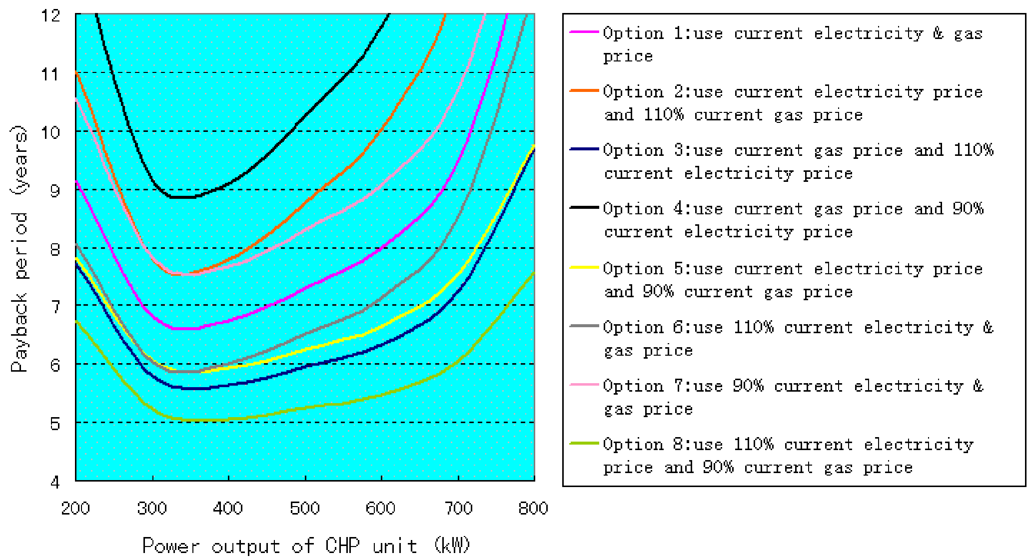


Figure 16: Payback of CCHP system as a function of fuel prices and system type.

The results of these predictions show the sensitivity of the payback period to the size of the unit and the relative costs of the fuels. Payback periods of less than six years are possible but show quite a high sensitivity to capital costs and fuel prices.

5.3.8 Conclusions

This project successfully produced a mathematical simulation tool for analysing the potential for CHP and CCHP in supermarkets. It was used to predict the payback periods of CHP and CCHP system under a wide range of operating conditions showing the financial viability of the respective schemes. CCHP, has an improved load match compared to CHP, when used with absorption chilling but this form of refrigeration is not a well recognised technology in supermarkets.

It was concluded that CHP and CCHP schemes are sensitive to both fuel price and operating regime. Higher load factors, as expressed by EFLH, tend to be more favourable. With regard to fuel price effects the most favourable situation is with increasing electricity price with decreasing gas costs. In practice the exact opposite has been occurring in the UK, significantly impacting on the take-up of CHP by the food retailers. However to encourage the take up of CHP, the UK Government has introduced a series of financial measures for quality CHP systems.

5.4 STORE ENVIRONMENT AND DISPLAY CABINET INTERACTIONS

5.4.1 Aims and Methods

This project determined how the store environment and the cabinet energy performance are related through a combination of in-store measurements and environmental test chamber experiments.

5.4.2 Introduction

This research concentrated on store/case interactions, in particular how the store environment influences the energy performance of the display case. Following a literature review of previous investigations in this area, the research was targeted on two key areas of interaction:

- Impact of local air flow patterns - in particular the balance between low level extraction and high level extraction.
- Effect of temperature and relative humidity of store air.

5.4.3 Literature survey

Initially, a brief literature search was made to look for any previous research on this subject. Little or no information was found covering the dependence on case performance of ventilation systems, including velocity and temperature. However, there were several papers referencing the dependence on relative humidity. Most indicated that lowering store relative humidity to at least 35% should provide significant savings in the store energy bills. It was also suggested that this could allow the anti-sweat heaters to be turned off with further energy savings.

Two ASHRAE papers - (Paper 3686 (CH-93-16-1); 'Effects of Store Relative Humidity on Refrigerated Display Case Performance' (RH Howell) and paper 3687 (CH-93-16-4); 'Calculation of Humidity Effects on Energy Requirements of Refrigerated Display Cases' (RH Howell) both suggest that significant savings can be made in defrost energy at lower store humidity, even given the extra HVAC energy required to dehumidify the air supplied to the store. Savings of between 5% and 30% would be achieved depending on the type of display cabinet.

5.4.4 In Store Measurements

The overall aim of this project was to investigate the effect of the store ventilation strategy on the power consumption of refrigerated display case units. The effects of the ventilation strategy conditions could not be satisfactorily tested by comparing the power consumption of the units in different stores with varying conditions, together with other uncontrolled external factors such as different case types. Therefore the test procedure was to use an environmental chamber to simulate store conditions and conduct the testing on a standard set of cases.

Consequently, in order that environmental conditions in-store could accurately be replicated, it was necessary to take aisle and near-case measurements of these conditions at two stores – one employing a standard ventilation strategy, and the other employing displacement ventilation. The measurements were carried out using an environmental trolley, allowing temperature, relative humidity and air speed to be measured at a range of heights and locations.

5.4.4.1 Displacement Ventilation

In the store employing a displacement ventilation system, measurements were taken at various points between two facing chiller cabinets, as shown in the diagrams below. Of particular importance in this system are the high level (recirculated) inflows located 1.83m apart along the corridor of the store (although not directly above the centre of the corridor). These inflows (collectively known as the Reycol unit) blasted air down from the ceiling at high speeds; this air would then diffuse and mix at low level. Air was recirculated locally within the store to avoid the cold 'corridor effect'. See figures 17 and 18 below for plan and side views of the area studied.

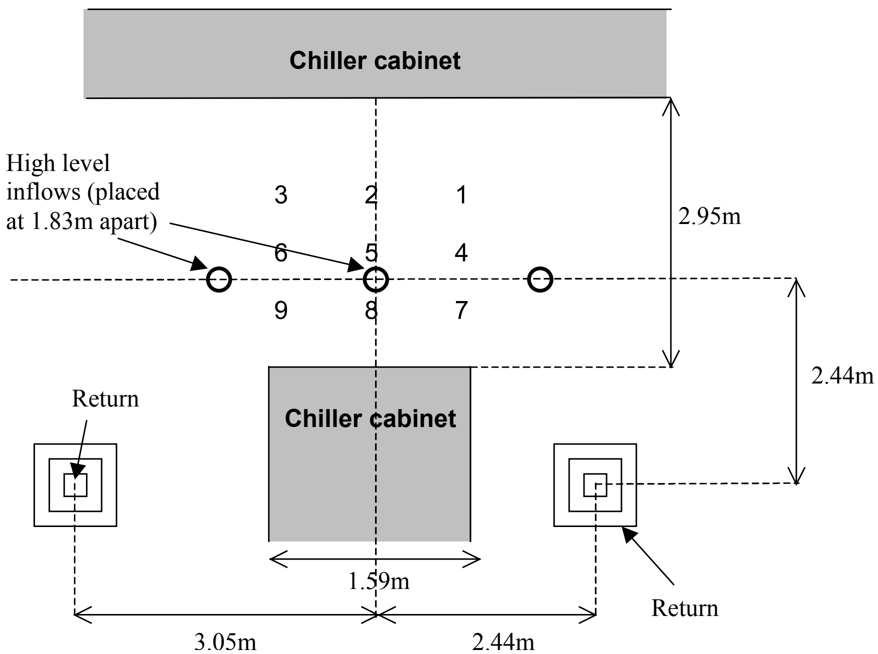


Figure 17: Plan view of cabinet layout and measuring locations.

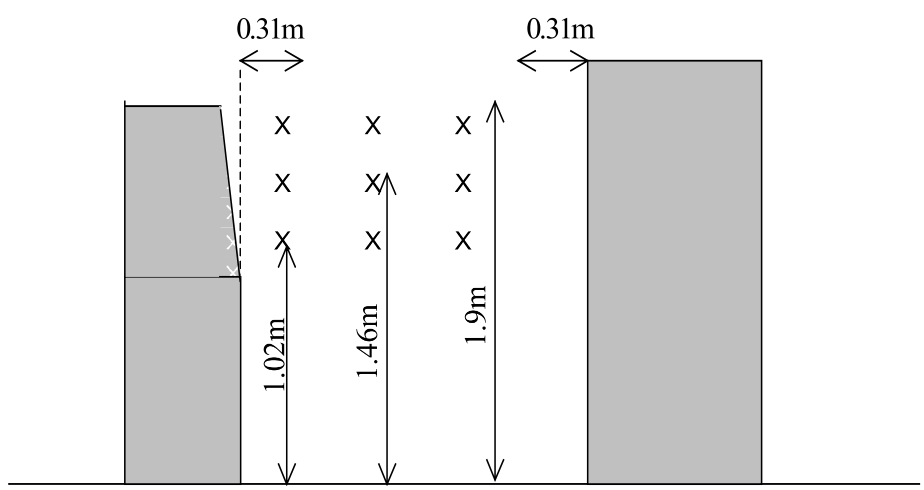


Figure 18: Side view of cabinet layout and measurement locations.

The results of the monitoring are show in the table below.

Position	Height								
	1.02 m			1.46 m			1.9 m		
	Temp (^{o}C)	Velocity (cms^{-1})	RH (%)	Temp (^{o}C)	Velocity (cms^{-1})	RH (%)	Temp (^{o}C)	Velocity (cms^{-1})	RH (%)
1	15.9	16.4	38	16.1	18.3	38	17.2	17.1	40
2	15.9	15.8	37	17.0	14.8	39	17.2	16.5	40
3	15.8	15.0	38	17.0	12.9	38	17.2	19.8	40
4	16.3	24.9	38	16.9	18.8	38	17.0	14.5	40
5	18.1	55.1	38	18.0	55.0	38	18.0	21.9	40
6	17.4	32.5	38	17.1	32.5	38	17.1	14.3	39
7	16.8	15.6	38	16.9	13.7	38	17.6	14.1	39
8	16.5	21.0	38	17.1	14.5	38	17.9	16.5	39
9	16.5	18.0	39	17.1	16.9	38	17.6	19.3	39
Mean values	**16.6**	**23.8**	**38.0**	**17.0**	**21.9**	**38.1**	**17.4**	**17.1**	**39.6**

Table 9 Results of air movement study for displacement ventilation system.

The mean temperatures at low, mid and high level can be seen in Table 9 (air speed recordings of less than 10cm/s were taken to be 10cm/s when calculating the mean values). Temperature and relative humidity conditions within the store appeared to be fairly consistent and not vary greatly with location. Air temperature at the 27 locations ranged from 15.9^{o}C to 18.1^{o}C – an overall variance of 2.2^{o}C. The highest measured temperature was to be found at a height of 1.02m at location 5 (see Figure 17), with locations 4, 6, 7, 8 and 9 also displaying slightly higher temperatures at low level. As the height increases the temperature range falls, suggesting that the larger low level variance is likely to be due to the introduction of slightly warmer air through the Reycol unit inflow.

There is little variance in temperatures between the three heights, possibly due to the high level inflow referred to earlier. The overall mean temperature found in the study area was 17.0^{o}C which does at first appear to be rather cold, although it should be remembered that these measurements were taken in late evening at the end of February (when outside temperature was likely to be fairly low).

The Reycol unit accounts for a large proportion of the variation of the air speeds found in the study area. At a height of 1.9m the unit appeared to pass air in between the measuring locations, as neither location 5 or 8 recorded significantly higher airflow measurements at a height of 1.9m. Lower down the air appears to diffuse and higher air speeds were recorded, in particular at location 5 (possibly due to the fact that the inflow unit was positioned angling in towards the centre of the corridor).

5.4.4.2 Standard High Level Ventilation Supply and Extract

The other monitored store employed a standard ventilation strategy. Air was introduced by a large, high level flow located over 2 metres away from the cabinets. Returns were located approximately 6.8 metres away from each other, with one return being directly above the corner of one of the cases. The measurements followed a similar protocol to that in the displacement ventilation store.

The results of the measurements are given below in Table 10.

Position	Height								
	1.02 m			1.41 m			1.8 m		
	Temp (^{o}C)	Velocity (cms^{-1})	RH (%)	Temp (^{o}C)	Velocity (cms^{-1})	RH (%)	Temp (^{o}C)	Velocity (cms^{-1})	RH (%)
1	19.5	13.7	42	21.3	15.2	40	22.8	10.2	40
2	20.5	15.5	42	21.2	<10	40	22.9	11.1	40
3	20.5	16.5	42	21.1	19.3	41	22.9	10.0	40
4	20.3	21.9	41	21.6	12.1	41	21.4	15.5	40
5	20.2	29.3	42	21.8	20.0	41	21.5	<10	40
6	20.5	16.7	43	20.8	19.6	41	21.7	11.8	40
7	19.9	19.5	42	20.8	15.9	41	20.6	13.0	40
8	20.1	22.4	42	20.5	25.0	41	20.8	17.4	40
9	20.3	23.1	43	20.5	20.0	41	20.9	9.6	40
Mean values	**20.2**	**19.8**	**42.1**	**21.1**	**17.5**	**40.8**	**21.7**	**12.1**	**40.0**

Table 10 Results of air movement study for standard ventilation system

Mean values of temperature, air speed and humidity at each of the three heights as shown in Table 10 show that conditions in the conventional store were reasonably constant with respect to location. Higher temperatures were found at the higher levels, as is to be expected, while air near the returns was slightly cooler than the air across the corridor (particularly at high level). Air speeds were higher at low level and at most locations around the return (although location 7, closest to the return, did not experience the higher levels of air speed that may be expected). Relative humidity was also consistent with location, increasing slightly with the cooler temperatures at low-level (as the two quantities are inversely related), although in general the ventilation system at the Warrington store appears to produce consistent in-store atmospheric conditions.

5.4.5 Environmental Test Chamber Experiments

It was intended that the chamber set-up would have the capacity to produce air flows around the cases very similar to those experienced in the two stores. Air was conditioned to the correct temperature and humidity in a separate sealed and insulated room before being passed into the main chamber through a network of pipes and ducting. Two fans were used for this; one forcing air into the main chamber, while another was connected to the return piping. A third fan was connected (as required) to give the high level inflows shown in Figure 19, and was placed inside the main chamber so that it could be used to recreate the Reycol unit as seen in the displacement ventilation store. Ducting was placed at the back of both the freezer and chiller units to attempt to return most of the heat produced by these units back to the conditioning room.

A series of experiments, with a half height glass door with open well freezer, and also an integral refrigeration cabinet, were performed in the chamber replicating the in-store conditions.

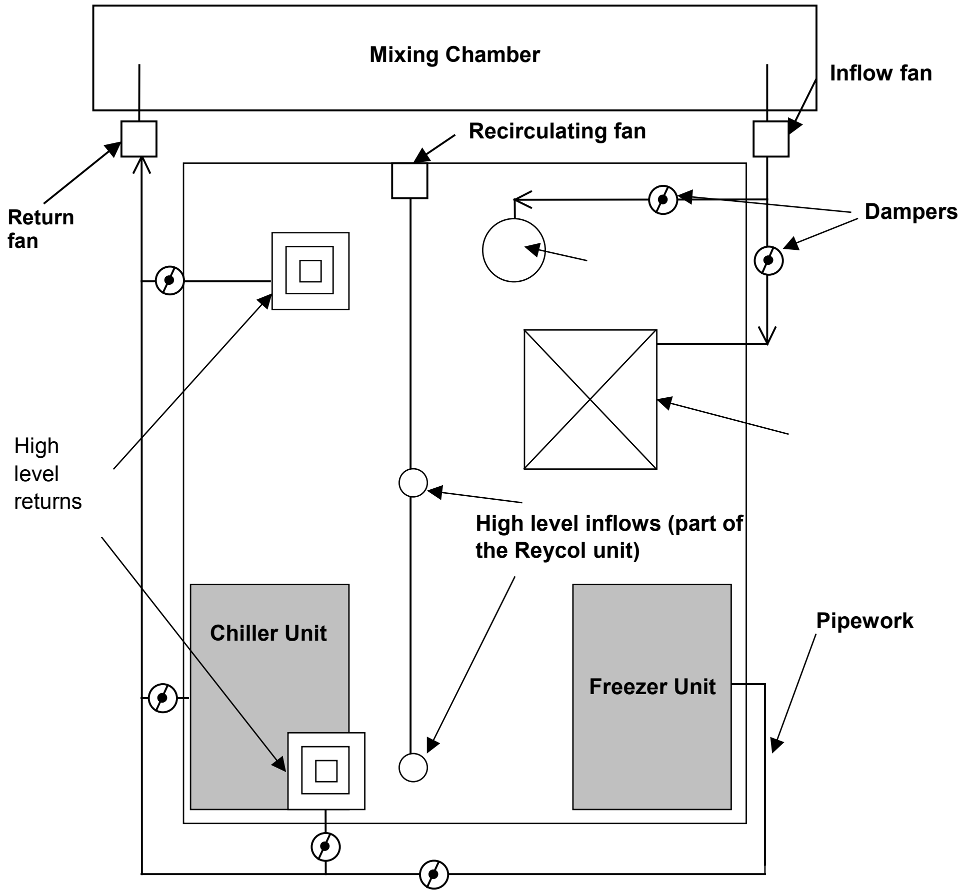


Figure 19 Schematic plan view of the environmental test chamber.

Three variable conditions were tested;

- Ventilation strategy – standard or displacement
- Store humidity and temperature
- Low level air extraction.

For each of these series of experiments the following aspects of performance were measured:

- the temperature distribution in the aisles
- the freezer power consumption
- chiller power consumption.

The layout enabled both standard and displacement ventilation conditions to be recreated, as well as allowing the balance between high and low level extraction to be varied. In order to ensure that conditions within the chamber were fairly similar to conditions within the store, airflow measurements were taken at 27 positions around the units, in a similar manner to those in-store measurements detailed above.

Room temperature and relative humidity were measured at a height of 1.5m in a position well away from either of the cases. In addition, in order to evaluate the effect of each ventilation strategy on customer comfort, three thermocouples were suspended in the ‘aisle’ mid-way between the two units at heights of 0.2m, 0.9m and 1.6m.

Readings of product temperature were taken from thermocouples placed inside the test ‘product’ loaded into the freezer, in order to ensure that the units were operating as would be expected in-store (i.e. keeping the product at the correct temperature). Both average power consumption and total energy consumption were recorded in order to allow for comparison between the various conditions.

Each test was run for a 24-hour period, during which a number of different values were logged at 2-minute intervals. This process was then repeated for a number of different atmospheric conditions, in order to test the effect of; ventilation strategy, relative humidity, and mixed level extraction.

5.4.6 Results

Ventilation Strategy
Under displacement conditions the 'corridor' temperatures were slightly cooler (temperatures at 1.6m were over 1 degree lower on average than under standard conditions). This is almost certainly due to the effect of the Reycol unit introducing slightly cooler air than was initially in the aisle. However, the corridor temperature was far more constant, reflecting more accurately the room temperature.

The ventilation strategy had little effect on the power consumption of the freezer, but for the chiller cabinet it appeared that ventilation strategy did have some effect on power consumption. At 19^{o}C chiller energy consumption with displacement ventilation was 5.4% higher than standard, while at 22^{o}C there was a 4% difference. Total energy consumption for the chiller cabinet was far higher at 22^{o}C than at 19^{o}C for both ventilation strategies.

Relative Humidity
The freezer unit did not perform differently under the different humidity levels and there was no visible trend between humidity and energy consumption. At room temperatures of 19^{o}C there was also little variation with relative humidity in the energy consumption of the chiller. At a temperature of 22^{o}C however, energy consumption ranged from 34.35 kWh at 35% humidity to 39.70 kWh at 50%, an increase of a little over 15%. One reason why humidity has little effect on chiller energy consumption at 19^{o}C but a larger impact at 22^{o}C may be the increased moisture found in the air at the higher temperatures.

Low Level Extraction
The effect of mixed level extraction on the performance of the chiller unit was measured only at 22^{o}C for both ventilation strategies. Measured temperature and humidity values were very similar for the four tests carried out, while product temperature was also constant for both the standard ventilation tests. The energy consumption of the chiller varied little between 100% and 40% low level extraction (by 0.1 kWh over 24 hours) and this variation can be attributed to experimental error.

Similar results were found for displacement ventilation, with the product temperature being very similar. The energy consumption of the chiller unit varied little between 40% and 20% low level extraction (by 0.3 kWh over 24 hours). Consequently, with the small differences measured, the test demonstrates that varying the amount of low level extraction has a minimal effect on power consumption.

Temperature
It was initially considered that temperature may have a significant effect on power consumption. From the measurements, the average total energy consumption of the freezer unit at 19^{o}C was 48.6 kWh, whereas at 22^{o}C average energy consumption stood at 49.6 kWh. An even larger increase in consumption with temperature was seen for the chiller unit where average consumption at 19^{o}C was 30.5kWh; this jumped to 37.6kWh for 22^{o}C (an increase of 23%). It is thought that temperature has a much larger effect on energy consumption than humidity or ventilation strategy.

Voltage

Some observations made during earlier tests showed that voltage supply may have an effect on cabinet efficiency. Tests were carried out supplying different voltages of 240V, 230V and 220V to the cases and monitored the effects of this. Testing was conducted at 50% humidity and standard ventilation, while atmospheric and ventilation conditions remained constant at 22°C. Large energy savings were seen as the voltage was dropped to first 230V (causing a 6% saving) and then to 220V (causing a further 5.4% saving). Dropping the voltage in this manner had no negative impact upon product temperature – in fact product temperature was lowest at 220V.

Door Usage

It was suggested that leaving the freezer doors open permanently by around 50mm may have some effect on energy consumption, particularly at higher humidities, and would be more representative of in-store use. In order to test this theory it was decided to run two tests with the freezer doors partially open, one at 35% relative humidity and one at 50%, and to compare the results with the two tests already conducted at these humidities. The results suggest that door opening of this nature has little effect on energy performance.

Effect of the Reycol Unit

Another suggestion was that whether the Reycol unit was on or off may have a significant effect on the energy consumption of the units under displacement ventilation conditions.

It was decided to test this by running a test at 50% humidity and 22°C room temperature with the Reycol unit off to see the effect. This test was compared to an earlier test under the same atmospheric conditions. It was found that the Reycol unit kept corridor temperatures cooler at the higher levels, yet at low-level corridor temperatures were warmer. The room temperature well away from the cases was also slightly higher with the Reycol unit on.

Contradictory to the freezer performance, it was found that total energy consumption of the chiller was much higher with the Reycol unit switched on, and substantial savings of 10.9% were seen when it was switched off. The product was also kept much cooler when the Reycol was off, possibly due to the lack of warm air being forced into the case by the Reycol inflows.

5.4.7 Conclusions

Both temperature and air movement can have a significant impact on cabinet energy use. The store conditions have a much greater impact on the chiller than the refrigerated cabinet because of the lack of isolation of the internal refrigerated space from the store environment. Increasing the store temperature caused in excess of a 20% increase in energy use. There is obviously a need to select the heating and ventilation schemes to suit the types of display cabinet.

The lack of impact of humidity in these studies is probably as a result of the dry bulb air temperature being lower than those research results that show a significant impact of %rh: this lower range of %rh being more representative of the UK situation. It is also possible that with fixed defrost cycles the energy use is the same although the frost build up may be more closely related to the humidity.

6 CONCLUSIONS


6 Conclusions

6.1 RESEARCH SUMMARY

The research carried out by the UK has identified areas where significant energy savings can be made in supermarket refrigeration. Some of these technologies are being adopted but a number of issues remain to be addressed in order to achieve these potential savings in all applications.

Simulations to assess the feasibility of a wide range of **secondary systems** and refrigerants in a typical supermarket found that the energy use of the secondary systems simulated would typically be greater than for other systems and as a consequence they would deliver higher operational CO_2 emissions. In terms of overall TEWI, the secondary system would be better than an R22 direct system with 15% annual refrigerant leakage rate over a 15 year plant life. The lower TEWI of a secondary system is lessened as the leakage rate of the equivalent direct system decreases. However, there are some anecdotal reports that refrigerant leakage rates in DX systems in practice may be much higher than 15%. In order to more clearly establish existing refrigerant leakage rates in DX systems, specific research should be undertaken.

The **energy used for defrosting** evaporator coils in supermarkets can amount to 30% of the operating energy. Different means of reducing the defrost energy have been researched by laboratory studies of defrosting techniques. For low temperature applications, the use of gas defrosting is more energy efficient but the considerable extra capital costs lead to long payback periods. Payback periods for medium temperature systems are of the order of 9 years, but for low temperature applications this can be up to 28 years. Consequently, electric defrost is most commonly used for low temperature applications in the UK.

Optimum control of defrosting can also lead to energy savings. Optimising the defrost cycle and off time would save energy in most cases, but if a reliable means of initiating defrost on demand could be developed this would be the most energy efficient strategy.

Combined heat and power and combined cooling, heat and power are potentially efficient methods of providing the energy for supermarkets. A simulation model has been developed that allows the energy and heat flows in a supermarket to be modelled and will predict energy and cost savings arising from a range of measures taken to improve efficiency of supply or use of energy. The model was used to evaluate a range of options that showed the cost effectiveness of CHP/CCHP is sensitive to the fuel costs, and in particular the ratio of the gas and electricity costs. Additionally the number of hours for which the plant operates in a year is a key determinant of the cost effectiveness.

The **internal environment of the store**, in which the refrigerated display cabinet is located, can have a significant bearing on the energy use and performance. The temperature and humidity of the air and its movement around the cabinet are all likely to influence the energy use.

For the freezer tested, varying either of the store temperature, relative humidity and ventilation mode and rate had little impact on the energy consumption of the unit. For the chilled cabinet the energy consumption was more dependent on the store conditions. A change of store temperature from 19 to 22^{o}C increased the energy use by up to 20%. At the higher temperature of 22^{o}C, the relative humidity of the store environment increased the energy consumption by 15%, when raised from 35%rh to 50% rh. Increased ventilation around the chiller could also lead to some increase in energy use.

7 OVERVIEW

7 Overview

The UK participation in Annex 26 has proved most valuable in promoting the concepts involved in advanced refrigeration techniques for supermarkets. The Phase 1 review of the state-of-the-art, across a wide range of topics, led to four specific issues being identified for further study. These were addressed in Phase 2 phase of the programme and produced a wealth of information.

Each of the individual technologies has much to offer in terms of reduced environmental impact, either by increased efficiency or reduced refrigerant use. The scope for reducing the TEWI of supermarket refrigeration is significant by considering these measures alone. A key finding, when viewing the complete programme of UK work, is the interdependence of much of the technology that is currently being used to provide refrigerated displays in supermarkets.

There are two parallel, and related issues, which need to be considered at all times: the use of refrigerants, and the energy efficiency of all the systems. These projects have confirmed that many of the practices the retailers currently have in place are the most cost effective e.g. the use of DX systems at low leakage rates, and electric and off-cycle defrosting. However, some of the more expensive and/or innovative technologies could further reduce the TEWI of supermarket refrigeration.

What is required, therefore, is an integrated view of the needs of the supermarket. Some of these issues are discussed below to give an indication of the factors that interact and how they may be addressed in developing a design solution.

7.1 STORE LAYOUT AND ENVIRONMENT

Adopting this integrated approach needs an holistic view of the store and its refrigeration systems. Starting with the display cabinets the layout of the store can have a significant impact on the potential for adopting certain types of system. For example, dispersing cabinets throughout the whole of the store may make secondary systems too expensive, and also place a high heating load on the HVAC system. Confining the refrigerated equipment to certain regions of the store may help with heat recovery and ventilation strategy, it may even allow for local dehumidification. Although the impact on in-store temperatures and customer comfort should also be taken into account.

Open front chillers are more sensitive to air movement patterns, and the use of Reycol units with these may increase energy use. However, customer access to the product and their thermal comfort is naturally paramount from the supermarket owners’ perspective. Paying attention to the store environment, and careful selection of equipment and the ventilation strategy, will enable occupant comfort to be optimised whilst still maintaining energy efficiency. This integrated approach to design may even provide increased occupant comfort.

7.2 ENERGY SUPPLY SYSTEMS

From the perspective of energy use, the starting point of an integrated approach would be an analysis of the potential for CHP or CCHP. This technology offers potentially large savings in carbon dioxide emissions if the heat to power ratio is advantageous. Supermarkets, which typically have a high refrigeration load, offer the potential to change high electrical demand for the compressors into a heat demand for absorption chillers. This can increase the heat to power ratio, particularly in the summer months when the space heating demand is low, and therefore increase the attractiveness of a CHP. Adopting CCHP can also reduce overall TEWI because of the reduced need for environmentally damaging refrigerants. In the UK at present the ratio of gas to electricity prices is a major factor in the limited take-up of CHP in supermarkets.

7.3 DISPLAY CABINET DEFROST

Reducing energy use for defrost is achievable but there are a considerable number of issues that need to be borne in mind when selecting a solution. The current approach of fixed cycle defrost appears to be reliable but it is not the optimum method. Each defrost technique has its advantages and disadvantages and differing capital costs and energy use. Selecting the correct type of defrost control system can give great energy savings and there are a wide range of approaches. A clear favourite is 'defrost on demand' but there may be problems of simultaneous high defrost demand substantially increasing peak electricity loads. Reducing store humidity can reduce defrost energy, and where possible this should be investigated. As yet the trade-off between defrost energy and HVAC energy is yet to be determined.

7.4 SECONDARY SYSTEMS AND LEGISLATION

Secondary systems, as analysed here, are at an early stage of development in the UK. This therefore makes the technology appear less attractive as the initial teething troubles are developed out of the systems. For example, store design and layout made specifically for secondary systems could well reduce some of the higher capital costs and also reduce operating costs, thereby increasing the cost effectiveness.

The refrigeration sector has been subject to many legislative changes in recent years. These changes have greatly affected the industry by increasing the importance of selecting the correct refrigerant and avoiding losses from the plant. It is possible that further restrictions will be made to the types of refrigerants that are allowed. If this happens then a number of issues that have been addressed by this study will need to be reviewed. In particular, the role of secondary systems will change and they may become the preferred method of providing environmentally safe refrigeration. This would doubtless also change the economic situation of CCHP.

7.5 VARIABLE SPEED COMPRESSORS

Phase 1 of the project showed that compressor technology and the variable speed operation of compressors can have significant energy benefits. This is a further area of interest and will add another technological solution to the high energy use for refrigerated displays.

IEA Annex 26:

Advanced Supermarket Refrigeration and HVAC Systems

UK Report Phase 1 Individual Technology Reviews

February 2003

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Report A

Centralised, Distributed and Integral Systems - EA Technology

Short description of the topic/technology status - proven/developing

The traditional design of refrigeration system incorporated a central refrigeration system, one or more HT and one LT, which supplied refrigerant to each cabinet from headers in the plant room. These are referred to as centralised systems.

This system was convenient for large compressors, such as screw compressors, but led to high refrigeration leakage. With the advent of more efficient small compressors, such as vertical scrolls, other options become available.

Such compressors can be used in packs mounted outside the store but as close as possible to the cabinets which they are supplying with refrigerant via stub and duct pipework.

They may also be sited within the store rejecting their heat from one or more cabinets to either air within the store, or to a water loop which then passes to outside the store and rejects the heat.

If the system is self-contained so that the cabinet has a dedicated compressor and an air or water-cooled heat exchanger, it is referred to as an integral unit. These have been around for many years for smaller units, such as chest freezers, ice cream displays etc., using hermetic rotary or reciprocating compressors. The introduction of the horizontal scroll compressor has brought this option to larger cabinets, traditionally used in supermarkets.

An additional complication is the simultaneous introduction of hydrocarbon and ammonia based refrigerants (see paper on refrigerants), which have led to a switch to secondary refrigerants. These may be either on the cooling side passing cold glycol rather than refrigerant to the cabinets or on the heat rejection side using water as the cooling medium.

The heat removed from the refrigeration cabinets significantly cool the air within the store. In central and distributed refrigeration this heat is normally rejected outside the store (although it can be used to provide heat back into the store, but this is not usually energy efficient as it requires a high head pressure for much of the year). If integral cabinets rejecting heat into the store are used then this will require careful integration with the building services to avoid the need for significantly increased mechanical cooling to maintain comfortable space temperatures.

It was proposed by EA Technology that alternative designs could provide combined space heating and cooling and refrigeration, using a brine loop. Two systems were identified as being optimal from a TEWI viewpoint: a system which circulated brine at a temperature to cool the HT cabinets directly, with LT provided by either integral or shop floor distributed refrigeration rejecting heat to the low temperature loop, with the space heating and cooling met separately, or a water loop is operated to meet the requirements of the space cooling and the cabinets reject their heat into the loop and heat pumps extracting heat from the loop are used to meet any space heating requirements. From a capital cost viewpoint the second system is likely to be too expensive, but there are a large number of other designs which may prove attractive.

Applications - proven/potential

Central systems
Ammonia systems need to be sited in a plant room due to the toxic nature of the vapour. They are therefore likely to be used in centralised systems. Marks and Spencer installed an ammonia system at their Handforth Dean store. Four air-cooled ammonia chillers cool a propylene glycol brine to around -9°C, this is circulated around the store. This directly cools

the HT cabinets, whilst the LT cabinets are cooled by small packaged HFC404a scroll compressor with heat rejected into the glycol loop. Measurements indicated that this had a similar energy consumption to a standard R404a distributed system.

Distributed systems
The switch from centralised to distributed pack systems has concentrated on the Copeland Scroll compressor range. These allow good efficiency for HT and reasonable efficiency for LT systems in a single stage of compression, with increased efficiency via an intermediate compression port also being offered on the LT system.
The use of multiple compressors allows good matching of load without loss of efficiency. Such systems are offered by all the contractors to the supermarket industry.

Tesco constructed a store at Enfield in 1998 using hydrocarbon refrigerant HC1270 (Propene) distributing cooling using the proprietary Tyfoxit brine (this has the advantage of being low viscosity even at temperatures as low as -30°C).

Integral systems
Small integral cabinets have been used for many years for specialist stand-alone display in supermarkets. Frozen food stores have always used integrals rather than centralised or distributed refrigeration. However as they use chest freezers their refrigeration load is lower than a supermarket.

Hussmann have recently installed integral systems based on the horizontal scroll compressor in both an ASDA and a Tesco supermarket but results on performance are not yet available.

Cost - current/mature market

The cost of integral cabinets and their refrigeration system is similar to a centralised plant. However there are other significant cost advantages. The on site construction time can save up to eight weeks in store construction and this is extremely valuable.

The cost of a secondary rather than a DX system is about 10% higher at present. However once the market is more mature and the use of pre-insulated plastic pipework is accepted the cost of the secondary system are likely to be around the same to slightly lower cost than the DX system.

Completely centralised plant is much more expensive than distributed plant and is not widely installed, except when the use of ammonia is required for environmental reasons.

Performance

The performance of integral units is likely to be similar to a distributed system as the slightly poorer performance due to the wider evaporating pressure swing will be compensated by the lower suction line pressure loss. What is not yet clear is any secondary effects on the stores HVAC system.

Maintenance

Maintenance, particularly leakage control was a major issue for centralised refrigeration systems. This was the main reason for the move to distributed refrigeration with stub and duct pipework layout. Using secondary systems should have little effect on the maintenance requirement as similar lengths of pipework are use.

Integral units are often modular: smaller integral units are removed for maintenance and larger units can have the compressor/condenser replaced on site with these components removed for any maintenance. There is insufficient data to know if additional maintenance requirements will emerge, but it is expected that the maintenance cost of integral cabinets will be lower.

Environmental and safety issues

The environmental and safety issues are described in the Refrigerants paper accompanying this paper.

Existing analysis tools

Most contractors and consultants have proprietary software, usually in the form of spreadsheets, for calculating the energy requirements of different refrigeration systems. The main difficulty, where more development is needed is in the understanding of the effect of different type of heat rejection system (internal or external to the store) on the energy consumption of the HVAC system.

References

RAC magazine feature
IEA Annex 22 Final Report - Guidelines for Design and Operation of Compression Heat Pumps, Air Conditioners and Refrigerating Systems with Natural Refrigerating Systems with Natural Working

Report B

Optimisation of Secondary Systems - EA Technology

Short description of the topic/technology status - proven/developing

The requirements to reduce refrigerant leakage and to use hydrocarbon and ammonia as refrigerants due to their negligible global warming potential (GWP) has led to a rise in the use of secondary refrigerants.

The report on refrigerant selection deals with the safety implications forcing the use of secondary systems, this paper will cover their design and optimisation.

Applications - proven/potential

Secondary systems carry the coolth from the refrigeration system to the cabinet or cold room requiring the cooling. This transport can either be by a liquid, such a glycol, silicon oil or Tyfoxit or by a liquid which changes phase in the cabinet and condenses in the refrigeration system evaporator, notably CO2.

If a large compressor is used with limited controllability, beyond on/off, then there is an advantage to increasing the time constant of the system. This can be done by requiring a solid/liquid phase change within the circulating secondary refrigerant loop. this is also important in preventing a large pressure rise in CO2 secondary refrigerant.

Such a system may produce binary ice, or may heat and cool a phase change material held within a plastic sphere.

Marks and Spencer installed an ammonia system at their Handforth Dean store. Four air-cooled ammonia chillers cool a propylene glycol brine to around -9°C, this is circulated around the store. This directly cools the HT cabinets, whilst the LT cabinets are cooled by small packaged HFC404a scroll compressor with heat rejected into the glycol loop.

Tesco constructed a store at Enfield in 1998 using hydrocarbon refrigerant HC1270 (Propene) distributing cooling using the proprietary Tyfoxit brine (this has the advantage of being low viscosity even at temperatures as low as -30°C).

The OBS supermarket in Norway has a 35kW ammonia chiller producing binary ice which is circulated to the HT display cabinets.

The ICA Focus supermarket in Sweden has 165kW HT and 38kW LT cooling using ammonia as the refrigerant with propylene glycol circulating to the HT cabinets and cooling the condenser of the LT ammonia system which cools CO2 circulating to the LT cabinets. A 200L propylene glycol ice storage tank is installed in the CO2 line to condense it when the refrigeration system is off to prevent the CO2 pressure rising above 23 bar(-15C) when it has to be vented for safety reasons. This design has been replicated in other supermarkets in Sweden.

Sainsbury's have recently opened their Millennium Store using hydrocarbon refrigerants with heat rejected to borehole water, giving an excellent level of performance.

Overseas the IEA Annex lists systems in Sweden and Germany, both using HC1270 as the refrigerant. The Swedish system uses Propylene glycol and CO2 as the secondary refrigerants whilst the German system uses Tyfoxit.

Limitations

The main limitation on he increased use of Ammonia systems is the limited expertise remaining in the UK on the maintenance of such systems. (Forbes Pearson, Star Refrigeration).

Cost - current/mature market

The cost of a Hydrocarbon system with a secondary refrigerant is approximately 10% higher than the equivalent DX system. In the future this price is likely to reduce with the greater use of pre-insulated plastic pipework and cost reduction as experience grows. It is expected that the system cost will soon be equal to or slightly (5%) less than a DX system.

The cost of Ammonia systems is likely to be higher than Hydrocarbons. If Binary ice is used to stabilise the pressure then this will significantly increase the cost (2).

Performance

The performance of secondary systems compared to DX systems is difficult to generalise. The following factors must be considered:

Temperature lift across the heat exchangers: any increase in temperature lift will increase the work required from the system. The high efficiency of hydrocarbons and ammonia do counteract this and in many cases the difference in compressor power is very small. However the temperature lifts should be kept as small as possible and approach temperatures of 5K are often used, with a similar temperature change through the heat exchanger.

The other major power input is pump power to circulate the brine to the cabinets. This is not too serious a problem for HT with the viscosity of most brines being reasonable at -10C. However at LT the viscosity is high for most brines and the pump power is very significant. This is the reason for using CO_2, which due to its phase change requires much smaller pipes (capable of withstanding high pressures) and has a much lower pressure drop.

Existing analysis tools

Most contractors and consultants have proprietary software (usually spreadsheets) to calculate the likely performance of different systems. These do need care in use to take into account factors such as suction gas superheat and heat gain into the secondary refrigerant pipework.

Standard mechanical engineering pipework pressure loss calculations can be used to estimate the pump power requirements.

References

1 IEA Annex 22 Final Report - Guidelines for Design and Operation of Compression Heat Pumps, Air Conditioners and Refrigerating Systems with Natural Refrigerating Systems with Natural Working Fluids -December 1998. SINTEF report No: TRA4868

2 Substitutes for green house gases Per Henrik Pedersen DTI Denmark at www.mem.dk/publikationer/drivhus/green.htm

Report C

A Report on the Application of Combined Heat & Power (CHP) & Combined Cooling Heat & Power (CCHP) Schemes in the Supermarket – School of Engineering Systems & Design, South Bank University

Executive summary

Current Combined Heat & Power (CHP) Technology as Normally Applied

- CHP can reduce revenue costs and CO_2 emissions by up to 40%
- CHP is seen by the government as major component in reducing CO2 emissions
- CHP Schemes will not be subject to the climate change levy

Combined Heat & Power (CHP) in the Supermarket - Leading Practice

- CHP has been successfully applied in 2 supermarkets in the UK
- The low gas tariff that maybe negotiated is 0.55p/kWh realises large revenue cost savings.
- Payback periods of 4 years have been reported
- Payback can be improved significantly if demand for heat in summer is provided

Combined Cooling, Heat & Power (CCHP) in the Supermarket - Leading Practice

- Integrating a ‘heat powered cooling system’ (CCHP) provides a demand for summer heat
- Conventional commercially available technology may be used
- Primary energy / CO_2 emissions may be reduced by approximately 25%
- Much greater revenues cost saving would be achieved.
- CCHP has been installed in other markets

Tools for Analysis & Opportunities for Further Work

- The potential environmental and cost benefits of this technology are very large
- The cost of CCHP requires evaluation
- A small spreadsheet study would quantify the benefits.

Current CHP technology as normally applied

In recent years, it has become standard practice to consider the use of Combined Heat and Power (CHP) schemes in commercial applications. With the Kyoto agreement, there has been greater emphasis on reduced energy use and energy efficient systems such as CHP have been considered for new applications.

CHP schemes are often a more efficient means of power generation than conventional power stations as they produce electricity locally and minimise the distribution losses. However, they also allow heat output from the generation plant to be used for space or process heating. In applications where there is a combined heating and cooling requirement a very efficient means of energy generation is produced compare to the conventional method of providing heating and electricity. This is shown in Figures 1a and 1b, which shows the primary energy consumed by a conventional scheme compared to the CHP scheme to satisfy the same heating and electricity demand [Eatsop & Croft, 1990]. From this it may be seen that the CHP system uses nearly 40% less primary energy than the conventional scheme. In addition to the revenue cost savings associated with CHP, the lower energy consumption also produces environmental benefits since non-renewable energy reserves are preserved and environmental pollution is reduced. As a result a CHP scheme will be exempted from the Climate Change levy, when it is enacted.

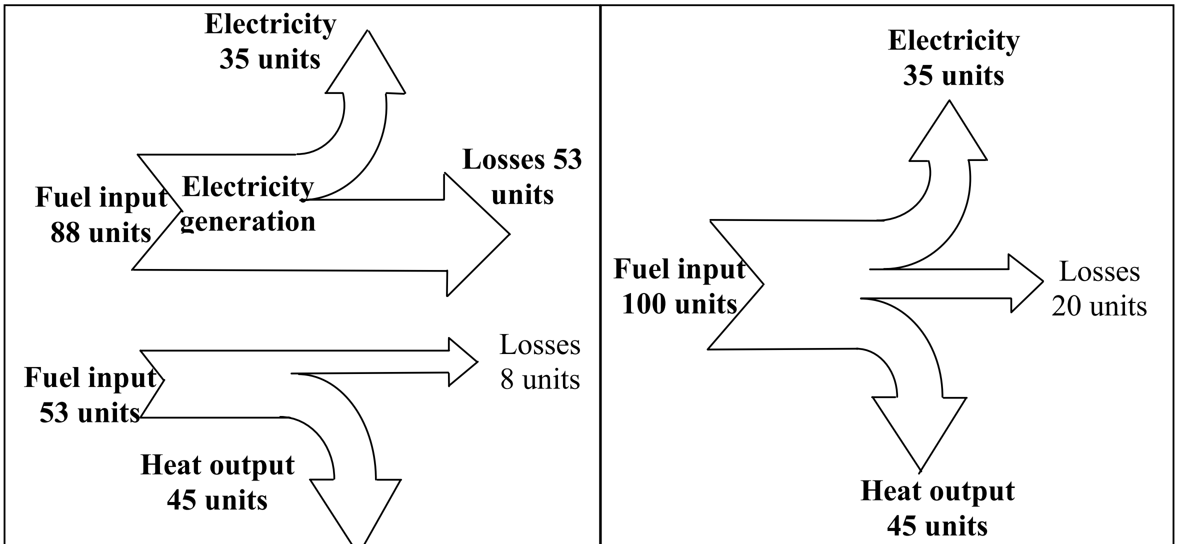

FIG 1A CONVENTIONAL HEAT AND POWER APPLICATION

FIG 1B COMBINED HEAT AND POWER APPLICATION

The equipment required for a CHP package is shown in Figure 2 and can be seen to comprise generally of three components:-

a) Prime Mover
b) Generator
c) Heat exchangers

The types of prime mover used include gas / steam turbines and reciprocating gas engines. The main differences between these are size/ capacity, heat to power ratio, efficiency and maintenance required. The generator produces electricity in a similar manner to that supplied from the national grid. Heat exchangers recover heat from the heat rejected during generation from the engine jacket, exhaust gas and water temperature of between 75 and 140 degree C can be produced.

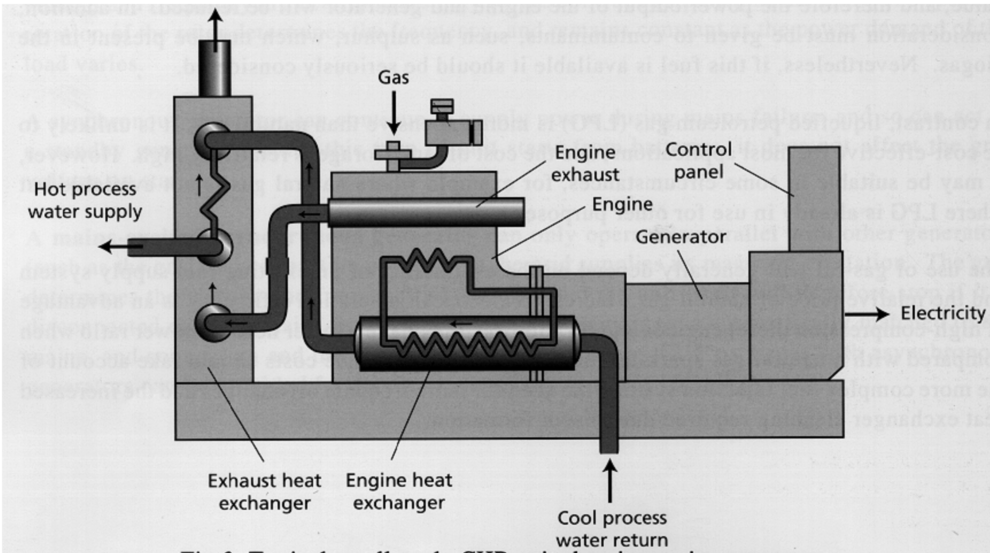

Fig 2 Typical small-scale CHP unit showing main components

CHP schemes have been applied for over 100 years and have been used to deliver electricity from between 15kWe and 100MWe. In most applications the main factor that determines economic viability is a high utilisation of the available heat and power. Most literature indicates that the CHP plant needs to be fully utilised for a period of atleast 4,500 hours per annum to be viable (< 4/5 year payback period) [ETSU, 1995]. With higher utilisation payback periods of less than 1 year have been reported [Tozer, 1999]. The main difficulty in achieving high utilisation is providing a heat demand during summer months.

Combined Heat & Power (CHP) in the supermarket - leading practice

CHP has been reported to have been applied in two supermarket applications in the UK. Sainsbury's used CHP at their Greenwich store to produce electricity locally on site and provide hot water for heating and toilet/ canteen facilities [Bunn, 1999]. Safeway employed an "Air CHP" package at their Milton Keynes store [David, 1996]. This used heat generated by the engine to heat air directly within an air handling unit. The advantage of this scheme compared to conventional CHP was reported to be a 5% increase in generation efficiency [Marriott, 2000].

To achieve high utilisation this scheme was operated continually throughout the year and this was achieved during period of low heat demand by rejecting excess heat to atmosphere. Adopting this control scheme enabled the CHP unit to offer a payback period of 4.5 years and this is consistent with theoretical viability studies reported by Maidment et al [1999a]. Calculations showed that in order to achieve this payback period significant heat rejection was necessary and this is demonstrated in Figure 3. It can be seen that under winter conditions some heat rejection was required, however, at ambient temperatures above 16^{o}C nearly all of the heat generated was rejected. As a result the CO_2 emissions /primary energy used by this scheme is not significantly different to that used by a conventional supermarket, even though the energy cost was significantly lower.

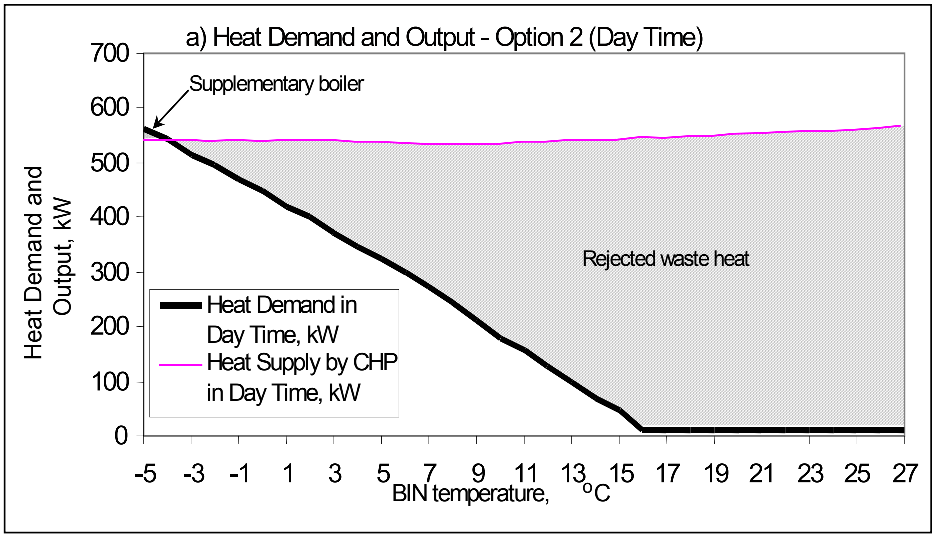

Figure 3 Graph of CHP heat output and demand vs ambient temperature for a typical supermarket

Combined Cooling, Heat & Power (CCHP) in the supermarket - leading practice

A summer demand for the waste generated by a CHP package can be provided if the system is integrated with an absorption cooling system. Heat from the CHP unit is used to power the absorption chiller which can provide cooling for the chilled food cabinets as well as the building itself.

Combined Cooling, Heat and Power (CCHP) schemes have been used in many applications including dairies, food processing, cold storage and pharmaceutical facilities, where it has been shown to produce large energy and revenue cost saving. Payback periods of <2 years have been proposed. CCHP schemes have not been applied in supermarkets, however, an initial desk study has shown that nearly 100% utilisation of the heat produced maybe achieved and this would present CO_2 emission /primary energy savings of approximately 20-25% compared with conventional schemes [Maidment et al, 1999b]. Further advantages of a CCHP scheme would be a very large reduction in energy cost (a continuous gas tariff maybe negotiated at 0.55p/kWh), the provision of standby generation plant and the scheme would also qualify the supermarket exemption from the Climate Change Levy.

A viability study has not been reported for the CCHP system in the supermarket. A number of CCHP schemes have been proposed to satisfy the supermarket cooling, heat and power requirement and a typical arrangement is showing in Figure 4. The CCHP alternatives will depend upon the type of absorption chiller and CHP engine used. The variants in absorption cooling system depend on type of refrigerant used, the number of cycle stage/ effects heat exchanger configuration, and the hot water temperatures required. The engine selected would need to be compatible with absorption system and an eutectic thermal may be required to ensure the cooling / heating / power outputs are coincidental.

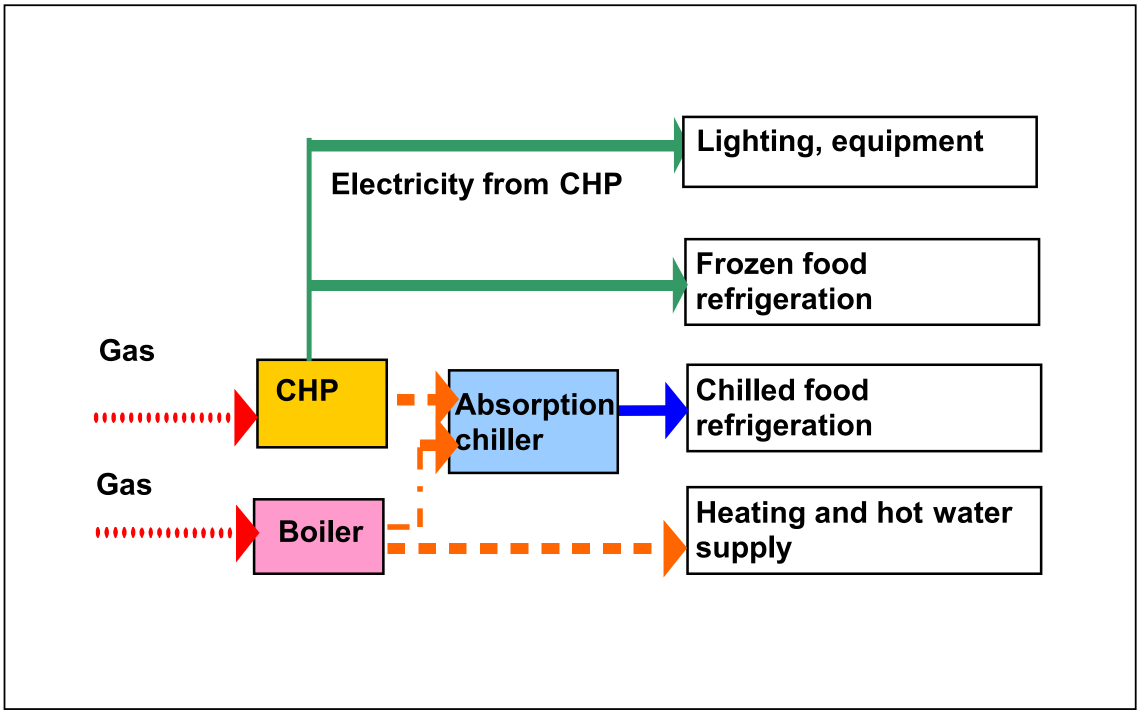

Figure 4 - One of the alternative CCHP schemes

Tools for analysis & opportunities for further work

A CCHP scheme using commercial equipment could be used in a supermarket to provide cooling, heating and power. The potential environmental and cost benefits of this technology are very large. A small spreadsheet study could quantify the benefits and advise on the preferred CCHP scheme that could be used in a supermarket.

This could be achieved using a purpose-developed spreadsheet that has previously been used to calculate the viability of CHP in supermarkets. With minor development the model could be used to quantify the viability of CCHP schemes.

Emerging technology

Conventional technology is commercially available for a CCHP scheme. Over the next 5-10 years emerging technology that may offer minor benefits are:-

1) Gas turbines - Several large companies including Volvo are developing turbines for mass market between 15 and 500 kWe output. As well as lower capital cost through economies of scale, these turbines offer reduced maintenance. [Langan, et al, 1999]

2)Fuel cells are receiving significant development and could be used in place of the engine in a CHP package. The advantage of fuel cells over a conventional engine is that they offer a 10% increase in electrical generation efficiency and they produce virtually zero pollution. [Kaarsberg, et al, 1999]

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Report D

Gas Engine Driven RAC Equipment - Calor Gas

Description of technology

Essentially the main difference between gas engine driven equipment is that the hermetic or semi-hermetic compressor is replaced with an open compressor, which is driven by the shaft of a gas engine. Either mains gas or LPG may be used.

Engine driven systems in general are commonplace in mobile RAC applications such as transport refrigeration and car air conditioning. It is therefore a broadly accepted technology.

Status

A proportion (5-10%) of residential and commercial split air conditioning in Japan uses gas engines for the main purpose of reducing peak electricity demand. Also a number of global companies have gas engine chillers available. In addition, a large number of companies manufacture gas engines and similarly, numerous companies manufacture open-type compressors.

Applications

Wherever a compressor is used, the motor can be replaced by a gas engine. Technology is well proven but is limited in use.

Limitations

Engines can only be positioned where the combustion products can be vented to atmosphere. Cost and equivalent CO_2 emissions must be equivalent of less than those associated with electric motor. The efficiency of the engine is generally the determining factor.
Capital costs will be greater and restrictive any many situations.

Cost

Since an engine substitutes the compressor motor, this technology will always be comparatively higher in terms of capital cost. This is generally in the order of one-quarter to one-half of the total unit cost (when considering split type HPAC). However, the issue of services supply may also be of consequence. Where a three-phase electrical supply is required, the avoidance of this may impact positively against the additional cost of the gas engine, although the gas supply must also be considered.

Taking the efficiency of the engines it can be seen that a basic running cost advantage is obtained where the cost of electricity is more than 2½ to 3 times the cost of gas (per kJ). This will be specific to location, daily and weekly operation times, and whether these services are used already.

Performance

COP in the case of gas engine systems takes into account the losses from the gas engine. The COP of the gas engine system (COP_{GES}) is the product of the refrigeration system COP and the engine efficiency (η_{GE}).

$$COP_{GES} = COP \cdot \eta_{GE}$$

Overall COP ranges from about 1 to 2.5, although typical values are around 1.3 to 1.5. Performance under part-load can be comparatively efficient since capacity control can be achieved through compressor speed variation.

Maintenance

Planned maintenance is necessary for engine systems, particularly for oil changes. The general problem of servicing these types of engines exists because conventional refrigeration engineers do not normally have this type of expertise. Obviously training is necessary.

Although noise levels would be perceived to be higher, data from existing Japanese equipment has indicated the levels to be comparable to conventional electric systems.

Environmental Issues

In comparison with electric compressors, the resultant equivalent CO_2 emissions of gas engine driven compressor are very much dependent upon the efficiency of the motor itself. This varies from nearly 30% for smaller engines and up to 40% for large engines. With respect to UK electricity generation emissions, the steady/full-load indirect emissions can be equivalent, or several percent less for higher efficiency engines.

The advantages of gas (natural gas or LPG) over conventional fuels are the level of emissions. CO_2, sulphur, NO_x and particulate emissions are significantly reduced and in some cases eliminated. However, combustion emissions do need to be deal with by release to the atmosphere.

Since an open-type compressor needs to be used, refrigerant leakage can be higher than in hermetic systems, which can obviously pose environmental, or safety problems.

Existing analysis tools

Not ascertained to date.

Report E

A Review Of Air Cycle Systems For Advanced Retail Refrigeration And HVAC Systems - Food Refrigeration and Process Engineering Research Centre, University of Bristol

Introduction

The growing understanding of the environmental impact of conventional refrigerants and of the limitations of vapour compression systems has focused attention on alternative refrigeration cycles.

Chlorofluorocarbon (CFC) and Hydrochlorofluorocarbon (HCFC) refrigerants have both ozone depletion and global warming potentials. The use of CFC refrigerants, such as R11 and R12 in vapour compression systems for topping up or refilling systems is banned from 1 January 2001. HCFC refrigerants, such as R22, are banned for use in new systems greater than 100 kW duty from 2001, with a ban for "virgin" HCFCs for topping up and refilling existing refrigerants from 2010 (an amendment may bring this forward to 2005/2008), (Butler, 2000). Hydrofluorocarbon (HFC) refrigerants, such as R134a and R404a, are ozone friendly but have high Global Warming Potentials ; 1300 and 3750 respectively (Carrier, 1999). The draft UK programme on Climate change issued by the Deputy Prime Minister on 9 March 2000 stated that there is a clear signal to industry that HFCs are not a sustainable technology in the long term, and they should only be used where there are no safe, cost-effective, practical and environmentally acceptable alternatives.

The alternatives are the natural refrigerants. Some of these have issues over toxicity and flammability, such as hydrocarbons, and ammonia. This leaves air, carbon dioxide and water. Using air as the working fluid has the advantage that it is free, environmentally benign and poses no health and safety risks.

Air cycle refrigeration has been widely used in the past (Cooper, 1989). The attraction was that it is simple and reliable compared with early vapour compression machinery, particularly on ships, where the escape of toxic and inflammable refrigerants such as ethyl ether, ammonia and sulphur dioxide was unacceptable. Gigiel *et al.* (1995) states the most suitable applications are those which can take advantage of the special characteristics of air cycle, such as a much wider range of operating temperatures, controlled air humidity and excellent heat recovery potential. The author identifies potential applications as replacing liquid nitrogen systems for food freezing, transport container refrigeration, fabric drying, building air conditioning and supermarket refrigerated retail display cabinets. For heating and ventilating systems air cycle could result in significant reductions in the overall energy consumption by providing simultaneous heating, cooling and power generation. In supermarket cabinets a centralised supply of compressed air, piped to each cabinet could be the only power input required. Expansion would provide cooling, while the work produced by expansion could be employed for display lighting. Heat rejection would provide a local balance between cold air spilling from the cabinet, lessening the requirement for in-store heating.

In the simplest form an air cycle system comprises a compressor, expander and heat exchangers. It is a gas cycle based on the joule cycle, or reversed Brayton cycle. Air is compressed, cooled at constant pressure in a heat exchanger, expanded with the extraction of work, and the resultant cold gas used to remove heat from the refrigerated space or product. The cycle can be operated in both open and closed forms. In open systems one of the constant pressure heat exchanges takes place at atmospheric pressure. For refrigeration the cold air can be applied directly to the refrigerated enclosure at atmospheric pressure and discharged to the atmosphere. The cycle is completed taking air back into the compressor, not necessarily from the refrigerated enclosure. In a closed cycle all the air is contained in a sealed system and heat transfer can take place at any pressure, subject to mechanical limitations.

Gigiel (1994) states that the joule cycle deviates substantially from the ideal Carnot cycle in that none of the heat transfer takes place at the maximum and minimum temperatures in the cycle. Its Coefficient of Performance (COP) therefore compares unfavourably with vapour compression cycles where a larger proportion of heat is transferred at higher and lower temperatures within the cycle. However, the energy consumption of a process is a better guide to the environmental impact than the COP of the thermodynamic cycle.

The cost of the refrigerant used to fill the system is of importance when considering the capital cost of a refrigeration plant and even more so when considering its running cost. These costs are zero with air. In addition the construction of the plant is easier, and hence cheaper.

An important difference between air and other refrigerants is the temperature at which heat is rejected from the cycle. In a vapour compression system the majority of the reclaimable heat is colder than 35°C and is thus not normally reclaimed. Recuperation between the air at the inlet to the compressor and the inlet to the turbine increases the compressor discharge temperature whilst decreasing the air temperature to the cold space, without increasing the pressure ratio across the turbine, hence the input energy. In air cycle the air is discharged from the compressor at temperatures of up to 200°C, thus making it a useful source of high temperature heat.

In the open cycle system, in which air is passed directly as the cooling medium into the refrigerated space, there is no need for a heat exchanger, saving the initial capital cost of the exchanger and the losses in efficiency caused by the temperature difference needed to overcome this resistance to heat flow.

The current technology as normally applied

Closed cycle

Air cycle air conditioning equipment has been developed and roof mounted on German Railway's ICE 2.2 train-sets. The system is based upon modern aircraft air-conditioning technology developed by Honeywell NGL in collaboration with Hagenuk Faiveley. The system is a closed loop. An electrically driven centrifugal compressor provides the first stage of the compression cycle. The air is then further compressed in a 'bootstap' combined compressor and turbine unit on the same shaft. The hot compressed air is then cooled in an ambient air to air heat exchanger before entering the turbine. The lower pressure air then passes through the load heat exchanger cooling the cabin air and then returns to the input of the motor compressor. The cooling capacity of the system is matched to the load by varying the motor compressor speed, and hence mass flow around the closed loop system. The system is pressurised to 3 bar on the low pressure side. This offers control of temperature without affecting air flow within the passenger compartment and prevents any direct path for turbo-machinery noise, which would occur if the system were operated in an open configuration. The noise is a maximum of 78 dB(A) outside the carriage and 55 dB(A) inside. Passenger rail management (1996) reports that the cost of ownership of air cycle systems is lower than the vapour compression equivalent. Even though direct energy costs were about 15% higher than for competing vapour compression systems. Weight savings of 25-30% are possible and space requirements are reduced. Maintenance is made significantly simpler by the absence of chemical refrigerant.

The cooling capacity of the unit is 31 kW at 32°C and 60% RH with an electrical input energy of 45 kVA. The performance is for a carriage with 70 passengers, static and in full sunlight. The pack space envelope is 3.3 x 1.4 x 0.55 m. Heating is provided by 38 kW electrical heaters, weight is 750 kG.

Honeywell identified the advantages of their system as ease of maintenance and equipment exchange for servicing, due to there being no refrigerant to remove. It remains operational when overloaded in high ambient temperatures, with minimal reduction in performance. There is only a reduction in duty if the system leaks as opposed to complete failure with vapour-compression. It has high reliability due to few moving parts and ease of manufacturing assembly.

Air products have developed a system which uses high-pressure air (83 Bar a) to produce air temperatures at –50 to –100°V. The closed cycle air refrigeration (CCAR) system has been installed in a processing facility at Kodak in New York State, in the U.S. It produces 210 kW of refrigeration at –73°C

Open cycle
Aircraft air conditioning is the most normal application for air cycle cooling (Rogers, 1994). Kajima Corporation have developed and installed an air cycle conditioning system for frozen storage warehouses, called AIRS. This is an open cycle, directly circulating cold air through the warehouse, controlling the temperature to –30°C. A motor driven compressor delivers air at 1.4 bar and 65°C. Energy is then rejected to ambient air or a water cooling circuit. The air is then further compressed to 2.1 bar by a bootstrap compressor driven by an expander on the same shaft. Again energy is rejected to ambient or water yielding air at 40°C. The air is then pre-cooled to –20°C by air at –30°C returning from the room to the inlet of the first stage compressor. After expansion the air is –55°C and is then introduced directly into the cold space.

Open systems are directly affected by moisture when ice is formed at sub-zero temperatures at the turbine outlet. This ice can deposit on surfaces and collect, thus increasing the pressure drop and reducing the pressure ratio across the turbine. Any large particles of ice can damage the rotating components. The AIRS system employs a filter at the outlet of the turbine which is periodically defrosted using hot air at 30°C from the compressor, by-passing the aftercooler and recuperator. Kato *et al.* (1998) report a back pressure of 5 kPa reduces the efficiency of the system by 15%. The system has been installed in a commercial frozen storage warehouse with a thermal load of 15 kW at –24°C with a COP of around 0.8 and compared directly with the existing R22 system. Kato et al. (1998) reports that the air flow ratio for the R22 system is eight times greater due to the lower temperature differential between the air off and room temperature. The fan energy required was greater than 10% of the total energy for the R22 system. Icing of the evaporator on the R22 system reduced efficiency and defrosting increased the thermal load of the cold store. This did not affect the AIRS system as the air was introduced directly into the cold store. The efficiency of the AIRS system was also less affected by ambient temperature.

The leading practice as being currently tested in trials

Retail display cabinets
Supermarket retail display cabinets using air as the refrigerant have been developed by Russell and Fitt (1998). The cabinets were developed to a fully operational prototype stage but were not demonstrated outside a laboratory environment. Full scale 2.44 m cabinets were developed to produce air temperatures of between –0.5 to 4.6°C in a chilled multi-deck cabinet , equating to a cooling load of 1.4 kW, and of between –27.4 to –37.6°C in the frozen well display cabinet, equating to a cooling load of 0.8 kW. The cabinet had only a single connection for supply of dry compressed air at 3.6 bar. The air was passed directly into an expansion turbine mounted beneath the cabinet and then passed directly into the chilled space. The air entered the cabinet through a series of nozzles at the rear of the cabinet. Cabinet air was entrained, thereby creating a recirculation of air within the cabinet. The open cycle system resulted in a loss of air from each cabinet equal to that introduced from the turbine, 0.042 kg/s. The mass of air circulated in the chilled multi-deck was 0.1 kg/s. The velocities required to reduce infiltration in the open topped frozen well were less than the upright open –faced multi-deck cabinet. This was due to the return air being used for pre-cooling the compressed air prior to expansion; therefore the mass flow of the air curtain was equal to the mass flow introduced from the turbine resulting in no recirculation of cabinet air in the frozen well.

The ejector system eliminated the need for a heat exchanger and fans within the cabinet, thereby increasing the space available for storing and displaying product and reduced energy consumption. The infiltration loads were less than for vapour compression cabinets. The duty for the multi-deck was reduced from 1.36 kW/m for a conventional vapour compression cabinet to 0.57 kW/m for the experimental air cycle cabinet. The duty of the frozen well cabinet was reduced from 0.67 to 0.32 kW/m (Russell and Fitt, 1998).

A turbo-fan driven by the expansion turbine was used to deliver warm air to the front of the cabinet to reduce the effect of 'cold feet'. The operation was such that any cooling effect from the cabinet was balanced by the warm air, thus imposing a neutral load on the store. The noise levels were similar to existing vapour compression cabinets due to the application of basic silencers.

Heating and cooling systems
Butler and Holder (1996) carried out a scoping study to explore the opportunities to exploit the environmental benefits of air cycles in buildings. The work showed that savings in energy consumption and more for heating and cooling CO_2 emissions were possible in buildings where there was simultaneous heating and cooling requirements.

Gigiel *et al.* (2000) reported on the development of an air cycle integrated heating and cooling system for buildings. The unit delivered hot water at 80°C for space heating and chilled water at 6°C for cooling applications. The unit was installed to condition a suite of offices and delivered 23.7 kW of heat and 10.3 kW of cooling. The unit used prototype rotating components used during the development of the Honeywell train conditioning pack. The unit was designed to integrate with existing building services. The performance of the system was limited by bugetary constraints of the research program, which resulted in the procurement of components with efficiencies lower than currently achievable.

As the heating and cooling provided is coincident, rejection of heat/coolth is required for off-design conditions. This can be either from the water circuit, allowing for rejection at any point within the building or directly from the air circuit, via ducted air to the plant. The COSP of heating is 1.49 and cooling 0.66 at full load. The individual COSP figures can only be compared with combined systems not separate systems (e.g. chiller and boiler) as only one electrical supply is required. Gigiel *et al.* (2000) state that the energy consumption of an optimised unit is less than that of conventional systems for buildings that have a significant proportion of their heating and cooling loads coincident. Therefore, emissions of greenhouse gases from this unit, both direct and from power stations supplying them with energy, are reduced compared to conventional systems.

The emerging practice that might be expected to be commercially applicable in 2-5 years

Air cycle retail display cabinets offer distinct advantages over vapour compression and secondary refrigerant systems. The cabinets only require a single connection of pressurised air (3.7 bar a) which allows a greater degree of flexibility. The cabinets have lower infiltration loads and impose a neutral heat balance on the store. They do not require heat exchangers or fans within the cabinet, using only a single discharge ejector pipe which increases the product space available.

To make air cycle systems competitive with vapour compressions systems for "cooling only" on cost alone, life cycle costs need to be taken into account as direct energy costs are greater for air cycle. Making use of the usable heating or cooling will allow the air cycle systems to compete on energy as long as high efficiency components are used. Advances in bearing and motor technology as well as electronic control and power device technology are producing quieter and cheaper controllable motors for compressors.

The development of lower cost high efficiency rotating machines would significantly increase the adoption of air cycle equipment. Traditionally, the designers of small diameter turbine wheels have not been too concerned with high efficiency. However, system developments in automotive turbine technology can result in greater efficiency as well as inexpensive rotating components. At the larger end of the market, Honeywell, has shown that medium scale production runs of rotating machinery can compete on price, when the whole system is taken into account. This cost reduces further as the economy of scale increases.

Gigiel *et al*. (2000) state that future development of air cycle systems for buildings should use gas or oil rather than electricity as the energy source to reduce the cost of energy to the system. The gas turbine is ideally suited to provide the compressed air needed for the air cycle system, as well as central heat and power.

The opportunities for integrating these technologies within the overall engineering systems in supermarkets

The application of air cycle refrigeration for supermarket retail display cabinets requires a source of dry compressed air. If the total air required for a typical supermarket based on the results of Russell and Fitt (1998) is supplied from a central plant via a ring-main distribution pipe. This would necessitate a pipe of diameter 720 mm at 3.7 bar. This is impractical for a supermarket system. If several air cycle packs were located around the store designed to meet the requirements of a limited number of cabinets. These could be sited around the perimeter or in an accessible roof void. This would require equipment with a small volume that is light and quiet in operation. This has been shown to be achievable with the train system.

Russell and Fitt (1998) used a single turbo-fan unit within each cabinet, supplied by a single flexible compressed air connection. If the Honeywell NGL bootstrap unit were used a run of 16 to 20 m of cabinet could be connected to a common supply duct. This reduces the degree of system flexibility . If recuperation is required it is practical to enclose the supply pipe within the return pipe to the recuperator thereby enhancing recuperation, and maintaining a single pipe connection. Drying systems needs to be energy efficient. Adsorption wheels offer low pressure drops, hence maintain the system efficiency, to dew points around –50°C, using rejected heat of compression to continuously regenerate the adsorbent. Heat can be recovered from the rejected moist air for space heating in the store if required.

Closed cycle systems for building air conditioning have been successfully integrated with existing air conditioning systems (Gigiel, 2000). The air cycle plant was contained within a centralised unit and connected directly to the existing hot and cold water system, replacing a boiler and chiller plant. Equally, the air itself could be distributed around the building. The development of open cycle systems for building air conditioning could take the same form as for air cycle retail display cabinets, with localised turbine units being fed from a central compressed air supply network. This system will mainly be applicable to new build, with limited scope for retrofitting.

The availability of models that might be used to help assess the performance of these technologies

Computerised models have been developed by FRPERC, University of Bristol to predict total system performance, integrating the air cycle plant with the distribution system. These models incorporate component performance data and have been verified against actual system performance (Gigiel, 2000). Commercial modelling packages exist for modelling thermodynamic cycles, including air cycles. Cycle-Tempo produced by Delft University of Technology in the Netherlands can be used to model and optimise energy conversion systems in steady state. The program offers the user a number of apparatus models, which can be selected and linked to each other in any combination. Compressors, expanders and heat exchangers are available. The thermodynamic properties of fluids involved in air cycle systems (gas mixtures including humid air, water/steam) are available.

Commercial modelling packages to predict the heating and cooling requirements of buildings are available. The best known and world-wide most applied tool which can generate these energy demands is TRANSYS package developed by the Solar Energy Laboratory of the University of Wisconsin-Madison.

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Report F

Refrigerants - EA Technology & Calor Gas

Short description of the topic/technology status - proven/developing

The Montreal Protocol to protect the ozone layer, and the EU regulations which implement the Protocol have led, over the last ten years, to major changes in the choice of refrigerant for use in supermarket refrigeration.

Prior to the Protocol three refrigerants were used: CFC 12 was used for some HT refrigeration, HCFC-22 was used for both high and low temperature refrigeration and CFC 502 was the most common refrigerant used for both HT and LT refrigeration.

The initial reaction was to increase the use of HCFC 22. This was often combined with the use of screw compressors which permitted liquid injection reducing the discharge superheat.

When HCFCs were also restricted under the Protocol other refrigerants were developed, generally blends of different HFCs.

Whilst HFCs have no ozone depletion potential they have large global warming potential if released into the atmosphere. For this reason the use of so called 'natural refrigerants' has aroused fresh interest; particularly the use of hydrocarbon refrigerants and ammonia. Air cycles are dealt with in another paper and water cycles are not considered practical (as their minimum operating temperature is 0°C) for supermarket refrigeration. This paper will therefore concentrate on hydrocarbons and ammonia as the new technologies compared to HFC404a as the conventional technology.

Performance

The performance of most of the HFC refrigerant blends is similar (within 5%) and depends on the detailed design of the system. (e.g. some refrigerants will benefit more by the fitting of a liquid line suction line heat exchanger than others)

Two tests carried out by George Barker showed an energy saving of 20% when a R290/R600a blend replaced CFC134a in an Impulse cabinet and a saving of between 3 and 6% when either HC290 or a HC290/HC600a blend replaced HFC404a in a Highline cabinet.

Tests by Linde on Copeland Scroll compressors showed a performance increase of 27% for HT operation and 11% for LT operation in switching from HFC404a to HC1270.

However in many cases hydrocarbons and ammonia require secondary refrigerants and this imposes additional energy consumption. This is discussed in the review of Integral, secondary and direct systems.

The above covers the energy consumption of the refrigerants, but many companies are also interested in the overall environmental performance. As none of the current refrigerants deplete ozone the main comparison must be the total environmental warming impact (TEWI). This adds together the contribution to global warming from the carbon dioxide produced by power stations producing the electricity to drive the refrigeration system, over the lifetime of the plant, to the global warming impact of the refrigerant which is lost from the refrigeration system over its lifetime. The latter should include any loss during commissioning, leakage during operation and any refrigerant which cannot be recovered from the plant at the end of its lifetime.

The British Refrigeration Association publishes a booklet to assist in calculating the TEWI of a system. The energy consumption can be estimated reasonably accurately, however the leakage rates are more difficult to estimate, and due to the high global warming impact of most HFCs, can have a very significant effect on the TEWI.

Applications - proven/potential

Many companies have assessed the use of ammonia or hydrocarbons for their store's refrigeration systems.

Marks and Spencer installed an ammonia system at their Handforth Dean store. Four air cooled ammonia chillers cool a propylene glycol brine to around -9°C, this is circulated around the store. This directly cools the HT cabinets, whilst the LT cabinets are cooled by small packaged HFC404a scroll compressor with heat rejected into the glycol loop.

Overseas the IEA Annex 22 report gives details of ammonia/secondary systems in USA and Denmark, Norway and Sweden with a range of secondary systems, including propylene glycol, Tyfoxit, CO2 (with phase change) and some Binary Ice storage systems.

Tesco constructed a store at Enfield in 1998 using hydrocarbon refrigerant HC1270 (Propene) distributing cooling using the proprietary Tyfoxit brine (this has the advantage of being low viscosity even at temperatures as low as -30°C).

Sainsbury's have recently opened their Millennium Store using hydrocarbon refrigerants with heat rejected to borehole water, giving an excellent level of performance.

Overseas the IEA Annex lists systems in Sweden and Germany, both using HC1270 as the refrigerant. The Swedish system uses Propylene glycol and CO2 as the secondary refrigerants whilst the German system uses Tyfoxit.

Integral refrigeration cabinets, both HT and LT, are available.

Limitations

The main limitations are introduced by the safety requirements for use of the refrigerant. The are discussed below.

Cost - current/mature market

The cost of the refrigerant is still a small part of the overall cost of the refrigeration system. Most refrigerants are priced at a similar level, with the exception of HCFCs which are increasing in price as their phase out date approaches.

At present integral units using hydrocarbons are more around 20% more expensive than those using HFCs. This is mainly because they are seen to command a premium as they are more energy efficient and environmentally desirable. In a mature market there is an additional cost for hydrocarbon units due to the safety requirements (use of non sparking electrical equipment however this is counterbalanced by other costs being lower for HC units, notably the lubricant, and the overall cost difference should be small.

The cost of secondary systems is discussed in the accompanying paper on secondary refrigerant systems.

Maintenance

The only main difference for maintenance is in the training of the maintenance staff in handling HC rather than HFC refrigerants. This can be provided in a one day course and is not a significant additional cost.

Environmental and safety issues

The environmental issues are the driving force behind the change in refrigerants and are discussed above. Safety issues are the main limitation in the application of ammonia and hydrocarbons.

Refrigerants are classified for safety according to their toxicity and flammability.
Class A have a TLV<400ppm and class B have a TLV>400ppm.
Class 1 are non-flammable, Class 2 have an LFL less 3.5% by volume and Class 3 have a lower LFL.

CFCs and all HFCs sold into the refrigeration market are Class A1. Ammonia is class B2 and all hydrocarbons are class A3.

Their use is covered by a range of both electrical and refrigeration standards.

In supermarkets classed as restricted access spaces this limits the charge per unit to 1.5kg. The low liquid density of hydrocarbons means that this will replace a unit containing around 3kg of R404a or R22.

The requirements for electrical safety are more complex as there is a wide range of standards which might apply which are not always consistent. However the following are likely to be sufficient for commercial refrigeration:

- Avoid the possibility of direct sources of ignition such as exposed electrical contacts or excessively hot surfaces (> 350°C).
- Ensure that electrical components only comprise solid state parts or have casings which are solid encapsulated or otherwise sealed to at least IP65 or are located externally to the casing of the refrigerant containing parts.
- Ensure that electrical terminations, including capacitor terminations are adequately tightened and secured against loosening and that adequate insulation is provided to avoid live parts shorting together.
- Ensure that any motor, including fans, pumps and compressors are of brushless design.

Existing analysis tools

Present analysis tools are relatively limited. Simple theoretical calculation of the theoretical comparison of the performance of refrigerants can be carried out using the thermodynamic properties listed in NIST REFPROP software. However this only covers direct drop in replacement and does not allow for factors such as system optimisation for each fluid, which is best left to the designer.

The calculation of energy comparisons of DX vs Secondary systems is covered in the accompanying paper on Integral, distributed or secondary refrigeration.

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Report G

Performance of Variable Speed Compressors – Department of Mechanical Engineering, Brunel University

Summary

Variable speed capacity control in refrigeration compressors for small to medium capacity refrigeration applications has been investigated over the last 30 years. This technology can now be found on a wide variety of small capacity packaged air conditioning and heat systems from Japanese manufacturers but has not yet gained wide acceptance in the commercial refrigeration sector. This paper reviews work carried out internationally on the development of variable speed refrigeration systems over the last few years and outlines the main factors impeding the wide application of the technology to the commercial refrigeration sector. The paper also presents results of experimental studies on variable speed refrigeration systems carried out at Brunel University using a semi-hermetic compressor widely employed in refrigeration packs in supermarkets. The results show that operation at variable speed can lead to 24% energy savings over fixed speed operation.

Introduction

The rapid advancement of computer and power electronic technologies offers considerable opportunities for better flow and temperature control in industrial and HVAC applications. Although the use of power electronic based variable speed drives (VSDs) has been widely adopted in the process industries it has so far made little inroads into the HVAC industry. The refrigeration industry in particular, has been very conservative in employing sophisticated capacity control techniques and has not fully benefited by the advances in electronic and control technologies to increase system operating efficiency, reduce energy consumption and achieve more precise temperature control. The main reasons for the reluctance of the refrigeration industry to adopt inverter based variable speed drives are the relatively high capital cost of VSDs and the fact that initial demonstration installations have failed to produce the expected energy savings [1].

Presently available off-the-shelf VSDs are not specifically developed for compressor motor applications and are therefore designed with several user-selectable control parameters. These parameters offer the flexibility to set-up the drive for its intended application. However, where VSDs have been applied to refrigeration applications, the main consideration has been their performance under steady-state conditions with little or no concern about their dynamic performance.

As the compressor is the largest energy consumer in a vapour compression refrigeration system, its transient and steady-state energy requirements have a direct effect on the overall system operating efficiency and cost. The transient performance of a compressor in a particular installation depends on the individual system design and may vary due to the start-up control strategy adopted. It has been shown in the literature that for a fixed speed system the transient losses can be up to 20% depending upon the number of on/off cycles and percent on-time during each cycle [2]. Other factors which affect the transient losses are the mass of the heat exchangers and the dynamics of the refrigerant within the system. The transient losses lead to a decrease in the seasonal COP and hence an increase in the operating cost of the system [3,4].

In a variable speed system, the transient losses are only present during the initial load pull down when the system starts-up from cold and at low load cycling. In most variable speed compressors, speed reduction is limited to between 25% and 30% of maximum speed to ensure adequate oil return and motor cooling at reduced speeds. When the load lies between the values corresponding to minimum and maximum speeds, after the initial start-up the system will track the variation in load by regulating the compressor speed without incurring additional transient losses. This paper presents the results of experimental investigations carried out to quantify and compare the transient start-up losses of direct mains and inverter driven refrigeration systems. The transient losses together with the steady-state performance of the two systems were then used to determine their seasonal performance in air conditioning applications.

Variable speed refrigeration compressors offer a number of advantages over conventional fixed speed systems. These advantages include:

a) improved steady state efficiency at part load operation
b) reduced overdesign capacity in refrigeration plant providing savings on capital cost
c) close control of temperature
d) built in `soft' starting and power factor correction producing running cost savings.

Recent attempts to introduce the VSD concept to commercial refrigeration systems in the UK, however, either failed or made little progress. The main reasons for this are:

i) very little attention was placed on the integration of compressors and VSD technologies. Commercially available systems mainly consisted of general purpose off- the-shelf inverters coupled to rotary or reciprocating compressors
ii) the use of general purpose VSDs with a large number of surplus components contribute substantially to the capital cost of the equipment which make them unattractive to a low first cost conscious industry
iii) very little information is provided by manufacturers on the performance of integrated VSD/compressor systems
iv) initial installations have suffered from reliability problems caused by unsophisticated and inadequately developed control systems.

Research at Brunel University over the last few years was aimed at providing answers to some of the above uncertainties, identify the most suitable compressor and VSD technologies for variable speed operation. This paper presents a review of the application of VSDs to commercial refrigeration systems and some results from the Brunel programme on the performance of three alternative compressor technologies driven by a Pulse-Width-Modulated inverter.

Refrigeration capacity control through compressor speed modulation

The basic difference between variable speed refrigeration and conventional refrigeration systems is in the control of the system capacity at part load conditions. In variable speed refrigeration the capacity of the refrigeration system is matched to the load by regulating the speed of the compressor motor in a way that the capacity of the system tracks the load dictated on it by varying operating conditions. Variable speed control can be realised in a number of ways which can be divided into two groups. Firstly, those in which the load is indirectly coupled to the motor (a constant speed motor and a speed control device between the motor and the load) and secondly, those in which the load is directly coupled to the motor (a variable speed motor). The advantages and disadvantages of each group of capacity control methods have been discussed in greater detail in reference [4]. In refrigeration applications, stepwise or infinitely variable control of the motor speed are considered to be the most flexible methods of speed control and significant energy savings have been reported through their application [5,7,8,9,10].

Stepwise speed control can be achieved by using multi-pole electric motors. The required compressor capacity is obtained by switching a finite number of poles to achieve the desired speed. Stepwise control is less costly than continuous speed control but step controlled motors have lower efficiency than constant speed motors [3]. Moreover, this method of speed control has the limitation of a fixed number of speeds which offers restricted compressor capacity control compared to continuous stepless capacity control.

Infinitely variable capacity control can be realised by using electronic variable speed drives to regulate the speed of the compressor motor. Since the torque-speed performance characteristics of an induction motor at low speed operation are the same as those at rated motor frequency, frequency variation is considered to be an efficient speed control technique.

Early work on variable speed refrigeration systems was directed towards the theoretical analysis of the concept of variable speed capacity control and the investigation of the problems associated with the mechanical design of the system. Most of the published work discusses the overall performance and benefits of the system rather than the establishment of criteria for the integration and optimisation of compressors and variable speed drives [5,6,7,8,9,10]. The investigators reported energy savings on a seasonal basis in the range 20% to 40% and considerable reductions in temperature fluctuations in spaces conditioned with variable speed air conditioning units. The investigators also pointed out various problems that needed further consideration. These included, improvements in the refrigerant throttling mechanism; adoption of more effective noise suppression techniques to reduce radio wave interference noise and harmonic noise generated by the inverter; enhancement of the reliability and performance of the inverter; improvements in the overall system design to reduce noise at high frequency operation and overcome vibration problems at low frequency operation. It was concluded that the cost of the inverter control system needed to be reduced further to expand its application.

ASHRAE research project RP-409 analysed a large chiller employing a variable speed controlled centrifugal compressor [11]. The results showed that variable speed control led to a 1.5% reduction in the compressor power consumption at maximum load and 40% reduction at minimum load.

Wong et al [12] carried out experimental investigations to compare cylinder unloading with variable speed control on a commercial refrigeration system. They found that with variable speed control, volumetric and isentropic efficiencies and COP increased when the compressor speed was reduced while cylinder unloading control exhibited reduced isentropic efficiency and COP. The authors evaluated the economic benefits of a variable speed compressor in another research paper [13]. It was shown that variable speed control leads to reduced energy consumption but for intermittent operation it may not be economically viable due to the high capital cost of the inverter.

Tassou et al [1,14,15,16,17] showed that variable speed control applied to air-to-water heat pumps could achieve 15% improvement in energy conversion efficiency compared to a conventional system. It was also found that superheat control with a thermostatic expansion valve was unsatisfactory during part load operation and was suggested that the problem could be effectively overcome by employing a microprocessor controlled motorised expansion valve.

McGovern [18] investigated the performance of a two cylinder open type reciprocating compressor over the speed range 300 to 900 rpm. Performance parameters such as mass flow rate, shaft power and compressor discharge gas temperature showed linear increase over the tested speed range, whereas the volumetric efficiency was found to remain almost constant at about 66% over the speed range. The variation in mechanical efficiency with speed was found to be very small, increasing from 92% to 94% as the speed increased from 300 rpm to 900 rpm.

In recent years scroll compressor technology has shown promising efficiency advantages over comparable positive displacement compressors. This is due to the smooth and continuous compression characteristics of the scroll design and the elimination of valve losses. Ischii et al [19,20] compared the mechanical efficiency and dynamic performance characteristics of scroll compressors with those of rolling piston rotary compressors. It was found that the scroll compressors exhibited better vibration characteristics than the rolling piston rotary compressor but lower mechanical efficiency. It was anticipated that the mechanical efficiency of scroll compressors could be improved through design optimisation.

The investigations of Senshu et al [21] on a small capacity heat pump employing a scroll compressor showed 30% improvement in annual performance efficiency compared with the conventional reciprocating compressor. The EER of the inverter driven heat pump at nominal load conditions, however, was found to be less than that of a constant speed system due to the inverter losses.

A feasibility and design study of a continuously variable capacity refrigeration system was carried out under the Energy Efficiency Demonstration Scheme on behalf of the Department of Energy [22]. A commercially available variable speed system was monitored in a supermarket application with a view to first assess the performance of an already installed conventional system and then to convert these units to variable speed for overall comparison. The investigation showed 56% power savings with high temperature (dairy applications) and a 30% savings with low temperature (frozen food applications). The energy savings achieved were attributed mainly to variable speed control and fully floating head pressure.

Rice et al [23,24,25] reported energy savings of the order of 27% for a modulating heat pump system arising from reduced cycling losses, heat exchanger unloading, reduced frosting/defrosting losses and reduced backup heating. It was found that increased motor slip losses and distorted inverter waveform decreased the conventional three phase induction motor efficiency by up to 20% depending on frequency and inverter type. It was suggested that a permanent-magnet, electronically commutated motor inverter combination could reduce these losses.

Riegger [26] compared the performance of commercially available small capacity rotary, reciprocating and scroll compressors to evaluate their performance under variable speed operation. It was found that all three compressors were optimised for 60 Hz operation and their energy efficiency ratio decreased above and below this rated point. The authors concluded that there is no straightforward answer to the question of which type of compressor is most suitable for variable speed operation because various factors such as capacity range, operating conditions and manufacturing cost influence their seasonal energy efficiency.

This report gives results of experimental investigations carried out at Brunel University on the performance characteristics of a semi-hermetic compressor widely used in supermarket refrigeration applications.

Test facility

The experimental investigations were carried out under controlled loading conditions in the laboratory. The test rig is based around a chiller of a nominal cooling capacity of 25 kW. The chiller is equipped with a shell and tube condenser, a shell and tube direct expansion evaporator coil, and a thermostatic expansion valve. The design allows the installation of different types of compressors and drives for comparative investigations. Test conditions on the chiller are achieved through a water loading system and a recirculatory air tunnel which acts as a balancing mechanism between the hot and cold sides of the system.

The test rig is equipped with a comprehensive instrumentation and PC-based data logging system enabling accurate measurement of thermodynamic and power parameters. Three-phase power was measured using a precision universal power analyser capable of measuring fundamental components as well as the harmonic contents of voltage and current up to the 50th harmonic.

The compressor tested was a reciprocating (semi-hermetic suction-gas cooled type) - 33.07 m3/h at 50 Hz, commonly found in supermarket refrigeration packs.

Tests were carried out at constant condenser pressures (head pressures - CHP) of 12, 14, 16 and 18 bar (absolute) and floating head pressures between 11 bar and 18 bar with the 11 bar pressure corresponding to the speed of 25 Hz and the 18 bar corresponding to the maximum speed of 60 Hz. The results of constant head pressure tests are described briefly in the following sections.

Performance characteristics

Effect of variable speed on main performance characteristics
The compressor speed influences various operating and design parameters such as cooling capacity, power consumption, COP, volumetric and isentropic efficiency. In this section the main performance characteristics of the compressor are shown under constant head pressure control of 13 bar, 14 bar, and 15 bar and variable speed operation in the range 25 Hz to 55 Hz.

Effect of Variable Speed on Cooling Capacity
The effect of compressor speed on the cooling capacity of the compressor at a constant condensing (head) pressure of 15 bar, 14 bar, and 13 bar is shown in Figure 1. It can be seen

that there is a linear increase in capacity as the compressor frequency is increased from 25 Hz to 55 Hz. The maximum cooling capacity is gained at maximum speed. It can also be seen that increasing the head pressure reduces the cooling capacity of the compressor.

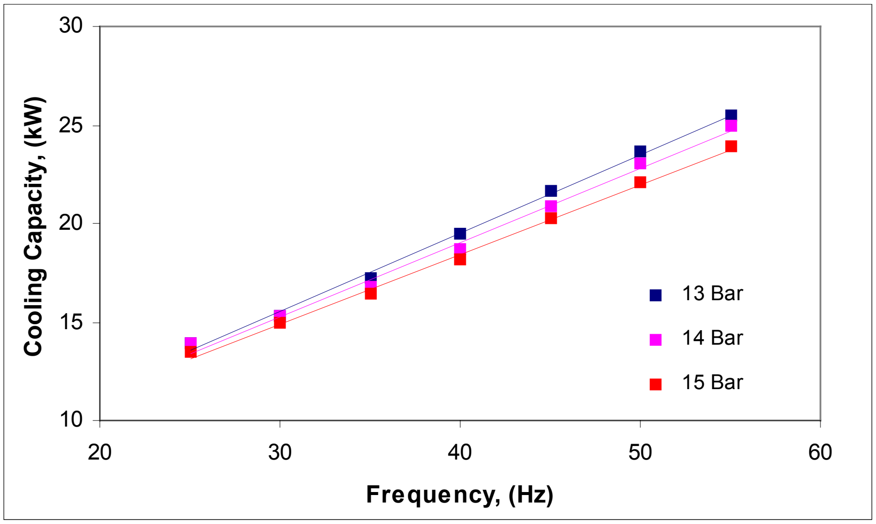

Figure 1. Variation of Cooling Capacity with Speed at three Constant Head Pressures of 13 bar, 14 bar and 15 bar

Effect of variable speed on power
Figure 2 shows the effect of variable speed on power consumption. It can be seen that for the three pressures the power increases linearly with the increase in speed. The increase in pressure also causes a higher power consumption as the compressor has to pump against a higher pressure differential.

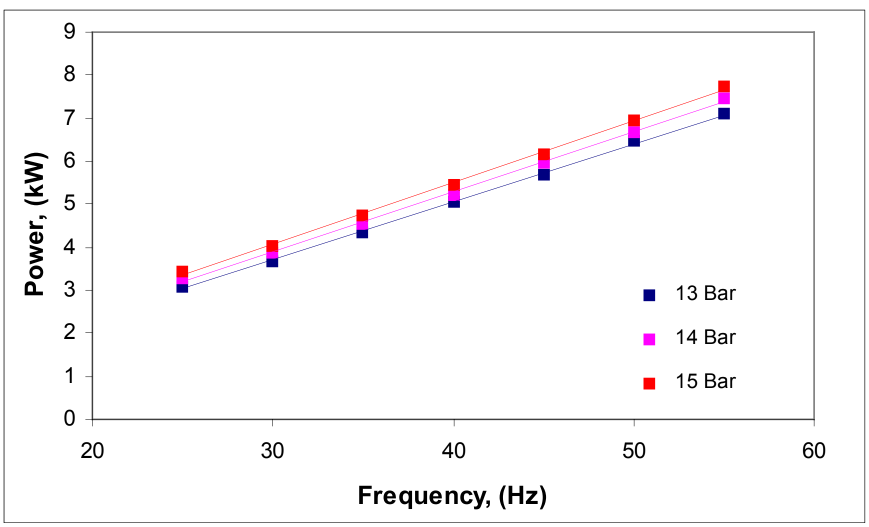

Figure 2. Variation of Power with Speed at three Constant Head Pressure of 13 bar, 14 bar and 15 bar

Effect of variable speed on COP
The performance of refrigeration systems is expressed as the ratio of useful heat transferred to work input, defined as the Coefficient of Performance (COP). Figure 3 shows the variation of

COP against the frequency for the three different constant head pressures of 13 bar, 14 bar, and 15 bar. It can be seen that the COP increases with a reduction in the compressor speed.

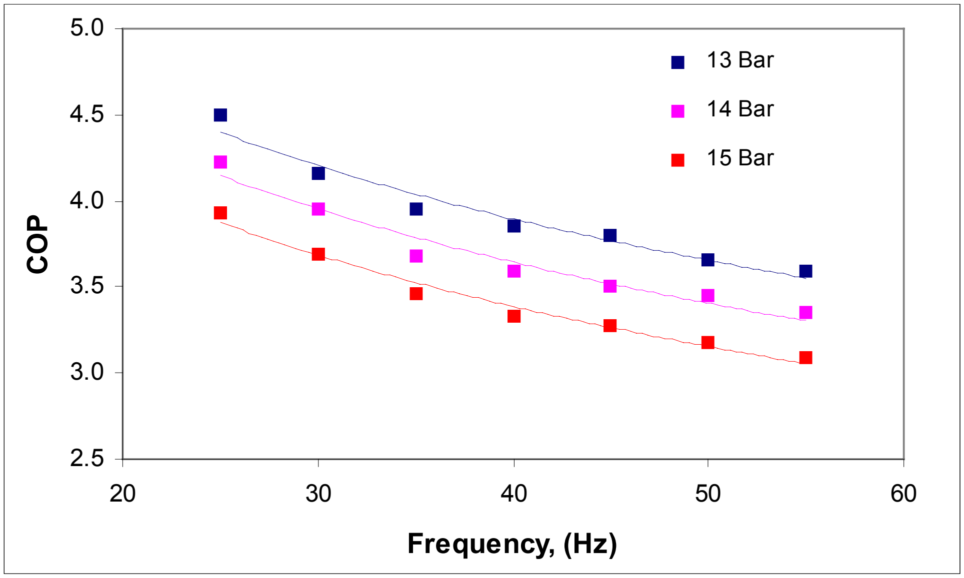

Figure 3. Variation of Coefficient of Performance with Speed at three Constant Head Pressures of 13 bar, 14 bar and 15 bar

Figure 4 shows the variation in the cooling COP with speed from its steady state value at 50 Hz for the three head pressures of 13 bar, 14 bar and 15 bar. It can be seen that the COP shows a rise of approximately 24 % as the speed was reduced from 50 Hz to 25 Hz for all three head pressures.

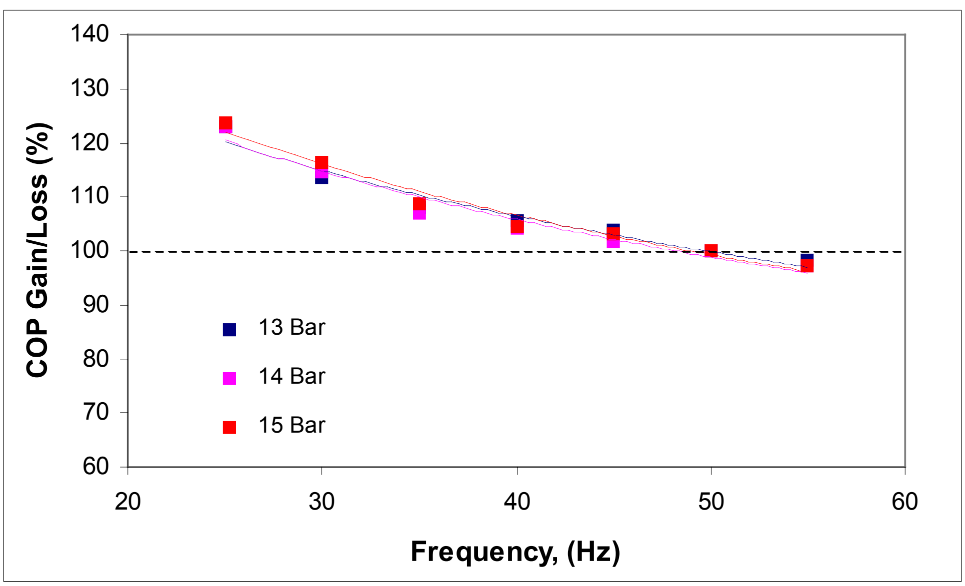

Figure 4. Gain/Loss of cooling COP with speed from a nominal value

Effect of variable speed on volumetric efficiency

Volumetric efficiency is defined as the ratio of actual volume of fresh gas induced to the compressor swept volume and is a measure of the compressor's ability to pump refrigerant gas. The variation of the volumetric efficiency for the three different head pressures is shown in Figure 5. It can be seen that the volumetric efficiency reduces with speed.

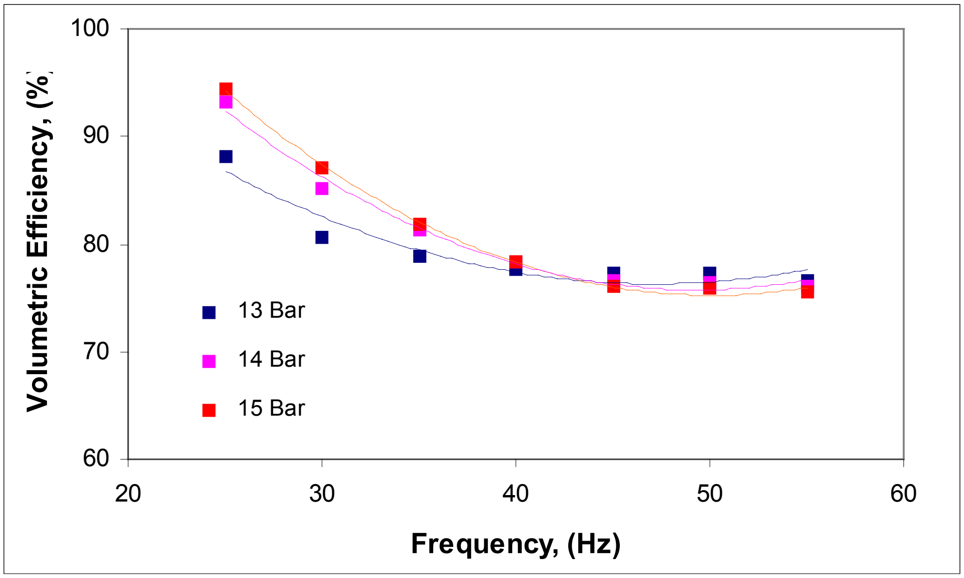

Figure 5. Variation of Volumetric Efficiency with Speed at three Constant Head Pressures of 13 bar, 14 bar and 15 bar.

Effect of variable speed on isentropic efficiency

The variation of the isentropic efficiency is shown in Figure 6. The three head pressures show an increase in isentropic efficiency with a reduction in speed. This is due to the lower friction losses and discharge temperature at lower speeds.

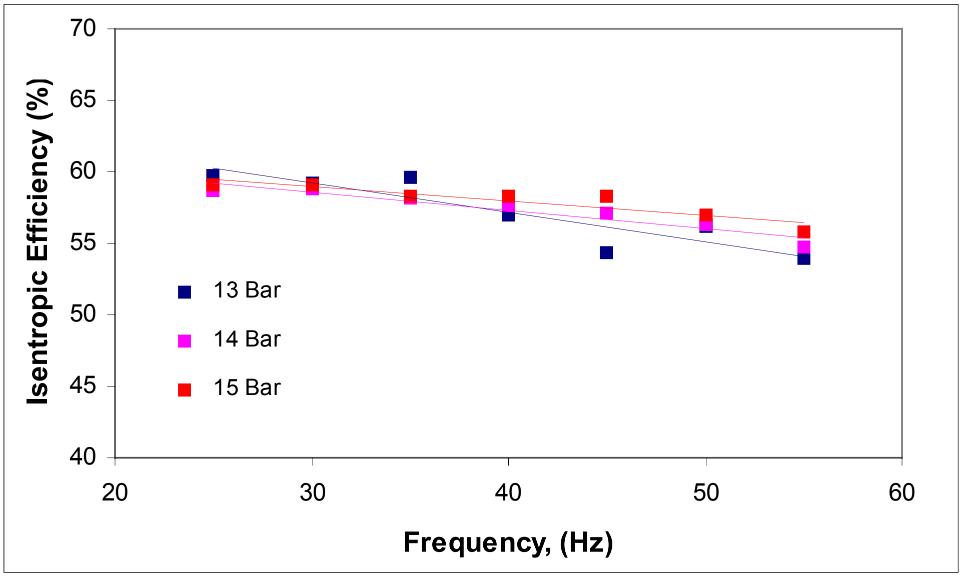

Figure 6. Variation of the Isentropic Efficiency with Speed at three Constant Head Pressure of 13 bar, 14 bar and 15 bar

FABER MAUNSELL

Variation of the specific power at variable speed
The variation of the specific power consumption with speed for the three head pressures is shown in Figure 7 below. It can be seen that the specific power shows a reduction with a decrease in speed. A decrease in the specific power indicates an improvement in the coefficient of performance. The specific power can be defined as the inverse of cooling COP.

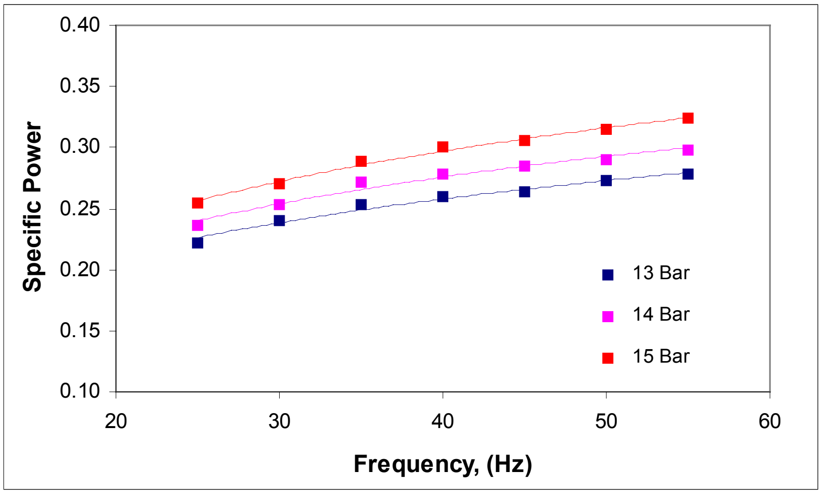

Figure 7. Variation of the Specific Power with Speed at three Constant Head Pressures of 13 bar, 14 bar and 15 bar.

Conclusions

1. The literature review revealed that the downward price trend and new technological developments are favourable for the increased use of VSD refrigeration in place of conventional refrigeration systems. There is also scope for achieving further energy savings by using high efficiency motors if their present costs become competitive with standard induction motors.
2. The development of an optimum variable speed refrigeration system is a function of several design factors and more research work is needed to fully understand the interaction of the components in an integrated VSD refrigeration system. Problems to overcome are the generation of harmonics by the inverter which affect both the supply and the motor, and the reduction of the motor efficiency at low speeds. Proper lubrication and cooling of the compressor at low speeds is also an important consideration. In the tests carried out the overall efficiency of the inverter was found to be 95%.
3. The experimental tests showed that savings of the order of 24% can be achieved as the speed is reduced from 50 Hz down to 25 Hz.
4. Variable speed can also lead to better control of the target suction pressure in supermarket refrigeration systems.

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Report H

Evaporator Coils and Frosting/Defrosting Issues – Department of Mechanical Engineering, Brunel University

Introduction

The design or selection of the evaporator coil in refrigeration systems is critical to the overall performance and life cycle cost of the system as it affects both the capital and running cost and to a large extent determines the temperature performance of the cabinet. In display cabinets and cold rooms the evaporator coil should provide air at a temperature below the temperature that the product has to be maintained at, to overcome the losses from the refrigeration fixture incurred by convection, conduction, infiltration and radiation. To reduce the space occupied by the coil, evaporator coils in the food retail industry are normally of the finned tube type with circular copper tubes with continuous aluminium fins. The fins are employed to increase a) the heat transfer surface area of the coil and b) the air side heat transfer coefficient through increased air velocity and turbulence. High fin densities, however, increase the air flow resistance through the coil which would in turn increase the air pressure drop and fan power. The refrigerant is normally arranged to flow through the tubes in a cross-counterfow direction to the air flow. To reduce the refrigerant pressure drop through the coil, a number of parallel refrigerant sections can be employed. The optimum number of sections is a compromise between pressure drop, the refrigerant side heat transfer coefficient which is a function of the pipe diameter and the flow rate in each tube and manufacturing cost. Depending on the surface temperature of the coil and the dew point temperature of the air, an evaporator coil will operate \`dry' or \`wet' if the air is cooled below its dew point. The performance of a dry coil will be a function of the surface area of the coil, the overall heat transfer coefficient and the logarithmic temperature difference between the hot and cold fluids. For \`wet' operating conditions the performance of the coil will be a function of the logarithmic enthalpy difference (Tassou, 1982)

The evaporator coils of most refrigeration systems in chilled and frozen food applications will operate with coil surface temperatures below 0 ^{o}C. Water vapour present in the air, in coming into contact with the coil surface, will first condense and then freeze if the condensate is not drained fast enough from the coil surface. Although a small amount of frost may improve the heat transfer performance of the coil by increasing the surface area and surface roughness which induces increased turbulence, significant frost accumulation deteriorates the coil performance by reducing the air flow and thereby the refrigerating capacity of the evaporator Stoecker (1957). Consequently, frosting is a major problem in refrigeration systems and evaporators need to be defrosted periodically to maintain system performance and temperature control.

Studies on frost formation

The majority of studies concerning frosting on finned tube heat exchangers have been experimental. The conclusions drawn from these studies are qualitative and the results are specific to the test conditions and coil dimensions and hence of limited practical use. Nevertheless, they do give an indication of the main factors affecting frost accumulation and the effect of frost on system performance. The growth characteristics of frost and frost properties in general have been well documented by Harraghy and Barber (1987). Also, O'Neal and Tree (1985), Padki et. al. (1989) and Kondepudi et. al. (1987), Datta (1999) have conducted extensive surveys in this area.

Almost all investigators agree that for finned tube heat exchangers the main factors affecting the rate of frost deposition and the general character of frost are; coil geometry (fin spacing, tube and fin arrangement), air temperature and humidity, air velocity and fan characteristics, cooling surface temperature and cooling time.

The influence of coil fin spacing was investigated by Stoecker (1957), Al-Sahaf (1989) and Lee et. al. (1996). These investigators carried out similar tests on coils with varying fin spacing. The general conclusion is that heat exchangers with narrower fin spacing have a higher pressure drop across them due to a smaller free flow area. Also, as the fin spacing gets closer the heat transfer rate increases, increasing the rate of frost growth and the total amount of frost accumulated due to the larger surface area. The implication of this is that a wider fin spacing with a larger coil block is preferable where frosting is likely to occur at the cost of lower heat exchange rate. The fin arrangement of a coil can be either staggered or in-line. In the staggered fin arrangement the pitch of the fins is variable, that is, the fins at the front end of the coil are more widely spaced than the fins at the rear end or vice versa. In the in-line arrangement the fins are equally spaced throughout the width of the coil. Researchers have shown that the reduction in air flow rate for a given deposit of frost is less severe for coils with staggered fin arrangement, than for coils with in-line fin arrangement, Lee et. al. (1996).

Gates (1967) and Al-Sahaf (1989) studied the effect of the number of tube rows, (along the flow direction), on the heat transfer and pressure loss characteristics of the coils during the frost formation process. The rate of heat transfer was found to be affected only marginally by the number of rows. The pressure drop through the heat exchanger, however, was found to increase rapidly with the addition of extra rows. Al-Sahaf (1989) observed that heat exchangers with fewer numbers of tubes but of larger diameter performed better than those with more tubes of smaller diameter when other conditions were kept constant such as block dimensions and flow conditions.

Gatchilov et. al. (1979) carried out a range of frosting tests on finned coils and found that the deposition of frost through the coil in the air flow direction depended largely on the air velocity and humidity. For lower air velocities and humidities the greater frost thickness was found on the downstream tubes with frost on the upstream tubes being thinner but more dense. For higher air velocities and humidities the situation was reversed. This result appears to contradict the findings of Gates (1967) and Niederer (1976). However, Gatchilov did point out that the frost on the upstream surfaces was more dense and as such there would have been a greater mass of frost on the upstream surfaces. It was also observed by Gatchilov et. al. (1979), Lee et. al. (1996) and Al-Sahaf (1989) that the frost first formed on the tubes before spreading to the fins.

The structure of frost changes with coil surface temperature. A higher inlet air temperature results in the reduction of frost thickness, the increase of frost density, and the decrease of thermal resistance. An increasing air humidity leads to a thicker frost layer, a decrease in frost density, an increase in the rate of frost growth and an increase of thermal resistance, Lee et. al. (1996), Al-Sahaf (1989), Gatchilov et. al. (1979) and Harraghy (1987).

Defrost methods

Currently, the most widely used defrost methods, as mentioned in are hot or cool gas, electric and off-cycle defrost.

Off cycle defrost
With off cycle defrost the refrigerant supply to the evaporator is switched off, either by switching off the compressor in a single compressor system or closing a liquid solenoid valve in multi-compressor systems. Defrosting is achieved by circulating air from the fixture over the coil to melt the ice. Since the heat for this method is obtained from the air circulated in the fixture, defrost is quite slow and its application is limited to fixtures maintaining temperatures of 1°C or above (or where an evaporating temperature of just below 0°C is being used to hold a positive cold space temperature).

Off cycle defrost can be controlled by a number of methods including: suction pressure control, time clock initiation and termination, time clock initiation and suction pressure termination, and time clock initiation and temperature termination. The operating costs of off-cycle defrosting are minimal, being simply the running cost of the fans. Its low cost and simplicity makes this method of defrost a popular choice when the evaporator coil controls the temperature of a space or fixture which is maintained at a relatively high temperature.

Electric defrost
Electric defrost is applicable to low temperature refrigerators (i.e. where room or produce temperature is below 0°C) but is also frequently used in meat and dairy refrigerators to give a rapid defrost period.

Electric resistance heaters are incorporated in the evaporator with a connection to a power supply. The heaters can be either inserted into specially formed pockets or dummy tubes within the fin block or positioned on the front face of the coil block. The former arrangement can provide greater heat exchange efficiency but is more difficult to accommodate in the manufacturing process and difficulties have also been encountered with heat transfer, Harraghy (1987). Sufficient space must also be left at the side of the evaporator to allow heater withdrawal from the fin block if they fail. These complexities have made the in-pipe arrangement less popular. The external arrangement is simple as the heaters are clipped to the exterior of the fin block and can normally be removed directly from the face.

When the heating element is in direct contact with the evaporator, heat transfer from the heater to the coil block is predominantly by conduction. When the heater is located between the evaporator fans and the evaporator coil, the predominant heat transfer mode from the heater to the coil is convection. In either case, a temperature-limiting device is placed on or near the evaporator to prevent excessive temperature rise if the defrost controlling device fails to operate.

In addition to the heaters installed to remove the frost from the fin block, an additional heater may be needed in the base of the drain pan to stop the defrost water from freezing before it leaves through the drain connection. Some form of heat is also required to keep the drain free from ice.

The electric defrost method simplifies the installation of remote display cabinets. The defrost cycle is normally initiated by closing a solenoid valve in the liquid line causing the evaporator to be evacuated. The evaporator fan(s) continue to run to allow heat available in the air to boil the remaining refrigerant in the coil. This process is commonly known as the "pump down" cycle. On completion of "pump down", the heating elements in the evaporator are energised and the evaporator fans remain on or off depending on the position of the heating element. If the heating element is placed on the face of the coil then the fans remain switched on so that the heat is blown onto the evaporator coil. When the heating elements are placed inside tubes and between fins, then the fans are usually switched off, as sufficient heat is transferred to the coil via conduction. In general, the electric defrosting heater power is approximately equal to the cooling capacity of the evaporator coil. After the evaporator is defrosted, the heaters are de-energised and the system put back in operation by opening the liquid line solenoid valve. The compressor of unitary equipment is then switched on from the high pressure limit of the low pressure switch.

Electric defrost is normally initiated by a timer and is terminated on one of three methods: a) time elapsed from defrost initiation, b) monitoring of the pressure rise of the suction gas, c) monitoring of the temperature rise of the evaporator surface or the off-coil air temperature.

One problem with electric defrost is that it causes steaming of vapour. This increases the humidity in the refrigerated space and can cause frosting on surfaces in the vicinity of the evaporator, which are not readily defrosted, Fontanel (1973).

The cost of electric defrost will depend on the size of the evaporator coil, and hence the total power of the heaters, and the length of time that the heaters are energised. There is no practical way of reducing the first cost factor for any specific size of installation, but good housekeeping and correctly set controls can minimise the latter. A significant proportion of the heat generated is lost to the cabinet air, which increases the system load.

Because electric defrost systems in most instances have heat applied external to the coil as compared with hot gas systems where the heat is applied internally and because of limitations on the amount of electrical heat that can be applied safely, electric defrost would normally require a longer defrost period, usually 1.5 or more times than for hot gas defrost systems, Crawford et. al. (1992).

Latent heat defrost - hot or cool refrigerant gas

Latent heat defrost, often called hot gas defrost, uses heat in the discharge gas from the compressor to defrost the evaporators. This is a method of defrosting which raises the temperature of the evaporator by passing through it a quantity of high pressure, high temperature refrigerant gas. In effect, during the defrost period, the evaporator operates in the same way as a condenser. Condensation of the hot gas into liquid in the coil releases the latent heat of condensation and this heat melts off the frost accumulated on the outside surface of the coil. The liquid refrigerant then rejoins the liquid line and is fed to the remaining evaporators on the system.

For this method to be successful, there must be an adequate supply of hot gas to the defrosting evaporator. Thus, there must also be sufficient suction vapour feeding the compressors. The effect of the defrosting process on the operation of the remaining 'cooling' part of the system must not be so large as to render it ineffective or unacceptably inefficient. Also, as the refrigerant liquid is required to flow back into the common liquid line and not accumulate in the evaporator, this method is used on installations with multiple evaporators each defrosting separately. The "rule of thumb" used in design is that the cooling capacity of any evaporator to be defrosted should not represent more than one third of the total system capacity, Harraghy (1987).

If latent heat defrost is used with a single evaporator system, a re-evaporator should be included in the system, to prevent liquid refrigerant from returning to the compressor. The particular means used to re-evaporate the liquid is the principal factor distinguishing one method of hot gas defrosting from another, Dossat (1991). Since in supermarket refrigeration systems hot gas defrosting is only implemented on multi-evaporator systems, re-evaporators are not normally necessary.

Hot gas defrost is potentially a fast defrost method since the heat is applied from inside the tubes and exerts a uniform melting effect. On large coils, supplementary electric heaters can be added to increase further the efficiency of defrost. A timer typically terminates the defrost cycle, although pressure/temperature termination can also be used.

In hot gas defrost the gas entering the evaporator is at a high temperature about 60°C and thus results in a rapid change of conditions in the suction piping adjacent to the evaporator and also in the evaporator's internal pipe-work. This has been known to cause thermally induced stress failures in the pipe-work and an alternative, more reliable, method known as the cool gas defrost was developed, Crawford et. al. (1992).

Cool gas defrost systems take saturated refrigerant gas from the top of the liquid receiver which is around 40°C, thus greatly reducing the temperature variation in the pipe-work and evaporator. There is less energy available in the refrigerant at this point, but this heat is sufficient to effectively defrost a display cabinet evaporator coil. As the majority of the heat applied to the evaporator during hot and cool gas defrost is contained within the fin block and not lost into the cooled space, these methods of defrosting should require a shorter defrost time than electric defrost, Walker et. al. (1990).

Work by Stoecker et. al. (1983) and Cole (1985) suggests that the higher the pressure in the evaporator during defrost, the faster the defrost; but the amount of energy needed to operate at high pressures may far outweigh the energy losses due to decreased system capacity from longer defrost periods. A report by the Electric Power Research Institute of the USA, (Walker et. al., 1990) suggests that the energy savings achieved by the use of lower condensing temperatures through floating head pressure control can be enhanced by maintaining the head pressure artificially high prior to the initiation of the defrost cycle.

Crawford et. al. (1992) pointed out that although hot gas defrost is the most popular in industrial applications, its energy requirements are not well understood. There have been no detailed experimental studies to investigate the factors that influence the length of the defrost period and determine the actual overall energy requirements on the refrigeration system or the refrigerated space. As both the hot and cool gas defrost methods require the evaporator to operate as a condenser, a number of precautions have to be taken when applying these methods. The evaporator has to be suitably designed and pressure tested for this type of application, as it will be subjected to high pressures during the defrost period.

Comparison between gas and electric defrost and design considerations

Kakol et. al. (1981) carried out experimental tests to determine the relative efficiency of basic hot gas and electric defrost. They observed that for electric defrost, heating of the finned coil was considerably less uniform than hot gas defrost. Some parts of the coil became overheated and others under heated and with the surface temperature of the heaters above 100°C, some melted water boiled away. However, the defrosting efficiency of the two methods of defrost was found to be approximately the same.

This result is a little surprising since hot gas defrost is generally thought to be more efficient because the heat transfer to the frost is via the primary and secondary surfaces of the cooler and the frost is therefore in the way of any heat being lost to the atmosphere. For electric defrost there is convective loss from the heaters to the atmosphere from that part of the heaters that is not in direct contact with the coil. Where the heaters are in direct contact with the fins there is also the additional factor of contact resistance at the junction between the heaters and the fins.

Sanders (1975) produced simplified one dimensional models for both hot gas and electric defrost of air coolers. For the electric defrost it was assumed that all the heat flow is by direct contact with the cooler metal and hence takes place from the cooler surface outwards towards the frost, (i.e. the same direction as for hot gas defrost). It was also assumed that electric defrost is characterised by a constant heat flux through the cooler surfaces which implies a variable temperature. Hot gas defrost was characterised by a constant temperature, pressure and heat transfer coefficient for the refrigerant within the coil, which implies a variable heat flux.

The calculation results showed that, for the hot gas model, an increase in thickness of the frost layer and/or a decrease in the defrosting medium temperature caused an increase in the actual defrosting time. This increase was however quite small when the frost layer was firmly adhered on the coil surface.

For the electric defrost model there was a sharp increase in defrosting time for a decrease in heat flux and defrosting times were generally greater than for hot gas defrost.

Stoecker et. al. (1983) considered the energy utilisation in hot gas defrosting of industrial refrigeration coils. From laboratory tests on an R-22 system and field tests on an ammonia system some features were identified that could reduce the energy requirements associated with hot-gas defrost. There were namely the minimum hot-gas pressures that could still achieve satisfactory defrost, the method of draining of condensed liquid refrigerant from the coil, and the defrost termination time.

For electric defrost, the main design parameters are the positioning and sizing of the electric resistance heaters. In small air coolers, the most common positioning of the heaters is on the face of the fin whereas larger coils have the heaters placed in dummy tubes Niederer (1975), Segal (1975) and Dowrani (1991).

For hot gas and reverse cycle defrost there are two main considerations. Firstly, there should be a big enough cooling load on the compressor so that sufficient discharge hot gas can be supplied to the defrosting cooler and secondly, that refrigerant which does condense in the coil is removed quickly so that it does not impede heat transfer from the coil to the ice during the defrosting process.

From the available literature it appears that the electric defrost method is widely used in small and intermediate sized plants, while the hot and cool gas defrost are used particularly in larger plants. Both methods have some advantages and associated drawbacks.

In general, the advantages of electric defrost are its lower capital cost, simpler installation and lower maintenance requirements compared to hot gas defrost. Its main disadvantage is that it takes longer to defrost and adds additional heat to the refrigerated space. This additional heat will have to be removed by the refrigeration equipment leading to higher overall electrical energy requirements. Failure of the electric resistance heaters and corrosion may cause problems including a higher fire risk than hot gas defrost, Mickan (1980).

Hot gas defrost is considered to be more efficient than electric defrost because the defrosting process is very rapid. Recovery after defrost is also rapid because there is less build up of heat in the coil during defrost. It also provides an internal cleansing of the tubes and positive oil return to the compressor. Hot gas defrost can be more economical in operation than electric defrost because energy is more usefully applied by using the heat normally discarded at the condenser and by inputting less heat to the space. Risk of food loss in the event of failure of a defrost termination controller is substantially reduced as no excessive heat is generated during the process, Sanders (1975).

Despite the advantages of hot gas defrost, electric defrost is more popular because fewer systems are thought to lend themselves to the hot gas method. Also, hot gas is considered to require more expertise in design and operation and in general has a higher capital cost than electric defrost.

The case for hot gas defrost is also not helped by the lack of detailed information regarding its performance compared to electric defrost, and of proper guide-lines for design other than experience.

Methods of defrost control

The defrost controls are very important for successful defrosting because they dictate the exact moment at which defrosting starts and the moment at which it will be terminated. Defrost controllers must be reliable since failure to defrost will lead to low coil surface temperatures, poor efficiency and in some cases complete blockage of the coil.

Theoretically, a refrigerator defrost system should only defrost when sufficient frost has accumulated on its evaporator to cause significant degradation of performance. Many schemes have been attempted over the years to sense the evaporator frost accumulation directly and initiate a defrost at an appropriate interval.

Defrost initiation
The commercial refrigeration industry has in general adopted a timing system to initiate defrost and even though this method is not optimum, it does serve the prescribed function of defrosting and keeping the evaporator free of frost, Harraghy (1987). Normally, electro-mechanically or electronically operated timing devices are used to initiate the defrost cycle at predetermined time intervals which depend upon the size and application of the refrigeration system. Time defrost initiation is very popular because it has low cost, it is simple to install and easy to maintain. The drawback with this method of defrost initiation, however, is high energy consumption during operation under low frost accumulation conditions.

Demand defrost
Theoretically, better energy efficiency can be achieved if the interval between defrost cycles is varied according to actual frost conditions on the evaporator. If excessive frost is allowed to accumulate on the evaporator, the heat transfer characteristics are significantly altered and the efficiency of the system deteriorates appreciably. On the other hand, optimal efficiency cannot be realised if defrost is initiated more frequently than necessary. Accordingly, automatic defrost systems should ideally vary the interval between defrosting operations according to actual need, initiating defrost only when an "optimal" amount of frost has accumulated on the evaporator coil. Such controls are generally termed "demand defrost" systems. Demand defrost controls usually incorporate some means of sensing the actual frost build-up or the accompanying loss of heat transfer capability, Eckman (1987). Successful implementation of such a system should result in energy savings with no decrease in performance. In refrigerated food applications, better food quality should also be maintained, Heinzen (1988).

<u>Adaptive defrost with Timer:</u> To improve on fixed time interval defrost, Healy (1962), Muller (1975) and Parken (1977) determined experimentally the optimum number of defrost cycles based on the actual environmental conditions and equipment configuration. The experimental

approaches used, however, were system specific and cannot be reliably used universally. Heinzen (1988) followed a similar approach in implementing defrost on demand for refrigerated display cabinets. In his defrost strategy the optimal defrost time (cooling time) was pre-determined for the particular refrigerated display cabinet and programmed into the controller. The cooling time was then adjusted on line as a function of the optimal defrost time, the previous frost accumulation time and the previous actual defrost time. A simple microprocessor based controller was used to implement this defrost strategy. Field test results demonstrated the ability of adaptive defrost control to adjust and compensate for the seasonal changes in humidity, while maintaining the integrity of the system's overall efficiency. A significant limitation of this system, however, is the requirement to determine an optimal defrost period for the particular system which makes the defrost strategy system specific.

Air Pressure Sensor - Air differential pressure device: With the build up of ice, the air passage through the coil reduces in area and hence the air pressure drop across the coil increases. The pressure drop can be measured by pressure tappings at the upstream and downstream face of the coil. When the differential air pressure exceeds a pre-set value, a pressure switch senses it and initiates defrost. This method of defrost initiation was investigated by West, (1976), Ciricillo, (1985) and Eckman, (1987). According to Tantakitti (1986) this method can be more efficient than time defrost but problems can occur by partial coil blockage with debris and varying air pressure due to wind, doors left open etc. Behrens (1976) recommended that the system can be made reliable by incorporating a timer override system which would eliminate false pressure signals.

Combined air temperature difference with air flow: Muller (1975) proposed the combination of air temperature and air flow measurements across the coil as a reliable and practical method for initiating the defrost cycle. Due to reduced air flow, the temperature difference across the coil increases. The air temperature difference is a direct scale for measuring the reduction in efficiency caused by the build-up of frost, and therefore can be used to determine the optimal point in time for defrosting. According to the investigator, the ambient temperature, air humidity and the degree of frost accumulation do not influence the optimum air temperature difference for defrost initiation. Also, the disruptions such as fan failure, opening of doors and interruptions in power supply, can be sensed by the change in the air flow rate, so the danger of premature initiation of the defrosting process is small. The problem with this method of defrost initiation is the difficulty in the accurate measurement of air flow through the coil.

Evaporator temperature and pressure: Frost on the evaporator coil causes a decrease in the temperature and pressure of the refrigerant and this characteristic can be used to initiate defrosting. As the pressure inside the coil also varies with ambient temperature, the use of a pressure sensor in combination with a timer can improve defrost initiation control, Dowrani (1991).

Measurement of thermal conductivity of frost using the temperature difference between the air and refrigerant/cooler metal surface: Apart from initiating defrost by monitoring the operating parameters, defrost can also be initiated by measuring frost deposit thickness or the thermal insulation effect of the frost layer. Llewelyn (1984), Ciricillo (1985) and Buick (1978) proposed several unconventional defrost initiation systems based on the direct measurement of frost thickness. According to Llewelyn (1984), the important parameter to monitor in a defrost control method is not the ice deposit thickness but the thermal insulation effect of the ice layer. His research focused on the thermal conductivity approach and developed an automatic defrost on demand control system named DD5. DD5 initiated a defrost when the thermal insulation of the ice layer on the evaporator reached a predetermined value and terminated it when the ice had cleared. Two identical temperature sensors were used along with a comparator amplifier. One sensor was in intimate contact with the evaporator coil and the other was positioned in the outlet air stream from the evaporator unit. With the build up of frost on the evaporator three separate effects occur. Namely, the heat transfer efficiency of the system reduces, the mean evaporator temperature reduces and a temperature gradient develops through the ice layer with the coldest point on the evaporator coil moving from its frost free position towards the low refrigerant pressure end of the coil. The decrease in heat removed from the refrigerated area with time results in an increase in the off coil air temperature and reduction in refrigerant temperature. This increasing temperature difference was utilised by DD5 to initiate the defrost cycle. The same method was tried by Buick et. al.

(1978) who also evaluated the use of different temperature measurements to initiate defrost. A slight deviation from this method was proposed by West (1976), who utilised the temperature difference between the evaporator surface and the ambient air instead of the evaporator air off temperature to initiate defrost. The disadvantage with this method of defrost is that the temperature difference between the refrigerant and air can also vary with the ambient temperature, fouling of the coil surface and variation in the air flow rate through the coil, Buick et. al.. (1978) and Tantakitti (1986). To overcome these problems this technique is usually used in combination with an override timer.

Monitoring the capacity of an Electronic plate capacitor installed on the evaporator coil: This defrost initiation method was based on the fact that the dielectric constant of ice differs from the air it displaces, and a signal proportional to the ice thickness can be derived, Buick et. al. (1978) and Llewelyn (1984). After several trials this technique was discarded because of problems caused by the variability of ice, sensitivity of the signal, critical positioning of the plates between which the frost build up was to occur and the need for high gain amplifiers.

Monitoring the resonant frequency of an acoustic oscillator installed in the evaporator unit: Since the resonant frequency is a function of mass and compliance, a marked shift in frequency occurs with deposition of frost on the coil, Llewelyn (1984). Although this method gave reliable and predictable results its practical application proved not to be feasible due to unacceptably high unit cost.

Measurement of frost thickness using infra-red detectors: The Torry Research Station devised a frost sensor consisting of an infra-red emitter directed at the edge of the cooler fin, Pearson (1986). A photocell placed close to the infra-red emitter picks up the radiation reflected from the frost on the fin and sends a voltage signal to the controller. Defrost is initiated when the electrical signal reaches a predetermined value, Pearson (1986). Woodley (1989) also used an infra-red detector to initiate defrosting. This system employs a patented infra-red detector firmly attached to the evaporator of the refrigeration system just below an infra-red emitter. The detector is in the form of an adjustable screw head and the screw can be adjusted to suit the size of the evaporator. The length of the defrost time is regulated by the controller and can be pre-set from 8 to 40 minutes to match the frost thickness. The two significant differences between the methods presented by the Torry Research Station and Woodley is the sensor position and defrost termination method. The Torry sensor is fitted at the edge of the cooler fin but the Woodley sensor is attached to a pipe between the fins on the air input side of the evaporator. In the Torry method, defrost is terminated by a refrigerant pressure differential sensor, whereas the Woodley method utilises a timer. The infrared sensors have been applied in industrial refrigeration systems but have not found widespread application in smaller systems because of the sensitivity of sensor positioning on the coil.

Measurement of frost thickness using a fibre-optic device: Paone and Rossi (1991) used a fibre-optic device to sense ice formation on the evaporator coil. This was achieved by transmitting light from a photo-diode through a fibre-optic probe to a fibre-optic receiver. The receiver was arranged in a way that it could either pick up the light reflected by the ice once the ice thickness increased to a given value or sense the interruption of light transmission through the coil. With the latter arrangement the light source was mounted on one side of the coil and the receiver on the other side. The experimental investigations showed that the transmission type performed better than the reflection type as the sensors suffered from intrinsic sensitivity to variations of ice reflection coefficient arising from changes on the frost density and structure. The high unit cost of the device hindered its use in the commercial world.

Use of Fuzzy Logic to implement defrost: Itoh et. al. (1995) monitored the difference between the air temperature at the exit of the evaporator and the inner duct (also, known as the back panel) of a display cabinet, along with environmental temperature and humidity, cooling time and defrosting time during the previous defrost to develop a thirty three rule based fuzzy logic algorithm. They showed that with fuzzy logic the controller could successfully discriminate between normal and abnormal frosting and also judge whether the system needed defrosting. During field tests they also showed that use of only the temperature difference to initiate defrost was inadequate to provide an accurate judgement regarding optimum defrost initiation. Other information was also required by the controller to facilitate the decision process. The authors did not indicate any drawbacks with this method of control but it is felt that it would require more than a simple processor to undertake the evaluation process involved in the implementation of fuzzy logic control.

Other demand defrost control systems currently available for centrally controlled retail food refrigeration systems are based on: a) determining the optimum cooling period prior to defrost based on the length of preceding defrost cycles (Danfoss, 1999) and b) determining the requirement for defrost from coil parameters such as the degree of superheat and the cycling rate of pulsed expansion valves (JTL Systems, 2000). Work at Brunel University which is funded by the Ministry of Agriculture, Fisheries and Food (MAFF) is aimed at the development of a demand defrost method based on Artificial Intelligence Techniques. When fully developed the defrost control system should be self-adaptive and non-cabinet and refrigeration system specific (Datta and Tassou, 1999, Tassou et. al., 2000)

Defrost termination

It is claimed that the defrost cycle can use as much as 30% of the operating energy of a refrigeration system, Joseph, (1985). Traditionally, defrost termination has been performed on a fixed time basis or by sensing a number of parameters such as the temperature rise of the air at the coil outlet, the temperature rise of the coil surface or the rise in the evaporator pressure.

A number of authors considered the use of variable defrost time but concluded that the amount of time required to completely defrost the coil under different operating conditions was difficult to predict accurately due to the non-homogeneous nature and complexity of frost formation on the coil and the transient nature of the defrost cycle, Bonne (1980), Heinzen (1988) and Pace (1975). The defrost time is also a function of the size of the evaporator coil.

Kuwahara et. al. (1986) examined the effect of early defrost initiation and increasing the power input to the compressor during defrost on the duration of defrost. Using these modifications they found that they could reduce the duration of defrost of a conventional heat pump from approximately 5 minutes to 2 minutes. They also tried to reduce the water left on the fin surface after defrost by applying a surface active agent to the fins to inhibit water droplets from clinging to them. This left the fins drier on completion of the defrost cycle and discouraged frosting to a certain extent.

Harraghy (1987) and Yoshlake (1987) concluded that sensing the air temperature near the evaporator could provide adequate control for defrost termination. The optimum position for the evaporator temperature sensing points, however, can vary depending on the geometry of the coil and the investigators found it difficult to find a suitable location for the sensor to ensure that all parts of the evaporator were free from ice when the defrost cycle was terminated. An appropriate solution to this problem could be the use of a number of temperature sensors located at different places over the evaporator coil.

Requirements for supermarket refrigeration systems

A number of different refrigeration system arrangements can be employed in retail food stores. The predominant types, however, are the remote and integral. In the remote arrangement the evaporators of the display cabinets are served by a centralised multi-compressor refrigeration system which is located remotely in a plant room. This arrangement offers a number of advantages which include centralised maintenance, reduced noise in the store environment and higher efficiency. Disadvantages include the large quantity of refrigerant in the distribution piping and the associated risk of leakage, and reduced flexibility in store arrangement. In the integral arrangement, the display cabinets are self-contained and incorporate all the refrigeration components within the cabinet. These components normally include a compressor, an air-cooled condensing coil, an expansion device and the evaporator coil. The main advantages of integral display cabinets are increased flexibility and reduced refrigerant inventory compared to the remote cabinets.

The most commonly used defrost methods in refrigerated display cabinets and other retail food refrigeration systems, are electric defrost, hot or cool gas defrost and off-cycle defrost. Off cycle defrost can be implemented on both integral and remote multi-compressor refrigeration systems. Remote systems also allow both hot and cool gas defrost techniques to be implemented. Multiple compressor refrigeration systems supply refrigerant liquid to a number of display cabinets which are piped in parallel. For this reason, the number of display cabinets which can be defrosted simultaneously is limited to avoid starvation of the compressors and system shut-down due to low suction pressure. For electric defrost it is also desirable to limit the number of cabinets that are defrosted simultaneously to reduce maximum demand.

In retail food stores in the UK, common practice is to initiate defrosting of both integral and remote cabinets on a time basis with the controller/timer set to provide a fixed number of defrost cycles per 24 hour period, normally 4 for medium and high temperature cabinets and 2 for frozen food cabinets. Defrost is normally terminated when a fixed defrost time has elapsed or when the cabinet evaporator air-off temperature has reached a pre-set value, whichever is sooner.

Since defrosting involves the application of heat to the coil to melt the ice, it penalises refrigeration system performance due to the fact that during the defrosting process energy is used while producing no useful cooling. Furthermore, during frosting and defrosting the cabinet and thus the product temperatures rise above the set limits for normal operation. It is estimated that the energy associated with defrost and anti-sweat heaters in a supermarket can be significant and depending of store size, humidity levels and cabinet mix, may exceed 30% of the total refrigeration system electrical energy requirements.

Research both in the UK and the USA has shown that the relative humidity in retail food stores will vary during the year with relative humidity in the summer being considerably higher than the relative humidity in winter. Condensate measurements during defrost have also shown that frost accumulation on the coil in winter is much less than frost accumulation in the summer indicating that a lower defrost frequency can be employed in winter without adversely affecting the temperature control of the cabinets (Tassou et. al., 2000).

In retail food stores where central monitoring and control systems are employed, demand defrost controls can be employed without significant additional cost. These can vary from simple controllers that vary the defrost frequency based on a single or a combination of parameters, such as humidity or dew point temperature of the air, to more sophisticated methods based on predictive controls. Work carried out at Brunel University which is funded by the Ministry of Agriculture, Fisheries and Food (MAFF) is aimed at the development of a demand defrost method based on Artificial Intelligence Techniques. When fully developed the defrost control system will be self-adaptive and non-cabinet and refrigeration system specific. Although it would not be particularly difficult to apply such a system to centrally controlled remote cabinets, the challenge will be to develop cost effective demand defrost control systems for integral cabinets employing their own individual controls.

Concluding remarks

1. The process of frost formation on evaporator coils has been researched extensively, but in the majority of cases the investigations were carried out on specific oil designs and flow arrangements. Although the published results are of interest in that they provide an indication of the influence of various design parameters on frost formation on evaporator coils they are of limited use to coil and display cabinet manufacturers that employ proprietary coil designs.
2. Only limited information has been published on the performance of display cabinet evaporator coils under frosting conditions in `real' supermarket environments. The control of the coil would have a significant effect on the rate of frost formation and defrost requirements, and studies on the influence of evaporator coil control on frost formation and the overall performance of the system will be of interest.
3. Individual evaporator coil manufacturers have their own individual approaches to coil design and would tend to standardise on tube sizes, fin types and heat transfer enhancements to reduce manufacturing costs. The trend in cabinet design would, however, be to maximise the performance of the coils through heat transfer enhancements and the use of larger coils, where possible, to reduce the difference between product and air temperature, reduce the rate of frost formation and improve the efficiency of the refrigeration systems.
4. Although it may be possible to use off-cycle defrost in certain applications by using larger and more efficient coils, wider application of the concept in high value product

cabinets would require more sophisticated predictive controls and better understanding of the influence of design and operating and control parameters on the rate of frost formation and defrosting of the coil. More experimental and theoretical work is required in this area.

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Report I

Refrigeration Controls - Honeywell & EA Technology

Introduction

A refrigeration control system is usually selected on the following priorities:
1. Capital investment levels.
2. Optimised plant operation, maintenance and energy costs.
3. Service requirements.
4. Retail 'down time'- on failure
5. Diligence Fulfilment.

This report covers the development of refrigeration controls for distributed refrigeration systems. It initially gives a very brief description of traditional controls; then gives a description of a fully integrated control system and finally looks at some of the intermediate steps along the way.

Traditional control

Traditional control separates the cabinet control from the control of the refrigeration pack or central compressor/condenser system.

The cabinet control is via a TEV to control flow into the evaporator and either a liquid line solenoid valve or an evaporator pressure regulator to control the cabinet temperature. EPRs are favoured if a wide range of evaporating temperatures are required by cabinets connected to a single refrigeration pack. Defrost is usually operated by timers and there is no control over the trim heaters.

The pack control operates to ensure that the evaporator pressure is maintained within a control band by sequencing compressors (for multi compressor packs) or by controlling the capacity of larger compressors, e.g. adjusting the slide valves on, or speed of rotation of, screw compressors.

An additional feature is that the condensing (or head) pressure may be held at a higher level than the minimum that the condenser could provide to ensure that the liquid pressure at the TEVs is high enough to allow sufficient refrigerant to pass through the TEV under all ambient conditions.

Refrigeration control – the ideal system

The best refrigeration controls should provide an integrated control system with handshaking between all components, predictive and self-learning control, trend mapping, and integration of all known variables to allow management of the design requirements to deliver the most energy efficient and robust system.

Integrated control is best provided via protocols (notably Lonworks) which allow interoperability of distributed intelligence. This has been adopted by major BEMS providers but is not openly available in all areas of refrigeration control and though a number of systems present themselves as Lonworks gateways, the ethos of LON can only be considered as early evolutionary and some way behind the more conventional building service systems that are widely available. Until interoperability becomes widespread control system selection may have to be made from suppliers whose systems will communicate with each other.

Communications links based on an RS 485 platform are commonplace across the refrigeration control industry with, at present, most systems based on proprietary protocols. Most of these protocols have some allowance to accept limited third party controls though, generally speaking, where they have this flexibility it is to a reduced control set. This makes cross boundary selection of components difficult where achieving the overall control strategy is the most significant aspect. The 'trade-off' aspect is that some control flexibility is lost to achieve a split supply base.

The refrigeration control strategy of any system has to retain, as its primary function, the delivery of refrigeration operation and from that starting point manipulate system performance and variables to achieve the best available operation and efficiency to match the agreed priority profile.

The control strategy has to include a natural inclination to easing delivery of refrigeration in the event of any failure thus protecting the primary function. System failure modes, default settings and backup systems all have settings to protect this objective.

In looking at the best available complete system the individual components are considered first and then how they interact to form the overall system.

Cabinet control
Individual refrigeration cabinets are generally deemed to warrant individual cabinet control.

Electrically driven expansion valves allow some ongoing non-intrusive adjustment post initial installation and arguably allow for closer superheat control than the more conventional T.E.V.'s. Minimum superheats ensure the maximum use of the evaporator surface bringing supplementary benefits including equalising ice build up across a coil, providing a more even air off the coil temperature and improved evaporator performance. Reduced De-Hydration of open product is a by-product of higher evaporating temperatures and with an option for variation of cabinet fan / air velocity combinations further gains are available.

Standard cabinet design at ISO ambient conditions sets extraction rates and evaporating temperatures. This is rarely the natural operating criteria in the UK. E.E.V's allow also for the details of their performance levels and achievements to be used within a wider control algorithm in deciding on the operating levels required of central plant: if all cases supported by a central refrigeration plant can be satisfied at this higher evaporating condition then the plant can be allowed to operate at this condition (less the suction line pressure drop) thus achieving significant energy savings.

Control of the cabinet defrost cycle potentially offers energy savings, in forced defrost systems, where omitting defrosts that are not required saves the energy of both defrost and subsequent plant pull down. Self-learning 'defrost on demand' systems are likely in the near future by trend mapping the cabinet air and / or coil temperatures. Current 'on demand' systems monitor a combination of the air temperature or coil temperature rise time to omit defrosts.

Cabinet trim heaters form a large proportion of cabinet electrical loads, particularly on Full Height Glass Door (FHGD) LT cases. By controlling them on / off sizeable energy savings are available. The ambient conditions are critical to prevent icing around trims though relatively low cost and simple stand-alone controls can manage this aspect for limited cost.

Refrigeration plant
The varying suction pressure requirements now need can be managed by allowing rates of downward adjustments, to meet refrigeration load, to be far more rapid than any allowed suction pressure increase. The minimum set point level should be as necessary to deliver the conditions for ISO performance at the case.

Allowing the compression ratio of the plant to be minimised can further increase the effective refrigeration capacity of this central plant. If the available heat exchanger surface in the condensing system is fully utilised this ratio can be minimised, effective plant capacity is increased and the number of compressors required to run is reduced.

Control logic to allow the condensing pressure/temperature relationship to sit at a normal level above the ambient conditions around the condenser will ensure sufficient sub cooling to ensure a positive liquid flow through the system and thus maintain the primary system function.

Control of condenser capacity can be made through control of the % of condenser used or by fan speed control. Managing the complexity and measuring the gains through additional complexity is significant here.

Example – Based on a 100KW Semi Hermetic HT pack selection

Operating Conditions	**40 ^{O}C – Cond. -10 ^{O}C – S.S.T.**	**30 ^{O}C – Cond. -10 ^{O}C – S.S.T.**	**40 ^{O}C – Cond. -5 ^{O}C – S.S.T.**	**30 ^{O}C – Cond. -5 ^{O}C – S.S.T.**
Refrigeration Capacity	**102KW**	**122KW (+19.8%)**	**130KW (+26.5%)**	**153KW (+50%)**
Compressor run current savings on a fixed 100KW criteria	**0**	**-25%**	**-13%**	**-36%**

This system now has a floating suction pressure, floating head pressure central plant arrangement supporting a variable load characteristic in its cabinet/cold-room performance. As the bandwidth of operation narrows the management of control needs to be fixed as variations in any of the associated components can erase the benefits that controls sophistication has delivered.

In systems of this nature the overall control philosophy has to be accepted by all parties as the control becomes critical to the plant operation. The comfort of built in redundancy that may at present mask commissioning deficiencies has, by narrowing the operating bandwidth, been removed in design.

Individual cabinet / plant combinations
Where individual close coupled plant / load combinations are used the control strategy has to be very different in that any controls sophistication has to be measured against the mechanical constraints of the system. Where the plant is selected to meet all operating conditions it will generically have an amount of redundancy in it's sizing for the operation over the year. The running time of the unit needs to be long enough to prevent short cycling therefore a wider operating bandwidth is needed.

De-Specification

The range of available controls between the outlined high end system and the traditional mechanical switching is wide with a number of nominal break points. Where the initial capital costs are a priority the selection of controls can be matched to the costs limitations with some reduction from the overall controls strategy.

- Very basic cabinet electronics deliver close temperature control and 'tuned' defrost termination, stand alone trim heat control (Austrian tests indicate a two year payback), the control system for the central refrigeration pack giving floating head pressure (no additional cost in some higher end systems)

- A) Basic communicating electronics allow central data gathering for diligence and monitoring.

- B) Proprietary electronics allow 2 way communication and adjustment, limiting 'sales floor' intervention.

- Integrated systems

  Options A and B ensure that initial infrastructure is in place that can be developed towards the top end system described.

Additional benefits

Having introduced a data communications back bone as the basis of the control strategy with a graphic user interface that delivers 'live' operational detail in graphical form the opportunity then exists to use this as the central control system for the HVAC system.

With the BMS industry moving towards 'fully open' systems the reverse opportunity may be available with some cross utilisation of components such as temperature and humidity sensing. These 'extended' systems are available and should form part of a wider evaluation, looking at the relative costs benefits of moving to a single system in either direction.

Central monitoring
Most store operations are currently run as 'islands' in that there is a store management procedure for the reporting and actioning of faults. Systems are widely available that allow remote access to monitor and adjust systems though proprietary software is generally required to facilitate this.

There is widespread development work throughout the industry to utilise the power of the Internet to allow for access of systems that will require access level control but remove the need for any specific software at the interrogating station. This is well developed for Lonworks based systems.

Commercial reasoning
The current refrigeration control systems generally require a single point of supply throughout the system to get the best available control with some moves towards development of open systems. At present different systems require a differing approach to their installation that is dictated by the format their development has taken and the communications protocols on which they are based.

A wider BMS market has the volume and cost platform to deliver R&D that allows interoperability to be more advanced. The integration of refrigeration controls into this sphere will require commercial pressure to be applied, but will allow lower cost systems for integration into the overall store BEMs system.

The likely cost of installing an advanced system with full communication over an existing system is likely to be of the order of £10,000 to £20,000 for the communication wiring with slight increases in hardware costs for a 4,000m^2 supermarket. Other costs will be the front end GUI and the remote access system. These costs are difficult to identify and need further investigation. If Lon becomes a standard in refrigeration control, as it is becoming in BMS control, then these costs are likely to fall.

Report J

Case/Store Interaction - Food Refrigeration and Process Engineering Research Centre, University of Bristol

Introduction

Refrigerated display cabinets play an essential part in food conservation, and are required to offer first, adequate temperature control (i.e. good quality of refrigerated food at point of sale) and secondly an attractive eye appeal for customers (Baleo et al, 1995).

Refrigerated display cabinets can be either opened or closed devices. In the first instance a door is used to separate customer from product, in the second direct access (no physical barrier) is provided. For marketing reasons the direct access (or open) cabinet is preferable. It is these cabinets which provide the greatest interaction with the store.

The needs of the customer to be able to view and handle the product unhindered before purchasing presents technical problems. The air curtain, which should provide a thermal but not physical barrier between the product and customer, is not perfect. Warm moist air from inside the store is entrained into the air curtain and mixed with the cold air, which is then drawn back through the cabinet's refrigeration system, providing a high thermal and latent load on the cabinet. The high energy consumption and large number of these cabinets in larger stores has an enormous effect on the store environment. Refrigeration costs for the food retail trade were estimated at £380m per annum (in 1998) (equivalent to 890 MW, i.e. two thirds of the power output from a power station such as Hinkley B) (Evans et al, 1998).

The humidity of the room in which the cabinet operates has a significant effect on the cabinet. The latent load is a significant proportion of the total load, 15-20% for a store operating at the correct humidity (ASHRAE, 1995). This can increase dramatically if store humidity increases. Higher humidity leads to more defrosts which require energy and raises product temperatures.

Cold air from the cabinet also falls from the cabinet onto the floor causing the customers' feet to become cold; this is commonly known as the 'cold feet' effect. This effect also removes heat from the store, 80 to 90% of the heat removed from the store by vertical refrigerators is absorbed by the display opening (ASHREA, 1995). Open cabinets therefore act as large air coolers, absorbing heat from the store and rejecting it via the condensers outside the building.

Open frozen-food display cabinets are probably the weakest link in the frozen food chain, since the temperature of the goods in the cabinet are affected by several factors, particularly the ambient conditions. Slight changes in room temperature, and particularly in the air circulation and radiative heat flux, cause considerable differences in the temperature of the load (Bobbo et al, 1995). Factors such as the position of the cabinet within a store, loading patterns, store conditions, food packaging etc. all influence food temperatures (James, 1993).

The current technology as normally applied

The air flow in refrigerated display cabinets largely determines the correct operation of the cabinet (Van Oort (1995): air absorbs heat from the product and takes this to the refrigeration system where heat is withdrawn and then sent back again to cool the food. The uniformity of air flow is important in order to create uniform temperature control. Optimisation of the air flow is often carried out by experience and 'rules of thumb'. 'Tried and tested' cabinets are manufactured and then modified to optimise for the particular conditions. Measurements of air velocity and temperature on a display cabinet is a timely exercise, as ambient conditions cause fluctuations in temperature and air velocities, and any changes require waiting for thermal equilibrium before measurements can be carried out.

Supermarket chains expect refrigerated display cabinets to comply with either British Standard EN441 or their own standards or a combination of the two. A cabinet which operated satisfactorily during the standard, however, may not operate satisfactorily in a supermarket, where the ambient conditions are not 'standard'.

EN441 defines that cabinets are tested with air movement parallel to the plane of the cabinet display opening and to the longitudinal axis at a speed of 0.2 +/- 0.1 m/s. If the cabinet is located near a door or a ventilation duct, air movement may be above 0.2 m/s and perpendicular rather than parallel to the plane of the display opening. Bobbo et al (1995) showed that by increasing the air flow in the test room from 0.15 to 0.28 m/s (both within EN441), completely different product temperatures are measured.

EN441 defines the temperature gradient in test rooms, this is measured before the test is put into operation. The test room temperature may vary from floor to ceiling but the vertical temperature gradient shall not exceed 2^{o}C/m and there shall not be a difference of more than 6^{o}C between the temperature measured at the floor and at the ceiling. This temperature gradient will increase dramatically when the test is started, and will vary depending on the geometry and air exchange rate of the test room. A higher temperature gradient will help the cabinet perform better.

EN441 defines that cabinets are loaded with packs so that they are full and a standard shelf configuration is used. In a supermarket environment shelves are often somewhere between full and empty. Product can also be straddling any load line and supermarkets can change the shelf configuration. Computational fluid dynamics (CFD) modelling by FRPERC (Foster, 1997) has predicted that these factors greatly affect the performance of the cabinet.

As cabinets are designed to meet these standards, by evolution they have been designed to work well in a test room and not designed to work well in a supermarket environment.

Designers of heat and ventilation of supermarket stores must take into account the cooling load of the display cabinets as this load can be greater than the design air-conditioning capacity of the store. Heat is being removed day and night, summer and winter regardless of the store temperature. As an example ASHRAE (1995) calculated that a store designed to operate at 24^{o}C will be cooled by almost 6^{o}C by the refrigerated display cabinets. The designers will take the cooling load of the display cabinets and subtract this from the total building air conditioning requirements. This often means that the stores work in heating mode more often than expected, even in the summer. With the benefit of a reduction in cooling load, also goes the disadvantage of an increase in heating load.

If an overall heating or cooling is delivered to the store based on one temperature sensor, then depending on the position of the sensor, either the refrigerated aisles will be too cold or the ambient aisles will be too hot. It is more sensible to have different heating/cooling regimes in different areas of the store.

Some stores apply a cooling or heating load to the ambient aisle depending on temperature and always provide a heating load to the chilled aisles. Other stores do not add heat to the cold aisles but instead extract or mix the cold air which the cabinets produce. Even with these systems temperatures between refrigerated aisles and ambient aisles can vary dramatically (Foster et al, 1998*)*.

Cooling from the refrigeration equipment does not preclude the need for air conditioning. On the contrary, it increases the need for humidity control. With increases in store humidity, refrigerated cabinet performance is reduced. For this reason store humidity should be kept below 55%RH (ASHRAE, 1995).

To save energy usage of the stores and help with temperature control of the display cabinets, some stores use night covers (blinds). These are blinds which are pulled down from the top face of the cabinet to the bottom when the store is closed. These blinds reduce mixing between the cabinet air and the store air. The main drawback with blinds are that they hinder the shelf stackers, who usually leave them up during the period of shelf stacking.

The leading practise as currently being tested in trials

Due to the problems with EN441 as explained above, a new standard is being tested (prEN 441 (experimental)). The same cabinets are being tested to EN441 standard at a number of different organisations. Key dimensions of the cabinet in relation to the room will be measured at each measurement site. The test rooms at the different organisations are different shapes and sizes. At the end of the tests an analysis will be carried out on the test results to establish whether the test results depends on the test room it is tested in, and if so, which dimensions are significant.

There are benefits to the performance of refrigerated cabinets by running stores at even lower humidity levels. There are two types of dehumidification system, electric vapour compression air conditioning and desiccant dehumidification. Desiccant systems can maintain humidity lower than vapour compression, 35%RH is claimed by Munters Superaire®. Test by Howell (1993) showed that by reducing store RH from 55% to 35%, the majority of display cases saved 20 to 30% in compressor energy, 40 to 60% in defrost energy, and 19 to 73% in ant-sweat heater operation. At the same time, the operating costs for the store air-conditioning increased by 4 to 8%.

Removing the cold air near the floor in refrigerated aisles is difficult when almost all supermarket ventilation systems extract from high level. A method currently being used at two branches of Marks and Spencer use a Hiross (Advanced Ergonomic Technologies (AET)) access floor ventilation system, which recovers cold air from floor level of refrigerated aisles. This cold air is then returned to the stores main air conditioning system and, AET claims, reduces the load on chillers and saves energy.

The emerging practice that might be expected to be commercially applicable in 2-5 years time

Air cycle refrigeration may become widely used in supermarkets, as air as a refrigerant has a number of advantages over traditional refrigerants. With air as the refrigerant, leaks do not cause a problem, environmentally and financially, also connecting and disconnecting cabinets to air lines is far easier than connecting to refrigerant lines (Russell et al, 1998).

A traditional retail display cabinet has the same amount of air leaving the cabinet as entering the cabinet (there is know net mass transfer between cabinet and store). An air cycle cabinet has cold air entering the cabinet from a remote source and therefore there is a net transfer of air from the cabinet to the store. This means that even if there was no entrainment between the cabinet's air curtain and the ambient of the store, there will still be cold air leaving the cabinet causing cold spillage. If the source of air for the air cycle system is not inside, but instead outside of the store, the store will be given a positive pressure of cold air which will have to be removed from the store. This cold air will require an extra heating load, which will need to be catered for by the stores air conditioning system, or from the heat generated by the air cycle unit.

Availability of models that might be used to help assess the performance of these technologies in the model stores

The temperature of products within refrigerated display cabinets are mainly governed by the movement of air around the cabinet. For this reason any numerical modelling of heat transfer in a retail display cabinet needs to predict this air movement. Computational fluid dynamics (CFD) modelling allows the prediction of fluid movement and heat transfer due to diffusion and convection, as well as other processes and is therefore the primary tool for numerically modelling these systems.

CFD modelling allows predictions of velocities and temperatures at an early stage in the design process of refrigerated display cabinets (Van Oert et al, 1995). Measurements can only be adequately carried out in test room conditions, as control of the temperature and air flow outside the cabinet is imperative to getting consistent results. CFD is therefore a useful tool to test the effects of different store conditions on the cabinet's performance.

The drawback of the use of CFD is that the user of CFD needs a thorough knowledge of fluid dynamics processes. Too many environmental factors act simultaneously and affect each other, and therefore important assumptions must be made to simply the solution of this problem (Schiesaro et al, 1999). For this reason, the model must always be validated by experimental test. The CFD user also needs to know boundary conditions to input in into the model.

There are many papers detailing the modelling of retail display cabinets using CFD Van Oort et al (1995) showed excellent agreement between calculations and experiment. Bobbo et al (1995) showed that the temperature of goods in the cabinet is affected by several factors, particularly the ambient conditions. Baleo et al (1995) showed that qualitative results are often identical or very similar, however the quantitative comparison sometimes show several differences. Schiesaro et al (1999) found CFD extremely useful in optimising air curtains, markedly reducing the time and also the cost, involved in the experimental trials.

Foster et al (1998) used a 3 dimensional model of an entire supermarket store to predict the effects of cold spillage from the cabinets into the store. The predictions were shown to be qualitatively but not quantitatively correct. Further work by Foster (FRPERC, 1999) of a single refrigerated aisle provided valuable information to the designers of supermarket ventilation.

If ‘in house’ CFD modelling is to be carried out an initial investment followed by running costs will need to be considered. The initial investment in CFD is hardware, training of staff and computing.

UNIX workstations used to be required to run the CFD code, these started at approximately £5 000, however increases in computer power have allowed CFD to be run on desktop PC’s, therefore the cost has come down to about £1 000. Computing will have to be upgraded at regular intervals to keep up with increasing complexity of the models generated.

The cost of the CFD code is approximately £12 000 plus a yearly maintenance contract which could be in the order of £2 000 to £3 000 per year. Either CFD modellers can be hired to carry out the work or current staff could be trained. An initial investment of time will be required for a new user of CFD to begin producing useful data, this maybe 6 months but depends greatly on the complexity of the models required.

At some point in the future the cost savings of CFD will be greater than the investment in CFD. As the productiveness of the CFD user increases with time a greater return from the investment will be achieved (FRPERC, 1996).

Another approach is to use a CFD consultancy company to carry out CFD modelling whenever needed. This will save investment costs but may become more expensive if used a great deal over a number of years. You can expect to pay upwards of £500 per day for CFD consultancy work.

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Report K

Heat Recovery for Supermarkets - Weatherite

Introduction

The title of this discussion document should be re-defined as energy recovery rather than heat recovery as in point of fact energy is available in the form of heat and coolth, which can be utilized dependant upon the application and the seasonal conditions.

The reduction in energy consumption is certainly becoming a major factor in the system design of a store but capital costs may dictate its viability.

Utilisation status

Many of the proposed methods of recovery are well proven and established, however some are in their infancy and continue to be developed within a few sample installations.

Applications

Identification of the many sources of energy within the store environment is necessary to establish the options available and these will include both heat and coolth sources. Consideration must also be given to the need and utilization of the recovered energy to establish its viability.

Heat Sources Within The Store	**Recovery**
Lighting	HVAC recirculation within luminaries
Stratification	High level return air to HVAC
Food Refrigeration	a. Direct integration of refrigeration system with HVAC b. Water Glycol loop with water cooled condensers for rejection and HVAC heat pump systems to enhance recovery c. Direct rejection of the integrated cabinet condensers to the occupied space
Extract Air from café/Bakery	a. Thermosyphon heat pipes b. Run around coils c. Crossflow plate heat exchangers d. Thermal wheels e. Heat pump technology f. Combinations of the above
Outside Air	Heat pumps
Ground Source Water	Heat pumps

Coolth Sources	Recovery
Refrigeration case spillage	Low level cold air retrieval to HVAC
Extract air from conditioned space	a. Heat Pumps b. Run around coils c. Cross flow heat exchangers d. Thermal wheels e. Combination of above
Outside air	Modulating dampers for free cooling
Ground source water	Heat pumps

Design limitations

The primary sources of energy will require review dependant upon site location particularly with regard to availability and cost as the will not be consistent. The requirement for the utilization of the recovered energy is the prime consideration followed by the design requirements of the system. Stores will vary in the need for heating, cooling, humidity control and noise.

It is important to evaluate the total energy within the store and not consider elements in isolation as the design may increase running costs of one element but produce greater savings within the global picture

Costs versus energy savings

The inevitable conflict between capital cost and energy savings will always be the subject of contention but these must be reviewed and compared in order to establish the viability of the recovery. The final decision will clearly be one of compromise.

Performance

Closer and more thorough monitoring of installations must be introduced to establish and confirm performance of the systems and reliability of the plant and equally important is the evaluation of the utilization of the recovered energy. Seasonal performance during winter and summer must be determined to appreciate a true annual report

Maintenance

The improvement in maintenance quality will only seek to enhance the effectiveness of the recovery. The nature of the systems may require additional skill levels to be improved with the necessary training

Environmental and safety issues

The analysis and conclusions should reflect the considerations of both the environment and health and safety.

In conclusion

The impact of wasted energy and the inevitable cost implications not ignoring the environmental issues will dictate a positive move towards the incorporation of energy recovery but it is important to consider the global picture and not just specific areas in order to establish the total viability of the scheme.

ORNL/Sub-62X-SX363C

Development and Demonstration of an Advanced Supermarket Refrigeration/ HVAC System

September 2001

David H. Walker

ORNL/Sub-62X-SX363C

DEVELOPMENT AND DEMONSTRATION OF AN ADVANCED SUPERMARKET REFRIGERATION/HVAC SYSTEM

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September 2001

Period: 05/08/97–08/31/01

Final Analysis Report

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EXECUTIVE SUMMARY

Supermarkets are the largest users of energy in the commercial sector with a typical supermarket consuming on the order of 2 million kWh annually. One of the largest uses of energy in supermarkets is for refrigeration, which is as much as half of the store's total.

The large majority of U.S. supermarket refrigeration systems employ direct expansion air-refrigerant coils as the evaporators in display cases and coolers. Compressors and condensers are kept in a remote machine room located in the back or on the roof of the store. Piping is provided to supply and return refrigerant to the case fixtures. As a result of using this layout, the amount of refrigerant needed to charge a supermarket refrigeration system is very large. A typical store will require 3,000 to 5,000 lb of refrigerant. The large amount of piping and pipe joints used in supermarket refrigeration also lead to significant leakage, which can amount to a loss of up to 30 percent of the total charge annually.

With increased concern about the impact of refrigerant leakage on global warming, new supermarket system configurations requiring significantly less refrigerant charge are now being considered. Advanced systems of this type include:

- Secondary loop – A secondary refrigerant loop is run between the display cases and a central chiller system. The secondary fluid is refrigerated at the chiller and is then circulated through coils in the display cases where it is used to chill the air in the case.

- Distributed – Multiple compressors are located in cabinets placed on or near the sales floor. The cabinets are close-coupled to the display cases and heat rejection from the cabinets can be done through the use of a glycol loop that connects the cabinets to a fluid cooler, or with direct outdoor air-cooled condensers.

- Advanced self-contained - Each display case is equipped with a water-cooled condensing unit. A fluid loop is connected at all condensing units and is used for heat rejection.

- Low-charge multiplex - The multiplex refrigeration system is equipped with control piping and valves to allow operation at close to critical charge, greatly reducing the amount of refrigerant needed.

While these systems use less refrigerant, energy use varies and can be much more than is now seen with centralized multiplex racks. The centralized refrigeration systems are mature in their technology and have been optimized in many ways, in terms of first cost and installation, reliability and maintenance, and energy use.

For an advanced refrigeration system to replace centralized multiplex racks such systems must offer an incentive for their use, such as reduced first cost, good return on investment for first cost, or lower operating and energy costs. Many of the features now used to reduce energy consumption of the centralized systems could be employed with the advanced systems to increase their energy efficiency. Examples include reduced head pressure and mechanical subcooling. The advanced refrigeration systems also have inherent characteristics, which could lead to reduced energy consumption if these features can be utilized as much as possible. An example is that both the secondary loop and distributed refrigeration systems employ significantly shorter refrigerant suction lines (close-coupling), which mean that pressure drop and heat gain is much less than is seen with presently installed central refrigeration systems. Higher suction pressure and lower return gas temperature can translate into lower compressor energy consumption. Scroll compressors used in the distributed refrigeration system (for reduced noise levels) can operate at a lower condensing temperature than reciprocating compressors, because scroll compressors have no suction valves. This feature could allow the distributed refrigeration system to operate at a much lower head pressure (i.e., floating head pressure). Similarly, the charge control method used with the low-charge multiplex system allows the use of lower minimum condensing temperatures with the multiplex system, which lowers energy use during winter operation.

HVAC also represents a large portion of the energy use of a supermarket, on the order of 10 to 20 percent of the store total, depending upon geographic location. The refrigerated fixtures installed in a supermarket have a major impact on the store HVAC.

A possible way to utilize refrigeration reject heat for space heating is through water-source heat pumps, when a glycol/water loop is used for refrigeration heat rejection. In this way the refrigeration reject heat is recovered to provide space heating. This method offers several advantages, which are that a much larger portion of the reject heat can be reclaimed, and the condensing temperature and head pressure of the refrigeration system does not have to be elevated for the heat pumps to use the reject heat. Refrigeration energy savings achieved by low head pressure operation can be realized along with the energy benefits seen through heat reclaim.

An investigation of low charge refrigeration and integrated refrigeration and HVAC was conducted. A model supermarket was formulated and energy consumption estimates were made for present multiplex refrigeration with air-cooled condensing and mechanical subcooling and the advanced, low-charge systems. A similar analysis was performed for the store HVAC where conventional rooftop units, refrigeration heat reclaim, and water-source heat pumps were examined and compared. Four locations with greatly varying ambient conditions were chosen as modeled sites for these analyses.

Figure ES-1 and Table ES-1 give the results for the comparison of refrigeration systems. Results from the analysis showed that the largest energy savings were achieved by the distributed and secondary loop refrigeration systems. The distributed system produced energy savings, ranging from 10.2 to 16.2 percent of multiplex consumption. Secondary loop refrigeration showed reductions in energy of 9.2 to 16.4 percent for the locations investigated. The secondary loop system had higher energy savings than the distributed system for the Memphis and Los

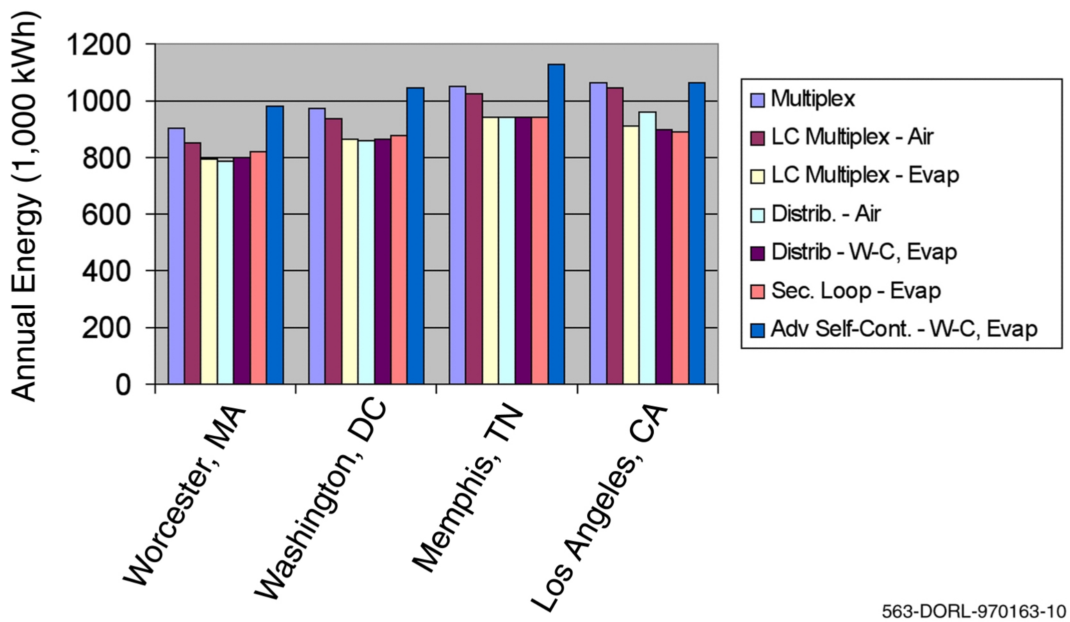

Figure ES-1. Annual energy consumption for low-charge supermarket refrigeration systems

Table ES-1. Annual energy consumption (kWh) for low-charge refrigeration systems for selected locations

Location	Multiplex Air-Cooled	Low-Charge Multiplex Air-Cooled	Low-Charge Mutiplex Evap Condenser	Distributed Air-Cooled	Distributed Water-Cooled, Evaporative	Secondary Loop Evap. Condenser	Advanced Self-Contained Water-Cooled, Evaporative
Worcester, MA	904,500	850,000	791,600	785,700	802,200	821,600	983,700
Washington, DC	976,800	935,200	863,600	860,500	866,100	875,200	1,048,300
Memphis, TN	1,050,200	1,027,100	941,500	942,800	943,200	940,400	1,126,800
Los Angeles, CA	1,067,200	1,042,600	911,300	958,432	894,400	892,400	1,066,800

Angeles sites, while the distributed system showed lower energy consumption for the Worcester and Washington sites. The low-charge multiplex system showed less energy use than the multiplex baseline for all locations. Savings ranged from 2.2 to 6.0 percent and 10.4 to 14.6 percent for the low-charge multiplex with air-cooled and evaporative condensing, respectively.

The energy savings achieved by the distributed refrigeration system can be attributed to close-coupling of the compressors to the display case evaporators, operation of the scroll compressors at 60°F minimum condensing temperature, and the use of evaporative heat rejection with the fluid loop. Savings seen with the secondary loop system are due to close-coupling of the compressors and the chiller evaporator, subcooling produced by brine heating for defrost, and the use of evaporative condensing. The refrigeration energy of the secondary loop system was found to be less than that of the distributed system, but the added energy associated with brine pumping negated some of this advantage. The energy savings seen with the low-charge multiplex system were due to the ability of this system to operate at very low minimum condensing temperatures. The minimum condensing temperatures were 40 and 60°F for low and medium temperature refrigeration, respectively.

Total equivalent warming impact (TEWI) estimates were made for the refrigeration systems for operation in Washington, DC. These estimates are shown in Table ES-2. The distributed and secondary loop systems both showed significant reduction in TEWI, compared to the multiplex system.

Table ES-3 gives the estimated operating savings for the low-charge systems due to reduced refrigerant leakage. For this analysis, refrigerant costs of $1.75/lb and $7.75/lb were used for R-22 and R-404A, respectively.

Figure ES-2 and Table ES-4 show the analysis results for operation of the store refrigeration and HVAC. For all locations, the integrated system consisting of the distributed refrigeration system and the water-source heat pumps produce the lowest operating cost (combined cost for electric, natural gas, water, and refrigerant). Operating cost savings were estimated to be 11.1 to 19.2 percent when compared to multiplex refrigeration with conventional HVAC.

Table ES-5 shows the estimated payback of installed cost premium for several low-charge refrigeration systems. The low-charge multiplex system had an immediate payback, since no installed cost difference exists between the low-charge and baseline multiplex systems. The distributed system showed paybacks ranging from 3.4 to 7.0 years, while the secondary loop system showed paybacks of 8.3 to 16.8 years. These payback values are extremely sensitive to the installed cost premium for these systems, which is highly variable depending upon arrangements between the supermarkets and their equipment suppliers and installers. These cost differences are likely to be reduced for either the distributed or the secondary loop systems as more such systems are implemented.

Table ES-2. Total Equivalent Warming Impact (TEWI) for supermarket refrigeration

System	Condensing	Charge (lb)	Refrigerant	Leak (%)	Annual Energy (kWh)	TEWI (million kg of CO_2) Direct	Indirect	Total
Multiplex	Air-Cooled	3,000	R404A/	30	976,800	13.62	9.52	23.15
	Evaporative	3,000	R-22	30	896,400	13.62	8.74	22.36
Low-Charge	Air-Cooled	2,000	R404A/	15	935,200	4.54	9.12	13.66
Multiplex	Evaporative	2,000	R-22	15	863,600	4.54	8.42	12.96
Distributed	Air-Cooled	1,500	R404A	10	860,500	3.33	8.38	11.71
Distributed	Water-Cooled, Evaporative	900	R404A	5	866,100	1.00	8.44	9.44
Secondary Loop	Evaporative	500	R507	10	875,200	1.13	8.54	9.67
Secondary Loop	Water-Cooled, Evaporative	200	R507	5	959,700	0.23	9.36	9.58
Advanced Self-Contained	Water-Cooled, Evaporative	100	R404A	1	1,048,300	0.02	10.22	10.24

Results for site in Washington, DC – 15 year service life.
Conversion factor = 0.65 kg CO_2/kWh.
Multiplex – 33.3% R404A (low temperature), GWP = 3260; 66.7% R22 (medium temperature), GWP = 1700.
Distributed and Advanced Self-Contained – 100% R404A, GWP = 3260.
Secondary Loop – 100% R507, GWP = 3300.

Table ES-3. Estimated operating cost savings for reduced refrigerant leakage

System	Annual Leakage (lb) R-404A	R-507	R-22	Savings ($)
Multiplex - (R-404A/R-22)	300		600	
Multiplex - Low Charge	100		200	2,250
Multiplex - Low Charge Evap Cond	100		200	2,250
Distributed Air-Cooled	150			2,213
Distributed Water-Cooled, Evap	45			3,026
Secondary Loop Evap Condensing		50		2,988
Secondary Loop Water-Cooled, Evap		10		3,298
Advanced Self-Contained	1			3,367

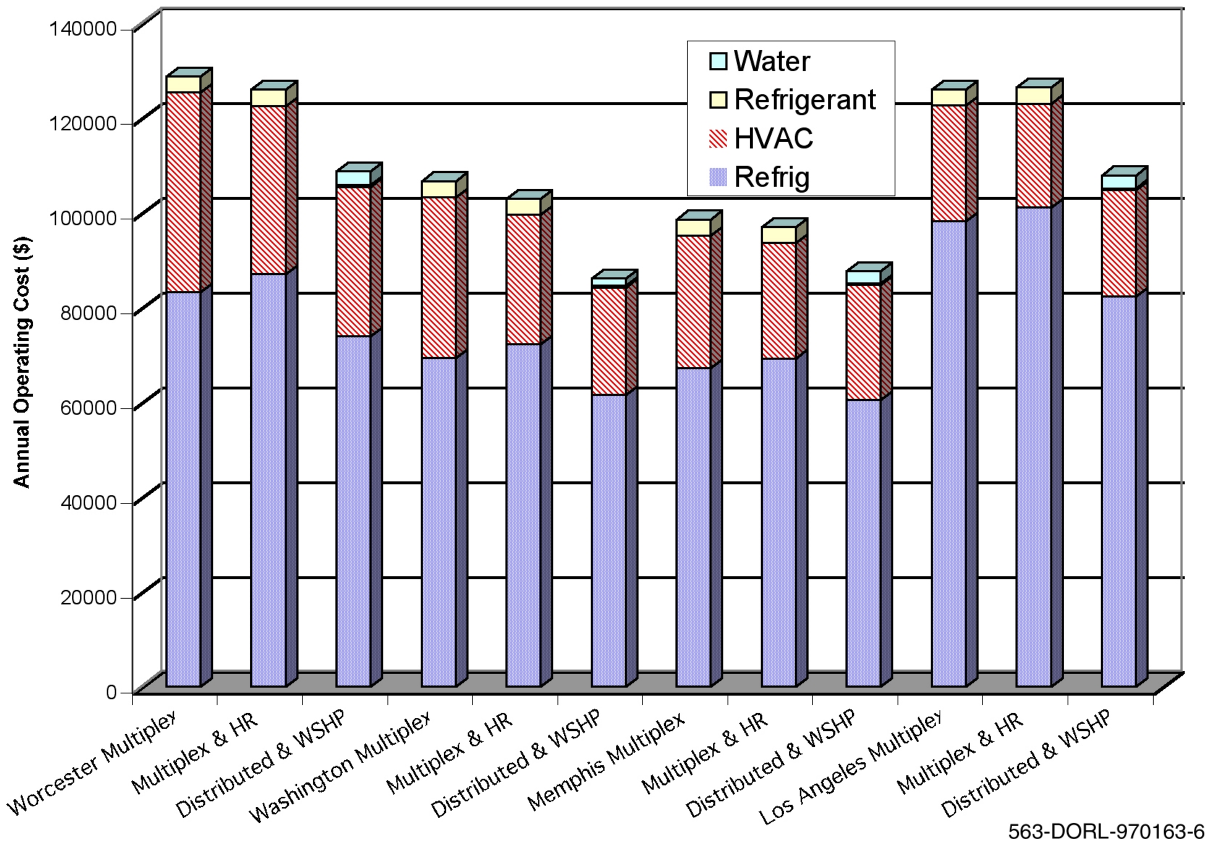

Figure ES-2. Operating cost of supermarket refrigeration and HVAC

Table ES-4. Annual cost savings achieved by distributed refrigeration and water-source heat pumps versus multiplex refrigeration and conventional HVAC

	Annual Operating Savings			
	with Heat Reclaim		Distributed Refrigeration and WS Heat Pumps	
Location	$	%	$	%
Worcester, MA	2,817	2.2	20,009	15.5
Washington, DC	3,732	3.5	20,469	19.2
Memphis, TN	1,568	1.6	10,929	11.1
Los Angeles, CA	-397	-0.3	18,079	14.4

Table ES-5. Estimated operating cost savings and payback for low-charge supermarket refrigeration (versus multiplex with air-cooled condensing)

Location	Low-charge Multiplex, Air-Cooled		Low-charge Multiplex, Evaporative		Distributed, Water-Cooled, Evaporative		Secondary Loop, Evaporative	
	$	Year	$	Year	$	Year	$	Year
Worcester, MA	7,264	0	11,176	0	10,977	5.5	9,153	16.1
Washington, DC	5,204	0	9,479	0	10,078	6.0	9,393	15.6
Memphis, TN	3,728	0	7,940	0	8,608	7.0	8,748	16.8
Los Angeles, CA	4,513	0	15,201	0	17,532	3.4	17,678	8.3

Table ES-6 shows the payback associated with distributed refrigeration and water-source heat pumps for combined operation of refrigeration and HVAC. The payback on cost premium for distributed refrigeration and water-source heat pumps was 4.2 years for Worcester, MA and Washington, DC, and 4.7 years for Los Angeles, CA. The payback for operation in Memphis, TN was 10.8 years. The lowest paybacks were seen for sites with large space heating loads. For these locations, the operation of the water-source heat pumps helped to reduce the combined payback.

The results seen in this investigation show that low-charge refrigeration systems can reduce energy and operating costs if properly designed and operated. Demonstration of these technologies by field testing, possibly in conjunction with water-source heat pumps for HVAC, will help develop best practices for these systems and also better quantify energy savings. This information will help to accelerate the use of low-charge refrigeration systems by the supermarket industry.

Table ES-6. Estimated payback for distributed refrigeration and water-source heat pumps

Location	Savings ($)		Payback (Year)	
	Refrigeration	Combined	Refrigeration	Combined
Worcester, MA	10,977	20,009	5.5	4.2
Washington, DC	10,078	20,469	6.0	4.2
Memphis, TN	8,608	10,929	7.0	7.8
Los Angeles, CA	17,532	18,079	3.4	4.7

1. INTRODUCTION

Supermarkets are the largest users of energy in the commercial sector. A typical supermarket (35,000 ft^2 of sales area) consumes on the order of 2 million kWh annually. Many larger superstores and supercenters also exist that can consume as much as 3 to 5 million kWh/yr.

One of the largest uses of energy in supermarkets is for refrigeration. Most of the product sold is perishable and must be kept refrigerated for storage and during display. Typical energy consumption for supermarket refrigeration is on the order of half of the store's total. Compressors and condensers account for 30 to 35 percent. The remainder is consumed by the display and storage cooler fans, display case lighting, and for anti-sweat heaters used to prevent condensate from forming on doors and outside surfaces of display cases.

Typical U.S. supermarket refrigeration systems today employ direct expansion air-refrigerant coils as the evaporators in display cases and coolers. Compressors and condensers are kept in a remote machine room located in the back or on the roof of the store. Piping is provided to supply and return refrigerant to the case fixtures. As a result of using this layout, the amount of refrigerant needed to charge a supermarket refrigeration system is very large. A typical store will require 3,000 to 5,000 lb of refrigerant. The large amount of piping and pipe joints used in supermarket refrigeration also lead to significant refrigerant leakage, which can amount to a loss of up to 30 percent of the total charge annually.

With increased concern about the impact of refrigerant leakage on global warming, new supermarket system configurations requiring significantly less refrigerant charge are now being considered. Advanced systems of this type include:

- Low-charge multiplex - Several refrigeration system manufacturers now offer control systems for condensers that limit the amount of refrigerant charge needed for the operation of multiplex refrigeration, which approach reduces the charge by approximately 1/3.

- Secondary loop – A secondary fluid loop is run between the display cases and a central chiller system. The secondary fluid is refrigerated at the chiller and is then circulated through coils in the display cases where it is used to chill the air in the case.

- Distributed – Multiple scroll compressors are located in cabinets placed on or near the sales floor. Scroll compressors are employed to minimize system noise in the sales area. The cabinets are close-coupled to the display cases and heat rejection from the cabinets can be done through the use of a glycol loop that connects the cabinets to a fluid cooler, in order to minimize refrigerant charge.

- Advanced self-contained – Self-contained refrigeration consists of compressors and condensers built into the display cases. An advanced version of this concept would used horizontal scroll compressors with capacity control, such as unloading, and water-cooled condensers with a water loop for heat rejection. The advanced self-contained refrigeration system would employ the smallest refrigerant charge.

While all of these systems use less refrigerant, energy use varies and can be much more than is now seen with centralized multiplex racks if the system design and component sizing does not take energy consumption into account. Examination of these systems on a total environmental warming basis (through TEWI) is needed to determine which has the least environmental impact.

The centralized refrigeration systems are mature in their technology and have been optimized in many ways, in terms of first cost and installation, reliability and maintenance, and energy use. At present, no regulations exist requiring reduction of refrigerant charge in supermarkets. For an advanced refrigeration system to replace centralized multiplex racks, such systems must offer an incentive for their use, such as reduced first cost, good return on investment for first cost, or lower operating and energy costs.

Many of the features now used to reduce energy consumption of the centralized systems could be employed with the advanced systems to increase their energy efficiency. Examples include reduced head pressure and mechanical subcooling. Compressor control strategies have been developed that can maintain suction pressure within a tight tolerance to a set point value. Similar control strategies that limit energy use have been implemented for condenser fans. Implementation of these energy-saving components and control strategies with the advanced, low-charge systems could lead to improved performance.

The advanced refrigeration systems also have inherent characteristics, which could lead to reduced energy consumption if these features can be utilized as much as possible. An example is that the secondary loop, distributed, and advanced self-contained refrigeration systems employ significantly shorter refrigerant suction lines (close-coupling), which mean that pressure drop and heat gain is much less than is seen with presently installed central refrigeration systems. Higher suction pressure and lower return gas temperature can translate into lower compressor energy consumption. Scroll compressors used in the distributed and advanced self-contained refrigeration system can operate at lower condensing temperature than reciprocating compressors, because scroll compressors have no suction valves. This feature could allow the distributed refrigeration system to operate at a much lower head pressure (i.e., floating head pressure), which has been shown previously (1-1) to be a method of significantly reducing refrigeration energy consumption. Low head pressure operation can also be obtained with the advanced self-contained refrigeration, if the compressors are equipped with capacity control. Low-charge multiplex refrigeration are operated at very low head pressure values, because of the improved charge control offered by these systems. Initial estimates suggest that incorporation of the energy-saving features in these advanced refrigeration systems could produce a reduction in refrigeration energy of as much as 10 to 12 percent of present use.

HVAC also represents a large portion of the energy use of a supermarket, on the order of 10 to 20 percent of the store total, depending upon geographic location. Refrigerated fixtures installed in a supermarket have a major impact on the store HVAC. For space cooling, the refrigeration removes both sensible and latent heat from the store, such that the sensible-to-latent load ratio is much smaller than is seen in most commercial buildings.

Also, because of the installed refrigeration, space heating is the dominant HVAC load. Reclaim of refrigeration reject heat for space heating has been done in past supermarket installations, but the amount of heat reclaimed is limited to desuperheating only due to operating considerations, which amounts to only 14 to 20 percent of the total amount of heat available.

A possible alternate approach to utilize refrigeration reject heat for space heating is through water-source heat pumps. The heat pumps can be installed in the glycol/water loop used for refrigeration heat rejection and use the refrigeration heat to provide space heating. This method offers several advantages, which are that a much larger portion of the reject heat can be reclaimed, and the condensing temperature and head pressure of the refrigeration system does not have to be elevated for the heat pumps to use the reject heat. Refrigeration energy savings achieved by low head pressure operation can be realized along with the energy benefits seen through heat reclaim.

The U.S. DOE initiated this research project, recognizing that advanced supermarket refrigeration systems have the potential of both environmental benefits and energy savings. The project involved investigation of low-charge refrigeration systems to determine configurations and designs that produced the largest energy savings compared to presently installed refrigeration systems. Along with examination of the refrigeration, integration of refrigeration and HVAC was also addressed to find an overall approach that provided maximum energy savings and operating cost reduction. Possible future project activities include installation of the combined refrigeration and HVAC system identified in a supermarket, and field testing and measurement of the performance of this advanced system.

Section 1 Reference

1-1. Walker, D.H., Foster-Miller, Inc., *Field Testing of High-Efficiency Supermarket Refrigeration,* EPRI Report No. TR-100351, Electric Power Research Institute, Palo Alto, CA., December, 1992.

2. DESCRIPTION OF SUPERMARKET REFRIGERATION SYSTEMS

A representative supermarket layout is shown in Figure 2-1. Refrigerated fixtures are located throughout the store, because of the large amount of perishable food products that are sold. These fixtures fall into two categories, which are display cases and walk-in storage coolers. The display cases are located on the sales floor and are designed to refrigerate food products while providing a place to merchandise them. Walk-in coolers are used to store food products during the time period between receiving the product and placing the product out for sale.

Refrigeration of the display cases and walk-in coolers is done through the use of direct expansion refrigerant/air coils located in each case or cooler. Refrigerant piping is provided to each coil to supply liquid refrigerant to the coil and remove evaporated refrigerant from the coil and return the gas to the refrigeration compressors.

The compressors are located in a machine room in a remote part of the store, such as in the back room area or on the roof. The system condensers are located either in the machine room, or more likely, on the roof above the machine room.

2.1 Refrigerated Display Cases

The purpose of refrigerated display cases in a supermarket is to provide temporary storage for perishable foods prior to sale. Most of the design characteristics and general shape and layout of display cases are based on marketing specifications and constraints. Display cases have been developed and refined for specific merchandising applications, and cases exist specifically for the storage and display of such items as frozen food, meats, fish, cheese, dairy products, and produce.

Despite the diversity of use and style, refrigerated display cases can be described as being of the following three general types (Figure 2-2):

- Tub (or coffin): The tub case is often used for the storage and display of frozen foods and meats. Tub cases operate at a very uniform temperature and require the least amount of refrigeration per foot of any display case type. The primary disadvantage of the tub is a low product storage volume per square foot of sales area taken up by the case.

- Multi-deck: The multi-deck case possesses the largest storage volume per square foot of floor area, because of the use of an upright cabinet and shelves. Refrigeration requirements are very high for multi-deck cases, including a large latent load portion due to the entraining of ambient air in the air curtain passing over the opening of the case.

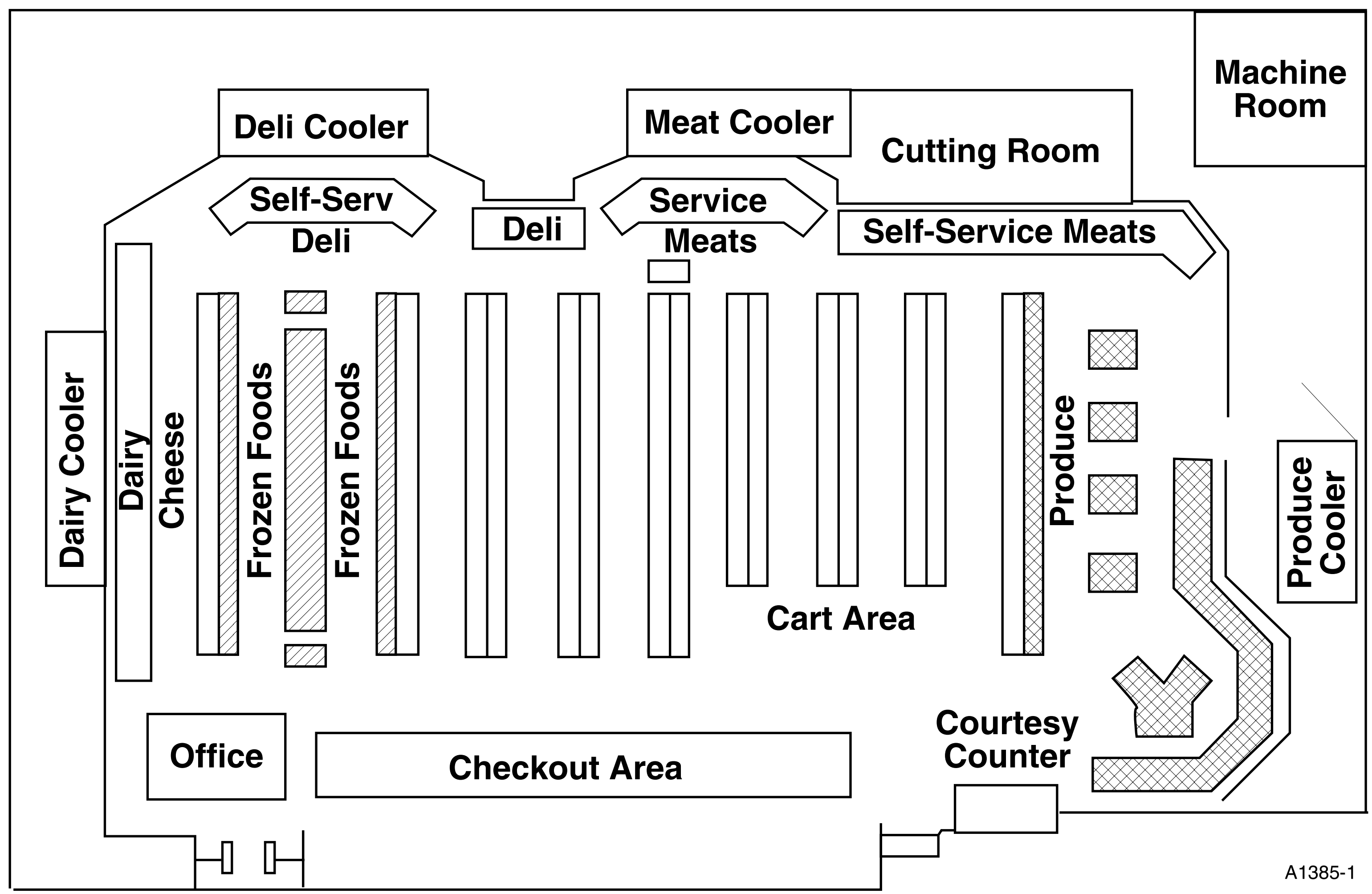

Figure 2-1. Layout of a typical supermarket

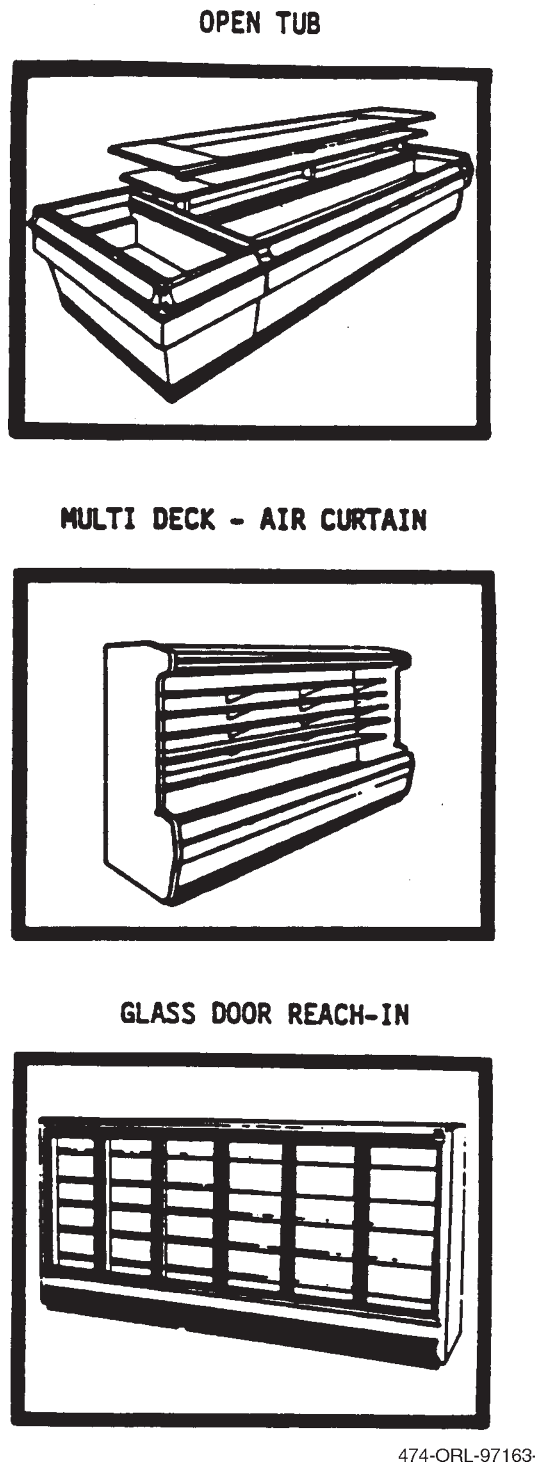

Figure 2-2. Display case types employed in supermarkets

- Glass door reach-in: The reach-in case has glass doors over the opening of the case; these must be opened for product removal and stocking. Reach-in cases are used in supermarkets primarily for frozen foods, because of their ability to contain the cold refrigerated air, which reduces the "cold aisle" problem. The refrigeration loads associated with the glass door reach-in case are normally less than those for the multi-deck but greater than for the tub case. Glass door cases are, however, equipped with anti-sweat electric heaters in the doors to prevent fogging and decreased visibility of the product.

The design refrigeration load of a display case is influenced by its type and operating temperature. Table 2-1 presents the design refrigeration loads for several display case types, along with a description of the installed fan, light, and heater wattage.

2.2 Walk-In Storage Coolers

Walk-in storage coolers are used to hold food product prior to stocking in the display cases. One walk-in is normally associated with each type of food product, such as meat, dairy, produce, frozen food, etc. For meat and produce, the walk-in cooler is also used as a preparation area where the foods are cut, uncrated, or packaged prior to display. Walk-in coolers are fabricated from pre-insulated panels that are field assembled. Refrigeration is provided by one or more fan coils located on the ceiling of the walk-in. Walk-ins are equipped with doors large enough to allow pallet loads of product to be brought in or out. Doors are often left open so as not to impede the entry or removal of pallets. These open doors allow ambient air to pass into the cooler. This air infiltration is the largest component of the refrigeration load of the walk-in. Vinyl strips are sometime hung over walk-in doorways to provide a barrier to air movement and help reduce the amount of ambient air entering the walk-in.

2.3 Compressor Systems

Two compressor system types are now found in most supermarkets. They are the stand-alone and the multiplex, parallel compressor systems. Stand-alone systems are found in smaller

Table 2-1. Typical refrigeration and electric requirements for supermarket display cases

Display Case	Evaporator Temperature (ºF)	Refrigeration Load (Btu/h/ft)	Electric Requirements (W)*	
			Fans and Heaters	Lights
Frozen Food Tub	-25	600	595	-
Multi-Deck Dairy	15	1,800	440	345
Single-Deck Meat	15	550	190	-
Glass Door Reach-in	-25	560	1040	345

*For a case length of 12 ft

supermarkets or in stores that are more than 20 years old. The more predominate system is the multiplex system, which consists of multiple compressors piped in parallel on common skids and grouped by suction temperature. The operating characteristics of each of these compressor systems are described below.

2.3.1 Stand-Alone Refrigeration Systems

The characteristics of stand-alone refrigeration systems (Figure 2-3) are the use of a single compressor for each display case lineup or walk-in storage box; and skid-mounted construction with all necessary refrigerant piping, control valves, receiver, electrical components, and condenser mounted with the compressor on a base or skid.

The type of compressor employed for the stand-alone refrigeration system is typically a semi-hermetic reciprocating unit. The operation of the compressor is controlled through the use of a suction pressure control strategy in which the compressor suction pressure is held between set points (cut-in and cut-out pressures) by cycling the compressor on and off.

The individual condensers employed in the stand-alone refrigeration system consist of a plate fin-type coil that is air-cooled by a fan individually installed on each condenser. Two types of condenser control are employed. For systems where the condenser is mounted on the same skid as the compressor, the fan operates when the compressor is cycled on and is turned off when the compressor is cycled off. For remote condensers, head pressure control is employed where the condensing temperature is maintained at the desired value by fan cycling. The cycling is controlled through the use of a liquid line thermostat that is set at the desired minimum condensing temperature.

Standard operating practice for stand-alone compressor systems calls for a minimum condensing temperature of 90°F, which avoids short time periods of compressor operation, prevents short-cycling of the compressor. Minimal compressor cycling is desirable primarily to maintain a uniform air temperature at the display cases.

Two types of defrost are employed with the stand-alone refrigeration system. For the medium and high temperature fixtures, off-cycle defrost is commonly used, in which frost is allowed to melt from the display case evaporator. For the very low and low temperature display cases, electric defrost is employed; electric heaters melt the frost from the case evaporator.

2.3.2 Multiplex Refrigeration Systems

The term “multiplex” refrigeration refers to the use of multiple refrigeration compressors piped to common suction and discharge manifolds, and mounted on a skid as shown in Figure 2-4. The skid also contains all necessary piping, control valves, and electrical wiring needed to operate and control the compressors and the refrigeration provided to the display cases and walk-in coolers serviced by this particular compressor rack. The discharge gas from the compressors is piped to a remotely located condenser. Liquid refrigerant returning from the condenser is piped back to the compressor rack, where a receiver, liquid manifold and associated

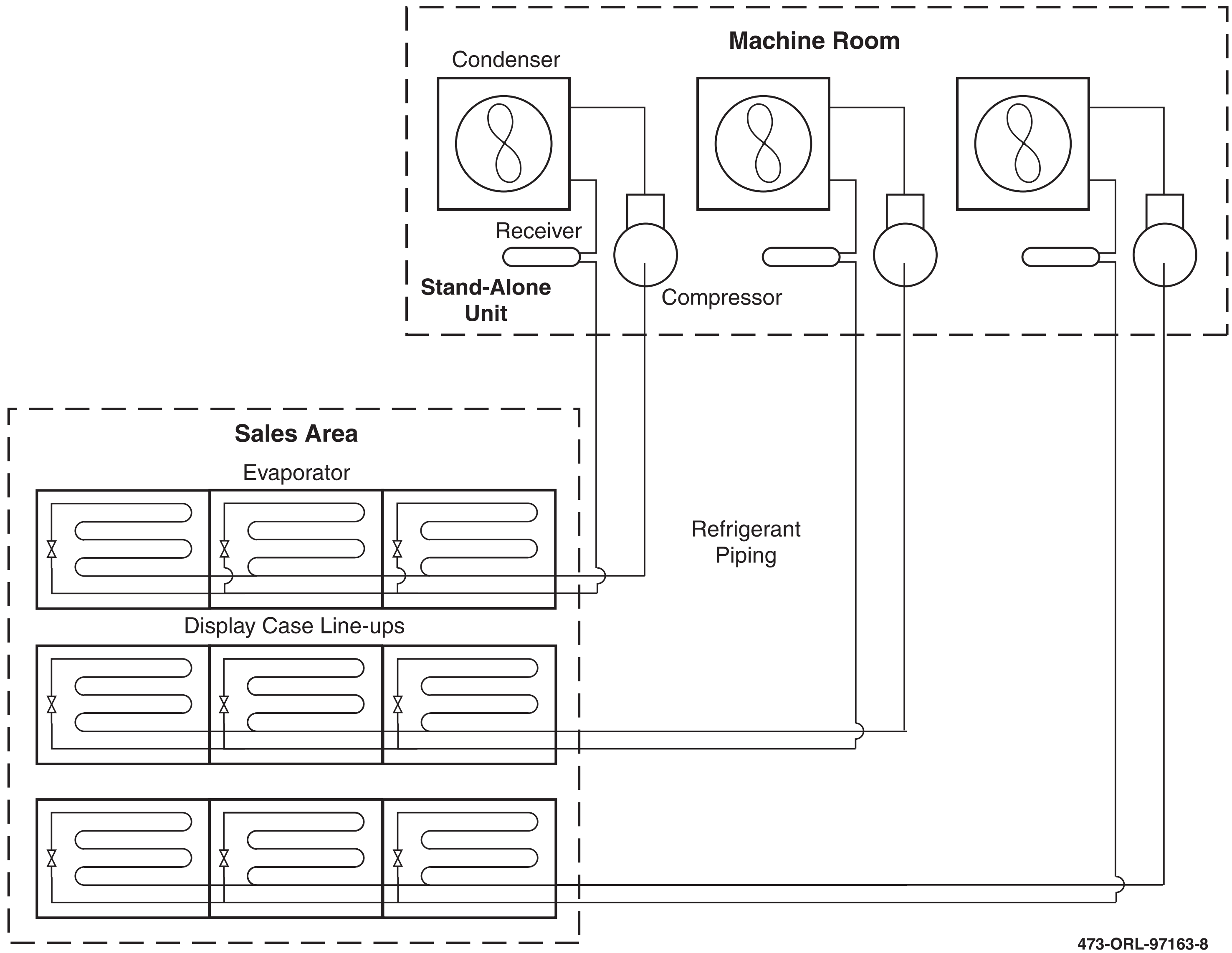

Figure 2-3. Elements of a stand-alone refrigeration system

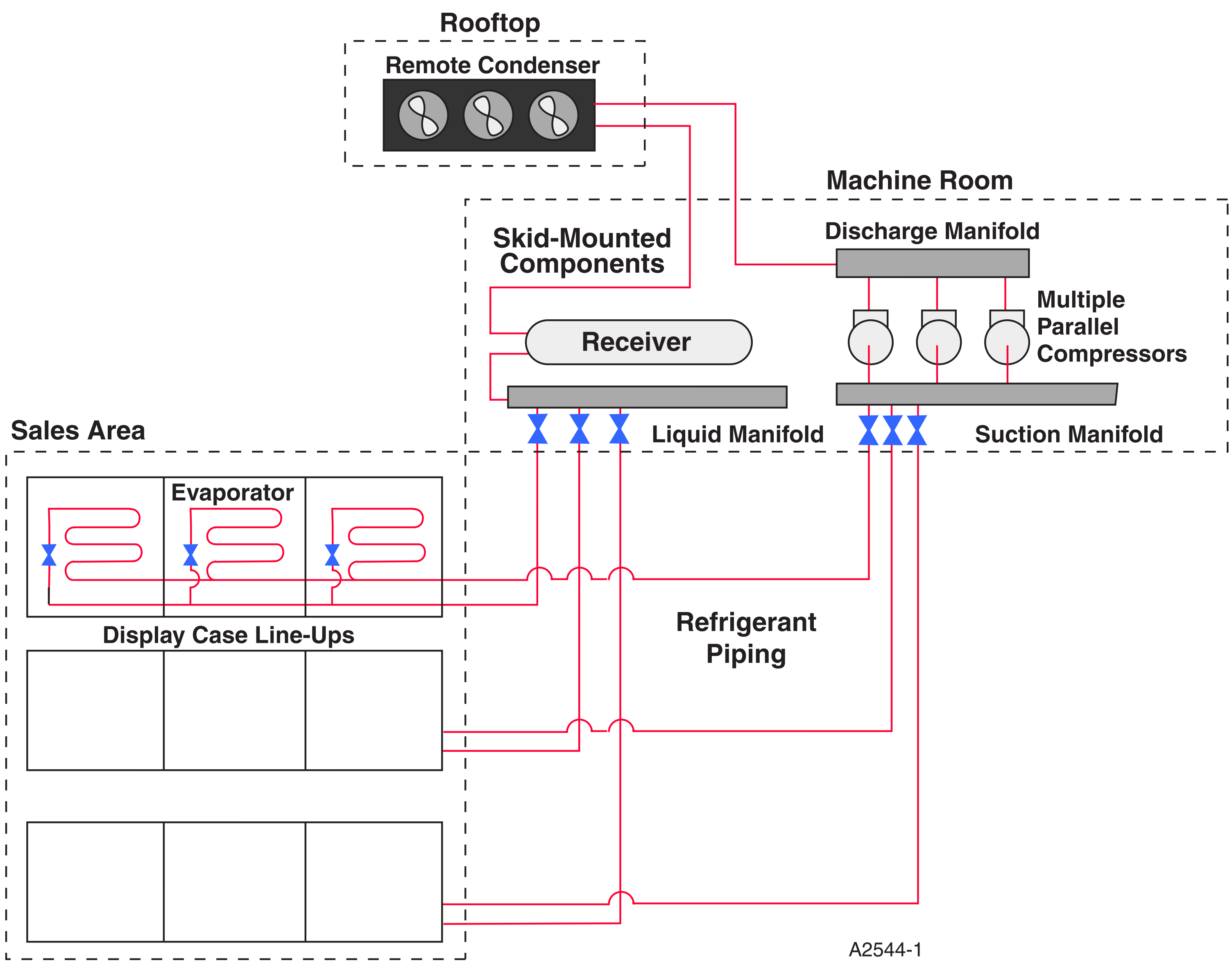

Figure 2-4. Diagram of a multiplex refrigeration system

control valves are located for distribution of the liquid to the cases and coolers. Each case or cooler circuit is piped with a liquid and suction return line that are connected to the liquid and suction manifolds located on the compressor skid. Valves used for control of each of these circuits are also located on the manifolds. The control valves employed consist of regulators to control suction pressure and solenoid valves used to control gas routing during defrost. The compressor is normally equipped with a number of pressure regulators used to control system head pressure, heat reclaim, and defrost. The rack will also have an oil separator in the discharge piping and an oil distribution system that will return oil to the compressors.

Typically 3 to 5 compressor racks will be employed to provide all refrigeration in the supermarket. The display cases and coolers are grouped and attached to the compressor racks based on required saturated suction temperature to maintain the desired case air temperature. A supermarket will have 1 or 2 low temperature racks to address all frozen food refrigeration requirements. The low temperature racks will typically operate at a -20°F SST. Refrigeration loads as low as -30°F, or as high as -10°F, will also be provided by the low temperature racks. In these situations, the suction manifold will be divided, and 1 or 2 compressors will provide the off-temperature refrigeration. The discharge of these "satellite" compressors will be piped to the common discharge manifold with the other low temperature loads so that a common condenser and liquid manifold can be used for all circuits on the rack. The remaining refrigeration circuits in the store are referred to as medium temperature and normally require a 20°F SST. Two or more compressor racks are needed to meet all medium temperature refrigeration requirements. Satellite compressors are also used for medium temperature loads requiring an SST significantly higher or lower than 20°F.

Multiplex systems commonly consist of three or four compressors that are sized such that operation of all compressors simultaneously can provide adequate capacity to meet the design refrigeration load. During off-design operation, the refrigeration load can be considerably less than the design value; at the same time, the refrigeration capacity of the compressors can increase with a decrease in ambient temperature and the condenser operating pressure. In this situation, the compressors operated by the multiplex system can be selected so that the capacity of the compressors closely matches the refrigeration load. The selection of, and the on-off cycling of the compressors is done based on suction pressure value measured at the compressor rack. The use of microprocessor-based controls allows more sophisticated control algorithms to be employed so that very close matching of the suction pressure and the set point value can be maintained with multiplex compressor systems.

Several advantages can be realized through the use of multiplexed compressor instead of single, stand-alone compressors for refrigeration. The matching of the capacity of the multiplexed compressors with the refrigeration load allows operation at the highest possible suction pressure to provide best compressor operating efficiency. In contrast, capacity control with a single compressor is accomplished by cycling within a larger suction pressure control band in order to prevent rapid on-off cycling that does not provide adequate air temperature stability at the display cases and shortens the service life of the compressor.

The use of multiplex compressors also allows operation at low head pressure because of the ability of the multiplex systems to match capacity and load continuously. The lowest head pressure seen in the operation of multiplex systems employing reciprocating compressors corresponds to a saturated discharge temperature (SDT) of 70°F, which is the lowest condensing temperature that is recommended for the operation of such compressors. Operation with low head pressure has been shown to produce energy savings of about 10 percent in compressor energy (2-1) when compared to a system operating at a minimum condensing temperature of 95°F. If scroll compressors are employed in a multiplex system, operation at lower condensing temperature than 70°F is possible. The lowest condensing temperatures seen for scroll compressors are 40 and 60°F for low and medium temperature refrigeration, respectively. These minimum values are set by the requirement that the compressor maintain a minimum pressure difference between suction and discharge in order to maintain proper oil flow for lubrication.

Other energy saving features associated with multiplex refrigeration is the use of mechanical subcooling for low temperature refrigeration. Figure 2-5 shows a piping diagram for mechanical subcooling. The liquid refrigerant flow for the low temperature system is passed through a heat exchanger where the liquid is cooled by refrigeration provided by a medium temperature refrigeration rack. The subcooling load is part of the total refrigeration load of the medium temperature rack and is shared by the compressors on the rack. Mechanical subcooling typically produces energy savings of approximately 8 percent for low temperature compressor energy consumption (2-1).

Multiplex refrigeration systems also often employ hot gas defrost, particularly for low temperature refrigeration. The piping arrangement for hot gas defrost is shown in Figure 2-6. Discharge gas is directed to the circuit requiring defrost and is passed down the suction line piping. The melting of the frost condenses the gas and the liquid is piped back to the liquid refrigerant manifold. Hot gas defrost replaces electric heaters in the display cases which heat the case air to remove frost from the coil. Savings obtained by the use of hot gas defrost is on the order of 4 percent of total compressor energy (2-1).

Reclaim of refrigeration reject heat has been done with multiplex refrigeration systems. Heat reclaim has been used for both water and space heating. Water heating is most prevalent and consists of a water tank equipped with a heat exchanger. The discharge gas from one of the low temperature racks is piped to the heat exchanger where the gas is desuperheated to heat the water in the tank.

Figure 2-7 shows the piping diagram for heat reclaim for store space heating. Operation of the heat reclaim circuit is controlled by a 3-way valve mounted in the discharge piping. The valve is actuated by a thermostat normally located in the store sales area. The discharge gas from the rack is routed to a coil mounted in the HVAC ducting, usually at the air handler. At the coil, the refrigerant is desuperheated and partially condensed by the heating of circulated store air. The refrigerant is piped back to the condenser where the remainder of the condensing occurs and the liquid is returned to the multiplex rack.

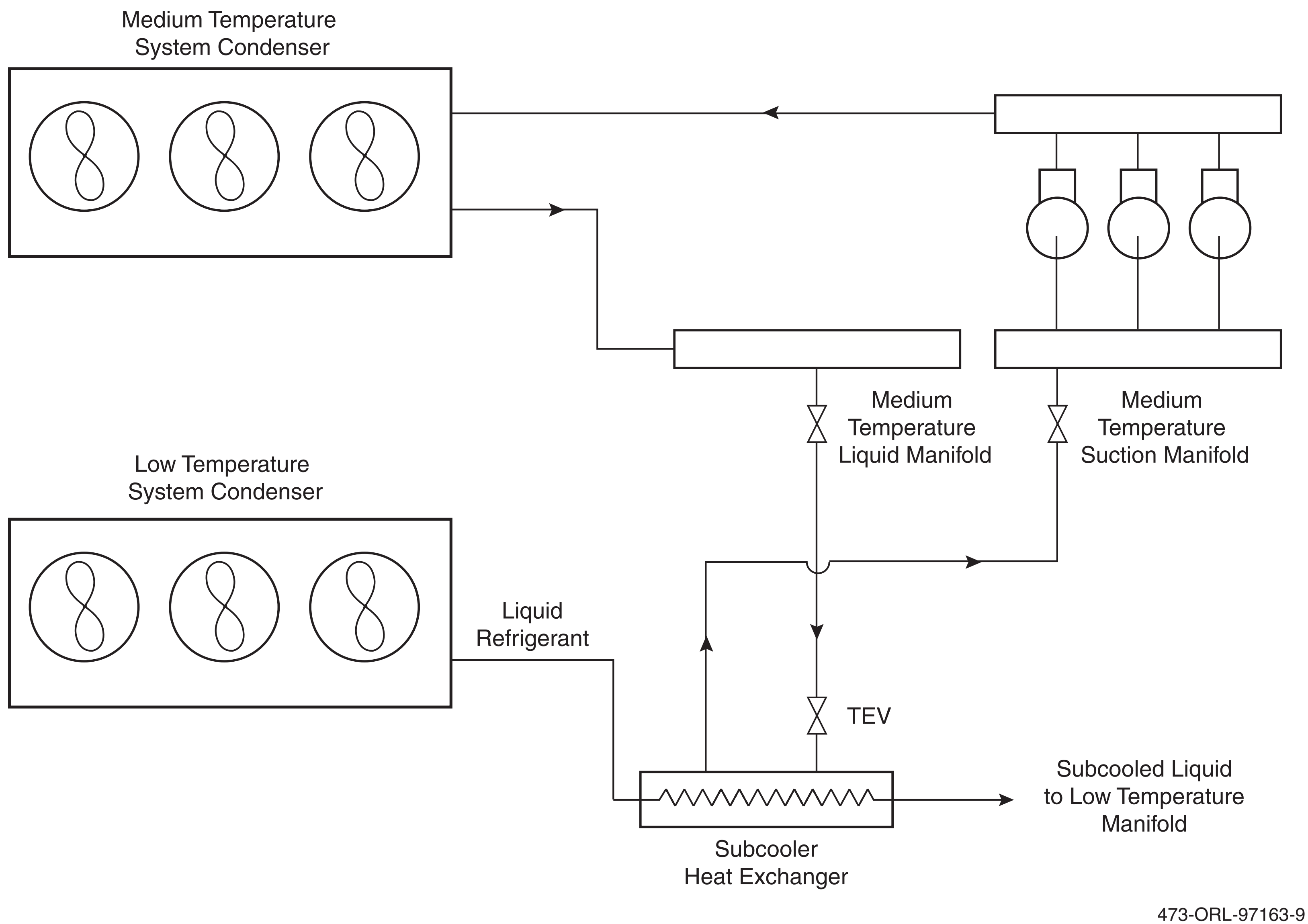

Figure 2-5. Mechanical subcooling of low temperature refrigerant

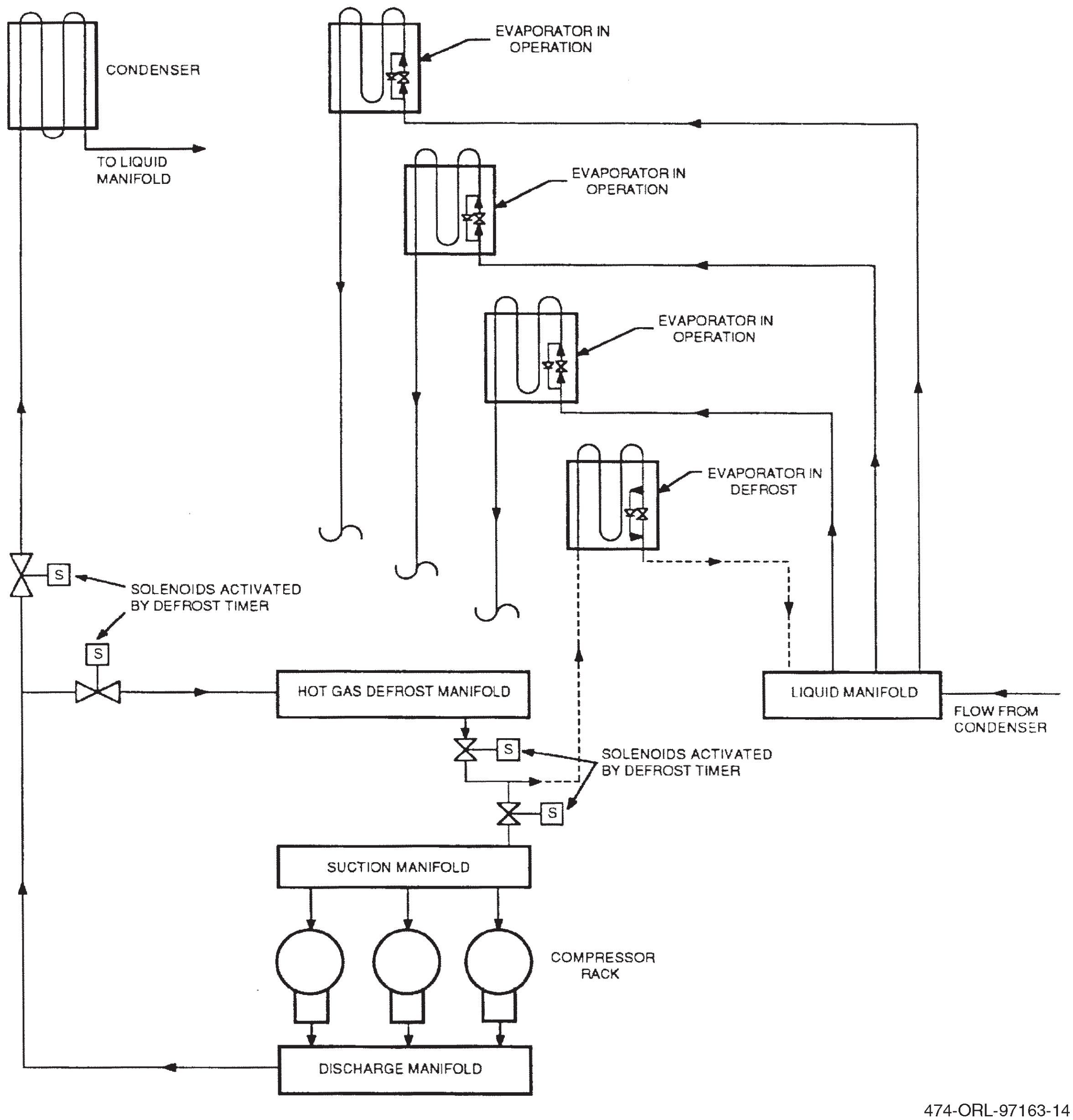

Figure 2-6. Schematic of hot gas defrost

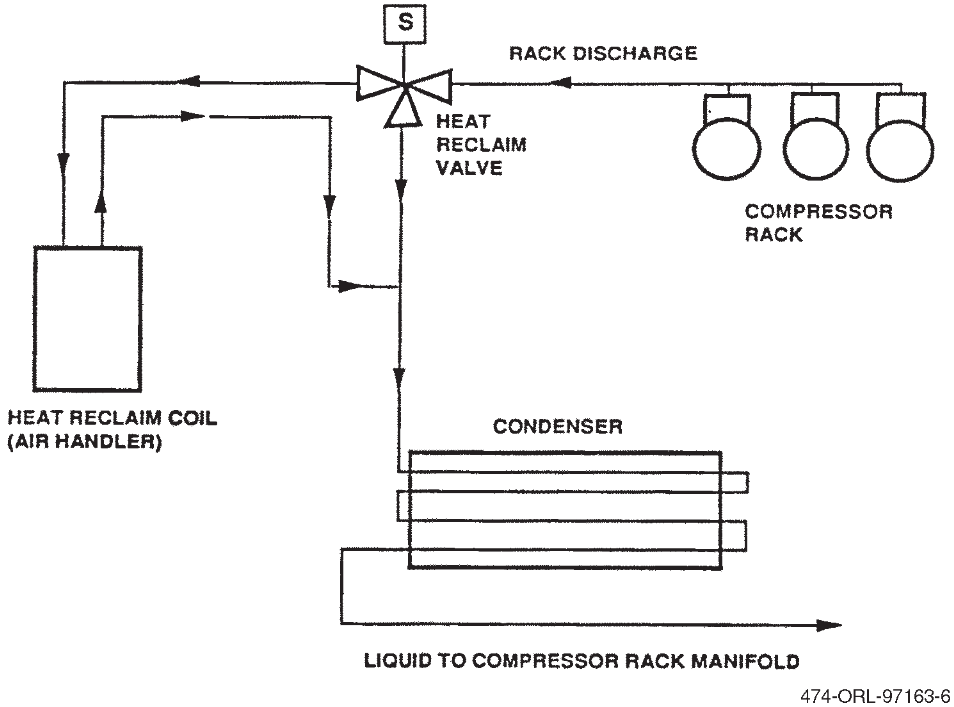

Figure 2-7. Refrigeration heat reclaim schematic

2.4 Compressors

The semihermetic reciprocating compressor is the most common compressor type used in supermarket refrigeration. In this type of compressor, the mechanical components of the compressor and the electric drive motor are contained within a common housing. The refrigerant gas returning to the compressor suction passes over the motor, providing cooling. The compressor housing is constructed of bolted sections, which can be separated to provide access for service and repair, thus the term “semihermetic.”

Semihermetic screw compressors have made some in-roads into supermarket refrigeration, primarily because the cost of such machines has dropped due to the use of advanced manufacturing methods. The reliability of screw compressor is accepted as being higher than that of reciprocating compressors, because no valves are employed and the screws are impervious to liquid refrigerant slugging. In terms of performance, screw compressors require higher power input than reciprocating compressors at equivalent load and operating conditions. Performance of the screw compressor can be increased by the use of an economizer, which provides liquid subcooling by evaporation of a portion of the liquid refrigerant in a heat exchanger that provides cooling to the remaining refrigerant liquid. The refrigerant vapor generated is injected in the screw compressor at a mid-screw location where the pressure is higher than the suction pressure of the compressor. The use of an economizer is akin to mechanical subcooling where medium temperature refrigeration is used to subcool the refrigerant liquid used for low temperature refrigeration.

Scroll compressors have been recently introduced in sizes appropriate for supermarket refrigeration. The primary advantage of scroll compressors is very quiet operation, making them suitable for use in the sales area. Scroll compressors are also tolerant to liquid slugging due to the use of moveable scroll elements, which allow the scrolls to separate if a liquid slug passes through. Scroll compressors show lower efficiency than reciprocating units for both medium and low temperature refrigeration. The loss in efficiency is caused primarily by re-expansion loss at the discharge, because no discharge valve is employed. Scroll compressors offer other energy saving possibilities. Mid-scroll injection of vapor can be used as a form of economizing to subcool refrigerant liquid. This procedure is similar to that described previously for screw compressors. Scroll compressors can be operated at lower condensing temperature than reciprocating compressors, because scroll compressors employ no valves. Lower floating head pressure values can be employed. The only limit is that a minimum pressure ratio, of approximately 2 to 1, must be maintained for proper operation of the scroll elements. For supermarkets, the systems in question are the higher end medium temperature such as produce or the meat prep area which can have an evaporator temperature as high as 35°F. This temperature will limit the minimum condenser temperature to about 60°F for refrigerants such as R-404A or R-507.

Scroll compressors used for multiplex systems, are used exclusively for distributed refrigeration for noise reduction in the sales area, and could possibly be used for self-contained, if compressors with capacities that match display case loads are available. These small capacity scroll compressors should also be equipped with capacity control, such as unloading to allow low head pressure operation.

2.5 Condensers

The most common type of condenser used in supermarket refrigeration is air-cooled. The reason for this is that air-cooled condensers require the least maintenance and have been shown to operate reliably in the non-operator environment of supermarket refrigeration.

Finned coil construction with 8 to 10 fins/in. and multiple fans are used. Typical face velocity requirements are on the order of 500 fpm. Fan motor sizes associated with air-cooled condensers are on the order of 1/2 to 1 hp. Multiple fans are employed to ensure air flow through the entire coil face. On/off fan cycling is used as a means to control condensing temperature and to reduce fan energy at ambient temperatures less than design.

Evaporative condensers are also used in some supermarkets, primarily in drier climates where a substantial difference in dry-bulb and wet-bulb temperatures exist. The evaporative condenser consists of a tube bundle, a fan for air flow, and a water sump and pump system used to spray water over the tube bundle. The refrigerant vapor is passed through the tube bundle where heat is removed and the refrigerant is condensed. The resulting condensing temperature can be close to the ambient wet-bulb. Evaporative condensers require less air flow than air-cooled condensers of equivalent rejection capability and can, therefore, be operated at a lower minimum condenser temperature without a fan energy penalty.

Water treatment and consumption are major issues that prevent more use of evaporative condensers in supermarkets. Water treatment is needed because of the evaporation of the water, which tends to concentrate dissolved minerals and other solids in the water, which eventually will precipitate and form deposits on tube surfaces. Exposure of the water to air also causes biological growth within the evaporative condensers in the form of algae and slime. Treatment of evaporative condenser water consists primarily of "blowdown" in which a fraction of the water is discharged to the drain and replaced with fresh water. The discharged water will carry away excess minerals and solids, which prevents solids precipitation. Biocides, such as chlorine, are also added to the water by automatic drip systems to prevent biological growth.

Water-cooled condensers are used primarily in urban stores where easy access to the outside for condensers does not exist. Stand-alone compressors or compressor racks are equipped with water-cooled condensers and glycol/water loops are used to circulate liquid between the condensers and a fluid cooler. The water/glycol loop is closed and is not exposed to the air so that algae and other biological formation do not occur. The only water treatment needed consists of maintenance of pH and the addition of corrosion inhibitors in the loop, which need be done on a very limited bases at time intervals of one year or greater.

The fluid cooler can be either air or evaporatively cooled. Air-cooled units require less maintenance since no water is exposed to the air. The use of a water loop with a dry fluid cooler results in higher condenser temperatures than seen with air-cooled condensing, because of the added temperature difference needed to transfer heat to the water. The dry fluid cooler must operate at a temperature higher than ambient dry-bulb in order to reject heat. Air flow and fan power requirements are similar to those seen with air-cooled condensers. Evaporatively cooled fluid coolers operate similarly to evaporative condensers and reject heat at a temperature close to the ambient wet-bulb. Water temperatures less than ambient dry-bulb can be obtained, which can reduce the condensing temperature to a value similar to that achieved with air-cooled condensing. Fan power requirements are of the same order as seen with evaporative condensers. Evaporative fluid coolers employ the same water treatment as is used with evaporative condensers.

Condenser control consists of maintaining a minimum condensing temperature based on a pressure reading seen at the condenser inlet. The condenser fans are cycled based upon a set point value of this pressure.

Pressure regulators are sometimes used to regulate the pressure of the condenser, often in combination with other control valves for functions such as hot gas defrost or heat reclaim. The regulator restricts liquid flow from the condenser, which builds the liquid level, causing the pressure to rise. Some subcooling of the liquid can occur when the flow is regulated in this fashion. Fan cycling is often used in conjunction with the pressure regulator for condenser pressure control.

Section 2 Reference

2-1. Walker, D.H., and G.I. Deming, *Supermarket Refrigeration Modeling and Field Demonstration,* Foster-Miller, Inc., EPRI Report No. CU-6268, Electric Power Research Institute, Palo Alto, CA., March, 1989.

3. ADVANCED REFRIGERATION SYSTEMS

The advanced supermarket refrigeration systems described here were designed specifically to reduce the amount of refrigerant needed for operation. Four such systems were identified consisting of low-charge multiplex, distributed, secondary loop, and advanced self-contained. The low-charge multiplex system is designed to limit the refrigerant charge to the minimum needed for operation. The distributed refrigeration system has the compressors located close to the display cases in cabinets on or near the sales area. Heat rejection is done with either air-cooled condensers located on the store roof above the compressor cabinets or through the use of a fluid loop that connects the condensers in the compressor cabinets with a fluid cooler on the roof of the supermarket. The secondary loop system employs a secondary fluid loop to refrigerate the display cases. A central chiller in a machine room, away from the sales area, is used to cool the secondary fluid loop. Heat rejection for secondary loop systems can be done with air-cooled or evaporative condenser, or by a fluid loop. The advanced self-contained approach employs compressors in 1 to 3 display cases and uses a fluid loop, like the distributed system, for heat rejection.

3.1 Low-Charge Multiplex Refrigeration

Several refrigeration system manufacturers now offer control systems for condensers that limit the amount of refrigerant charge needed for the operation of multiplex refrigeration. Figure 3-1 shows an example of such a control approach. A control valve is used to operate a bypass from the condenser liquid line in order to maintain a constant differential between the high and low pressures of the system. The refrigerant liquid charge is limited to that needed to supply all display case evaporators. No added liquid is needed for the receiver, which is included in the system, primarily for pump-down during servicing. All refrigerant liquid bypassed is expanded and evaporated through heat exchange with the discharge manifold. The resulting vapor is piped to the suction manifold for recompression and return to the condenser. The use of this control approach reduces the charge needed by the refrigeration system by approximately 1/3.

The control of the liquid charge by this method offers some energy-saving potential, because it has been found that compressors can be operated at very low head pressures when this control method is employed. The minimum condensing temperature values suggested for this low-charge system are 40 and 60°F for low and medium temperature refrigeration, respectively (3-1).

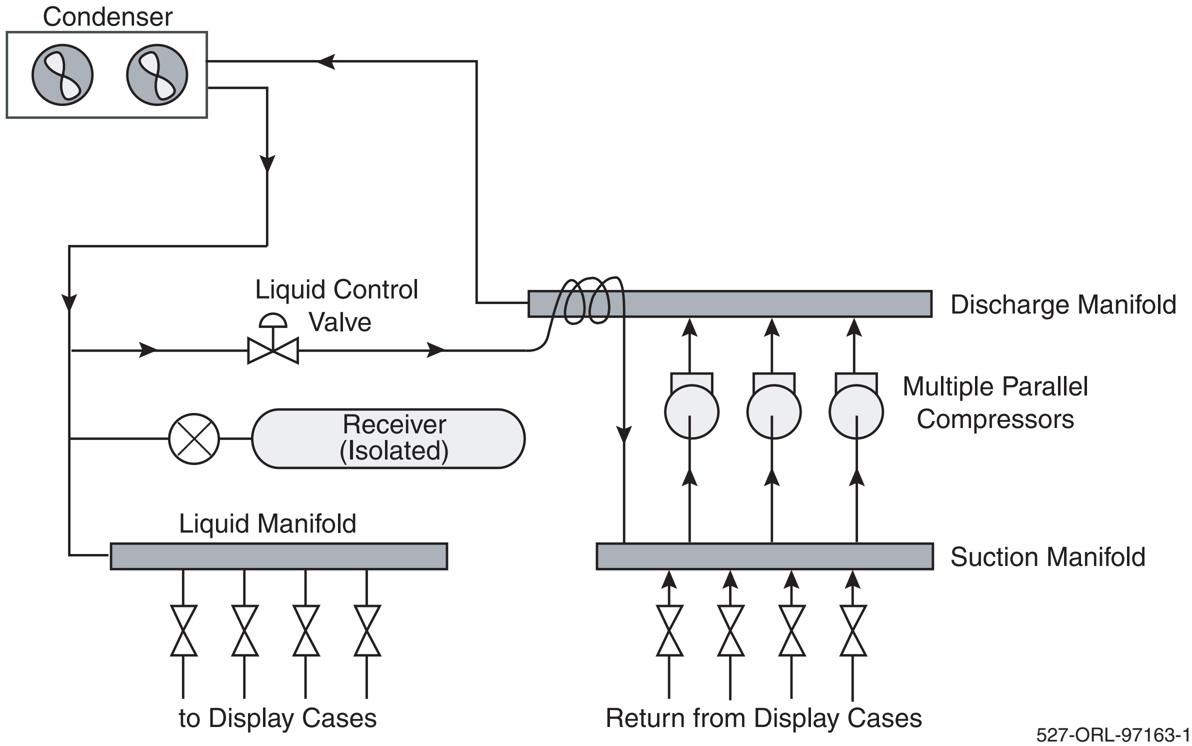

Figure 3-1. Piping diagram for the low-charge multiplex system

3.2 Distributed Refrigeration

Figure 3-2 shows a diagram of the distributed refrigeration system. Cooling of the display cases is provided by direct expansion coils as is done in present supermarkets. The difference is that the long lengths of piping needed to connect the cases with the compressor racks have been eliminated. The compressors are located in cabinets that are close-coupled to the display case lineups. The cabinets are placed either at the end of the case lineup or, more often, behind the cases around the perimeter of the store.

Figure 3-3 shows typical locations of the compressor cabinets in a supermarket. The cabinets are located within the store to provide refrigeration to a particular food department, such as meat, dairy, frozen food, etc. With this arrangement, the saturated suction temperature (SST) employed for each rack closely matches the evaporator temperature of the display cases and walk-in coolers. This is not always the situation seen with multiplex, since a single rack will often provide refrigeration to display cases with three or four different evaporator temperatures. The multiplex system must operate at a SST value that will satisfy the temperature requirements of all display cases connected. The better temperature matching seen with distributed refrigeration benefits the energy consumption of the system.

The refrigerant charge requirement for the distributed system is much less than is seen for multiplex refrigeration. The reduction in charge is due to the shortening of the suction and liquid

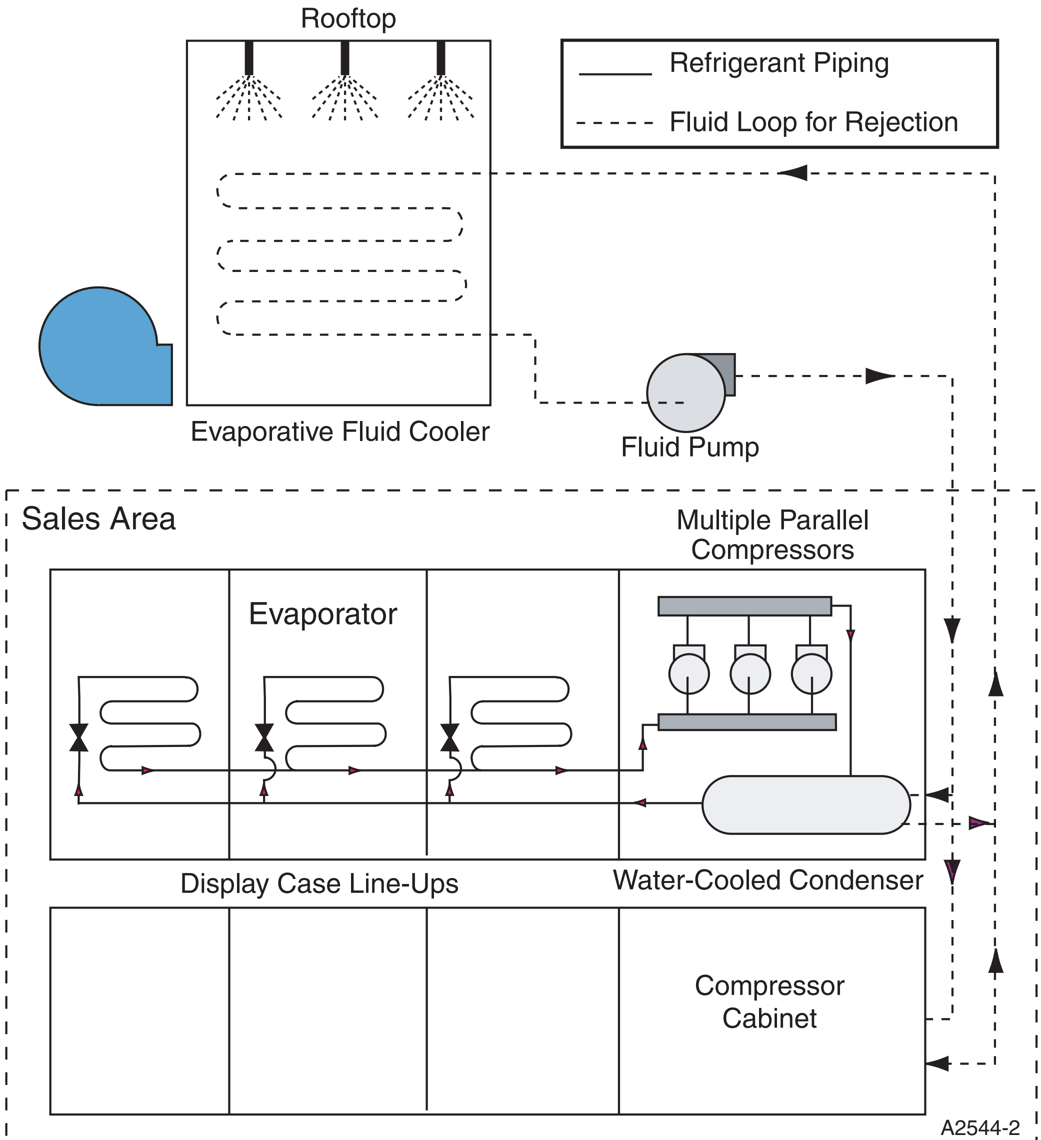

Figure 3-2. Description of the distributed refrigeration system

lines to the display cases. And the elimination of the refrigerant heat rejection piping to a remote condenser. The refrigerant charge associated with each compressor cabinet is about 90 lb. Since 9 to 10 cabinets are needed to provide all refrigeration in a supermarket, the total refrigerant charge is 810 to 900 lb.

Each compressor cabinet is similar to a multiplex rack. All necessary electrical and piping connections are provided within the cabinet, such that the only field connections are the refrigerant liquid and suction lines, fluid inlet and outlet piping for heat rejection, and electric service wiring. Multiple compressors are employed, which are piped in parallel so that multiplex operation can be used for capacity control of the refrigeration. Different size compressors are employed so that a mix and match approach can be used to maintain the desired suction pressure

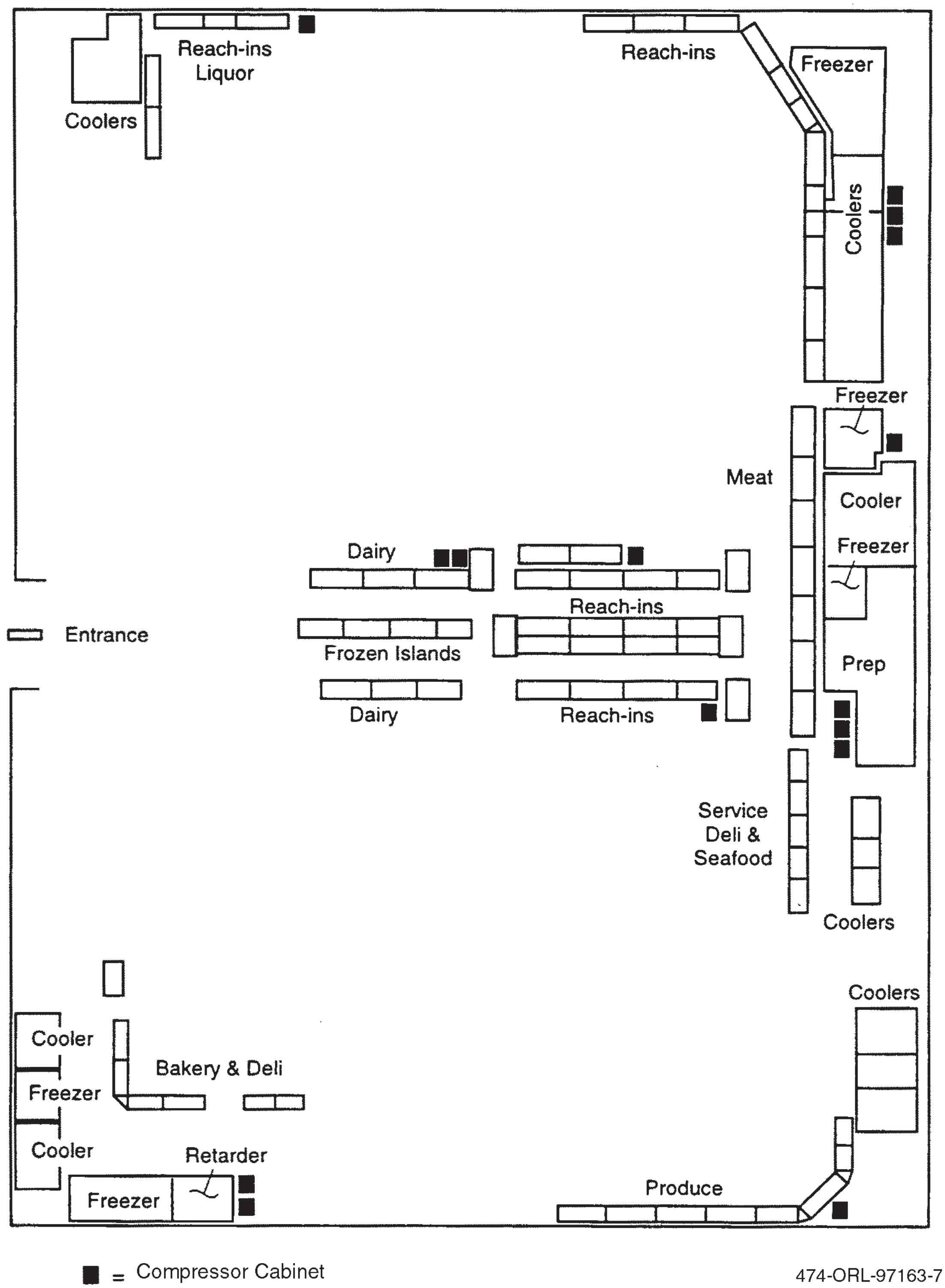

Figure 3-3. Supermarket layout using a distributed refrigeration system

set point. Usually, 3 to 5 compressors are installed in each cabinet. The cabinets are equipped with discharge and suction manifolds for parallel piping of the compressors. The suction manifold can be divided so that multiple suction temperatures are provided from a single cabinet.

The distributed refrigeration system cabinets can be equipped with hot gas defrost. The method used consists of a 3-pipe approach where a hot gas manifold is fed from the discharge manifold, and each evaporator in the system is equipped with a hot gas line that is controlled by a solenoid valve. During defrost, the hot gas flows down the hot gas line of the particular case being defrosted. The resulting condensed liquid refrigerant is returned to the cabinet by the liquid line.

The distributed refrigeration system employs scroll compressors, because of the very low noise and vibration levels encountered with this type of compressor. These characteristics are necessary if the compressor cabinets are located in or near the sales area. The scroll compressors offer several features that can produce significant energy savings for refrigeration operation. The compressors have no valves, and, therefore, can be operated at significantly lower condensing temperature. The lowest condensing temperature possible is at a suction-to-discharge pressure ratio of 2, which means for supermarket systems that the lowest condensing temperature possible is on the order of 55 to 60°F.

The scroll compressors are capable of mid-scroll injection of refrigerant vapor, which can be used to subcool liquid refrigerant. The subcooling is done using a heat exchanger in the liquid line that is mounted in the compressor cabinet. A portion of the liquid is taken from the liquid line and is expanded into the exchanger to provide cooling for the remaining liquid. The vapor generated at the heat exchanger is piped to the scroll compressors at their injection ports. The performance obtained by mid-scroll injection subcooling is determined from manufacturer's performance data for the scroll compressor when subcooling is applied. The amount of vapor injection is limited and the resulting subcooled liquid temperature will vary from 15 to 20°F below the liquid temperature at the outlet of the condenser.

The close-coupling of the display cases to the distributed refrigeration cabinets has other ramifications to energy consumption. The shorter suction lines mean that the pressure drop between the case evaporator and the compressor suction manifold is less than that seen with multiplex systems, which means that the SST of the cabinet will be close to the display case evaporator temperature. Typically, the SST of multiplex racks will be 2 to 4°F less than the case evaporator temperature. SST values for the distributed system will be about 1 to 2°F less than the case evaporator temperature. The shorter suction lines also mean that less heat gain to the return gas is experienced. The cooler return gas has a higher density and results in higher compressor mass flow rates, which means that less compressor on-time is needed to satisfy the refrigeration load. The return gas temperature rise seen for the distributed refrigeration system is normally on the order of 5 to 15°F, depending on the distance between the cabinet and the display cases and the evaporator temperature of the display cases. A greater return gas temperature rise is seen in low temperature systems than is seen in medium temperature. In comparison, the return gas temperature rise seen with multiplex systems falls between 40 and 65°F, due to the longer length of suction lines employed. The liquid temperature can also be affected detrimentally by line length if subcooling is employed. Heat gain to a subcooled liquid line will result in a rise in liquid refrigerant temperature before reaching the display cases.

The layout of a glycol/water loop for heat rejection for a distributed refrigeration system is shown in Figure 3-4. A central pump station is provided that contains the circulation pump and all valving needed to control fluid flow between the cabinets and the fluid cooler. Flow to and from the fluid cooler and pump station is provided by single inlet and outlet pipes sized for the entire system flow. The flow to each of the compressor cabinets is branched from these central supply and return pipes. The flow rate through a compressor cabinet is on the order of 10 to 30 gpm, depending upon the amount of refrigeration provided by the cabinet. The total flow in the heat rejection loop is about 300 to 350 gpm. This flow rate is continuous. Flow to each cabinet is controlled by manual balancing valves that are set at installation to ensure proper flow to each cabinet.

The lowest ambient temperature anticipated during operation determines the glycol concentration of the loop. The typical value of concentration is 25 to 33 percent, which is appropriate for ambient temperatures of 0 and –20°F, respectively.

3.3 Secondary Loop Refrigeration

Figure 3-5 shows the elements of a secondary loop supermarket refrigeration system. The difference between the secondary loop and direct expansion systems is that the display cases and storage coolers are refrigerated by a chilled secondary fluid loop. The secondary fluid is piped to

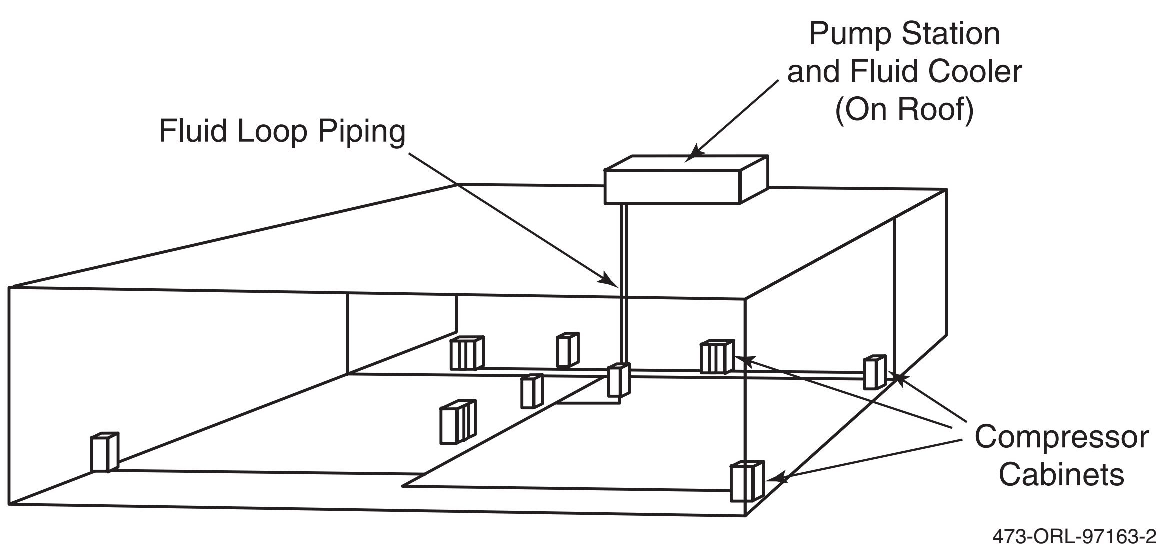

Figure 3-4. Rejection loop pipe layout for the distributed refrigeration system

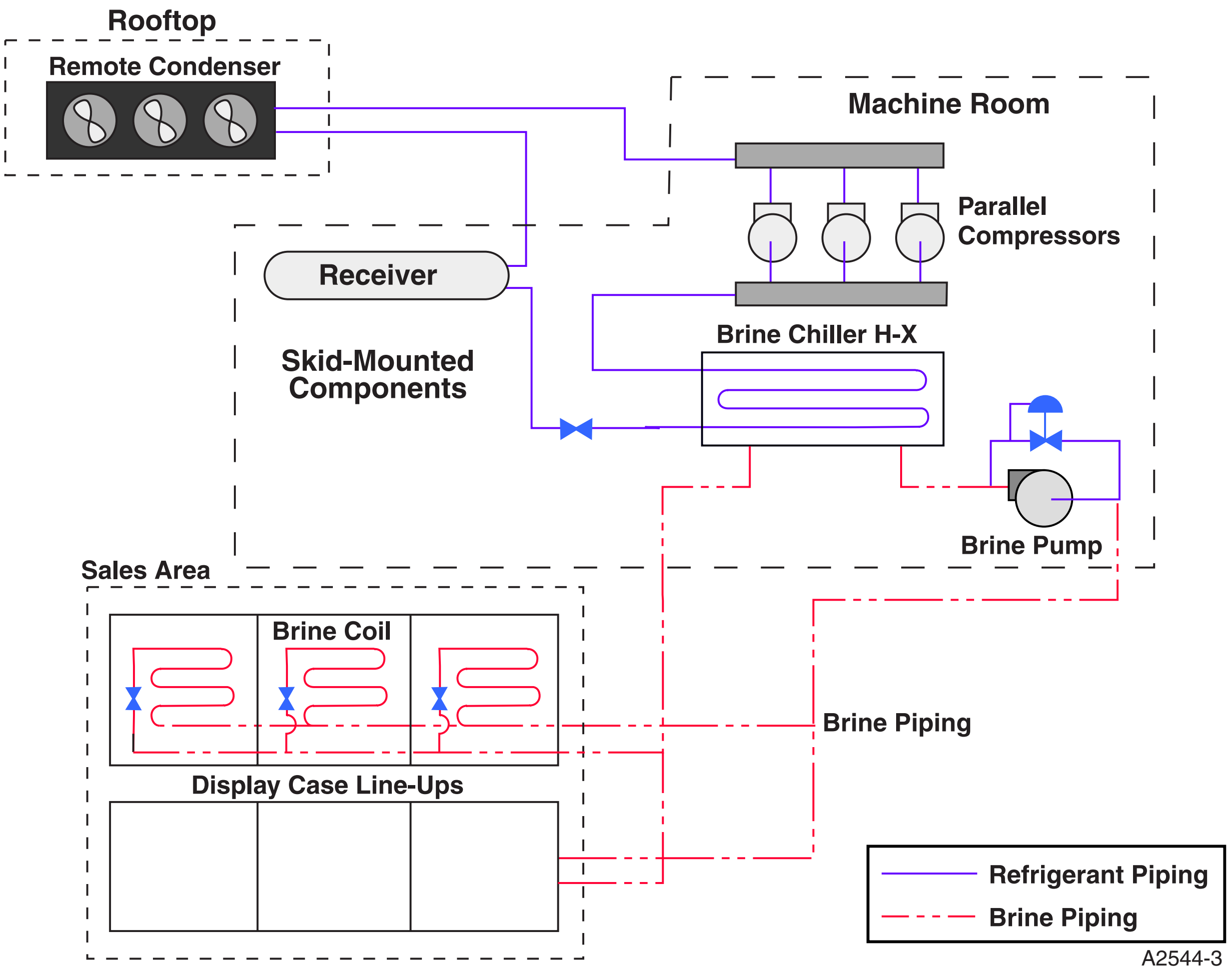

Figure 3-5. Elements of the secondary loop refrigeration system

heat exchanger coils in the display cases through which the case air is passed in order to provide cooling. The secondary fluid is chilled by a central system located in a machine room away from the sales area. The chiller is similar in configuration to the multiplex compressor racks. The only exception is that an evaporator is included with the chiller to provide refrigeration to the secondary fluid. The chiller package may also contain the pumps for the fluid loop. The multiple compressors are piped in parallel, and the suction of the compressors removes refrigerant vapor from the evaporator. The discharge of the compressors is to a common manifold and the discharge gas is piped to a remote condenser, normally located on the roof above the machine room. Secondary loop refrigeration can be configured to operate with air-cooled, evaporative, or water-cooled condensers. The use of evaporative condensing can produce the lowest average condensing temperature with lower fan energy than seen with air-cooled condensers.

The secondary loop refrigeration can be configured to operate with 2 to 4 separate secondary fluid loops and chiller systems. In the 2-loop configuration, the secondary fluid temperatures are 20 and -20°F for medium and low temperature refrigeration, respectively. As a consequence of the use of only two temperatures for refrigeration, all display cases and storage coolers must operate with these two temperatures. For this reason, the heat exchangers in the cases and coolers must be sized differently than those used for direct expansion operation. Additional loops can be employed if a large fraction of the refrigeration load is addressed by a secondary fluid temperature significantly different than 20 or –20°F. Possible examples of alternate loop temperatures are values of –10, 0, or 15°F, depending upon the air temperature required by these refrigeration loads. The use of multiple secondary fluid loops with temperatures closely matching the case air temperature requirements can result in more energy-efficient operation of the secondary loop system.

Secondary fluid loop piping consists of large main pipes that contain all fluid flow to and from the chiller. The piping is branched to the display case lineups as needed. The piping for the main and branched section is typically sized to have a fluid velocity of 4 to 6 ft/sec. In each lineup, further branching occurs to provide flow to each display case coil. A temperature control valve is used in each case coil to monitor case air temperature and to regulate fluid flow to maintain proper air temperature. The branch piping is also equipped with balancing valves that are used to set the branch flow rate. This adjustment is normally done at installation only.

The piping for the secondary fluid loops can be steel, copper, or plastic. The recommended type of plastic piping is constructed of high-density polyethylene (3-2). Insulation is also required. For the medium temperature piping, the recommended insulation is closed-cell foam (3-2). The recommended thickness of the insulation is 1/2 to 3/4 in. for piping underground or in air-conditioned space, 1-1/2 in. for pipe in non-conditioned space. For the low temperature piping, the insulation should be either styrofoam or polyisocyanurate foam. The recommended insulation thickness for low temperature piping is 1 in. for piping in air-conditioned space and 1-1/2 to 3 in. in non-conditioned space.

Secondary fluid flow rates are fairly high, since it is desirable to limit the temperature change of the fluid to 7 to 10°F while refrigerating the display cases. The resulting total flow rate for each loop is as high as 300 to 500 gpm. Because of the high viscosity of the secondary fluid at refrigerating temperatures and the fact that the fluid is circulated continuously, the energy associated with pumping is substantial and is a major component of the overall energy consumption of the secondary loop system. The flow rate of secondary fluid needed for each display case will vary as the refrigeration load changes. The total flow of fluid is also affected, but pump power remains constant, since control of the loop flow is achieved by bypass of flow around the pump. The discharge and total flow seen by the pump remain constant with this arrangement.

The central chiller systems employ multiple compressors that are piped in parallel. The compressors are multiplexed and are on/off cycled in response to the value of the suction pressure of the chiller evaporator. The chiller system employs either reciprocating or screw compressors. Screw compressors provide large refrigeration capacities needed to provide the refrigeration load associated with each fluid loop. Screw compressors do not exhibit as high an energy efficiency as reciprocating units, but are less susceptible to liquid refrigerant damage and, in general, are considered to be less of a maintenance issue. Reciprocating compressors can be used in the chiller system for better energy use characteristics.

Because of the location of the evaporator on the chiller skid, the compressors for the secondary loop system are considered close-coupled to the evaporator. The pressure drop and return gas heat gain are minimized in this configuration. Both these factors help to reduce compressor energy consumption.

The chiller system for the low temperature refrigeration can be equipped with mechanical subcooling. Two possible configurations can be used to supply the subcooling. The first consists of employing the medium temperature chiller to supply subcooling to the low temperature system. The piping arrangement for this type of mechanical subcooling is the same as that described for the multiplex system. A direct expansion heat exchanger is used to cool the liquid refrigerant used by the low temperature chiller system. The second approach is the use of an economizer heat exchanger with the screw compressors. The heat exchanger uses a portion of the liquid flow to refrigerate and subcool the remaining liquid. The vapor from the heat exchanger is piped to an intermediate pressure suction port on the screw compressor.

Secondary loop refrigeration systems employ hot secondary fluid to provide defrost to the display cases. In this system, a heat exchanger at the chiller is used to heat a portion of the fluid with the discharge gas from the compressors. The hot fluid is piped to the cases using a third pipe. The fluid leaving the defrosted case is returned to the chiller through the common return piping system. The use of hot fluid for defrost has been found to shorten defrost time substantially (3-3) and also reduce the amount of refrigeration associated with pull-down and recovery of the case to operating temperature. It is estimated that the energy use for defrost is about half of what is seen in multiplex systems employing hot gas defrost (3-3).

3.3.1 Secondary Fluids for Supermarket Refrigeration

The most commonly used secondary fluid in both commercial and industrial refrigeration applications is either ethylene or propylene glycol and water. Glycols are preferred because they are inert to all common piping materials and most nonmetallic gaskets and seals. Propylene glycol is also nontoxic and nonflammable and is the most suitable glycol for use in unattended operation. A propylene glycol – water mixture show reasonable heat transfer and pumping properties for medium temperature (+20°F fluid temperature), but have very high viscosity at glycol concentrations needed for low temperature refrigeration.

Commercial products exist for use as low temperature fluids and several investigations have been conducted (3-4,3-5) to identify possible candidate fluids for secondary loop systems. The desired properties of the candidate secondary fluids are that they are noncorrosive to common refrigeration construction materials, nonflammable under normal operating conditions, and are reasonably nontoxic. In addition to these requirements, good transport properties are needed to reduce pumping power. Heat transfer properties are of some concern, but of secondary importance compared to pumping.

The following is a listing of candidate fluids for use in secondary loop systems that have been cited in the literature (3-4,3-5). This list gives the general chemical name of each fluid along with several trade names.

- Propylene glycol/water.
- Potassium Formate/water.
  - Pekasol 50.
  - Freezium.
  - Hycool.
- Inhibited alkali ethanate solution.
  - Tyfoxit.
- Hydrofluoroether.
  - HFE-L-13938.
- Cyclohexene.
  - D-Limonene.
- Polydimethilsiloxane (Silicon Oil).
  - Syltherm.
  - Dowtherm.
- Synthetic Isoparaffinic Petroleum Hydrocarbons.
  - Therminol.

Evaluation of these fluids for use in secondary refrigeration was carried out by ref. (3-5) where the fluids were compared in terms of relative pumping power to the fluid Tyfoxit. The relation takes into account the flow rate of secondary fluid needed to address a refrigeration load Q, which is found from

$$Q = \dot{m}\ c\ \Delta T$$

where

$\dot{m}$ = the mass flow rate of the fluid
c = the specific heat of the fluid
ΔT = the temperature change of the fluid

The velocity of the fluid flow, V, can be determined from

$$\dot{m} = \rho\ V\ A = \rho\ V \frac{\pi D^2}{4}$$

$$V = \frac{4Q}{c \Delta T \pi D^2}$$

where

D = the diameter of the brine piping

The pumping power is found from the relation

$$\text{Pumping Power} = \frac{\dot{m}}{\rho}\,\Delta P$$

$$= \frac{\pi D^2}{4}\,V\,\Delta P$$

where

ΔP = the pressure drop across the fluid loop

The pressure drop is determined from the relation

$$\Delta P = f\,\frac{L}{D}\,\frac{\rho V^2}{2}$$

where

f = the friction factor
L = the total length of pipe

The friction factor, f, is calculated from

$$f = \frac{0.29}{Re^{0.2}}$$

where

Re = the Reynold number of the flowing brine

$$= \frac{\rho V D}{\mu}$$

where

μ = the kinematic viscosity of the brine

The relative pumping power is found by holding the refrigeration load, fluid temperature difference, and pipe diameter and length constant. The above relations are then combined to give an expression for pumping power in terms of fluid properties only relative to a base fluid

$$\frac{\text{Pumping Power}_A}{\text{Pumping Power}_B} = \left(\frac{\rho_A}{\rho_B}\right)^{-1.8}\left(\frac{\mu_A}{\mu_B}\right)^{0.2}\left(\frac{c_A}{c_B}\right)^{-2.8}$$

Figure 3-6 shows the relative pumping power for a number of candidate fluids, where Tyfoxit 1.20 is used as the baseline fluid. The figure shows only those fluids that had a relative pumping power close to that of Tyfoxit. Several fluids showed pumping power requirements that were 1 to 2 orders of magnitude greater. These fluids were Dowtherm, Syltherm, and HFE-L-13938. The fluid showing the lowest pumping power requirement was Pekasol 50 at a concentration of 90 percent.

3.4 Advanced Self-Contained Systems

A self-contained refrigeration system consists of a display case that has its own condensing unit mounted within the display case. Self-contained systems are presently used in supermarkets for a limited number of cases, where the cases are in a location inaccessible to refrigeration piping. An example is a refrigerated beverage case placed at the cash registers for spot sales. Self-contained units are also employed as add-on cases or for temporary display of special sales items.

Present self-contained display cases use small reciprocating compressors and air-cooled condensers. Heat is rejected directly into the sales area. Only a limited number of self-contained units of this type can be employed before noise and heat rejection levels interfere with store operation. Problems of this type caused store designers in the past to go to the remote machine room approach now used in most supermarkets.

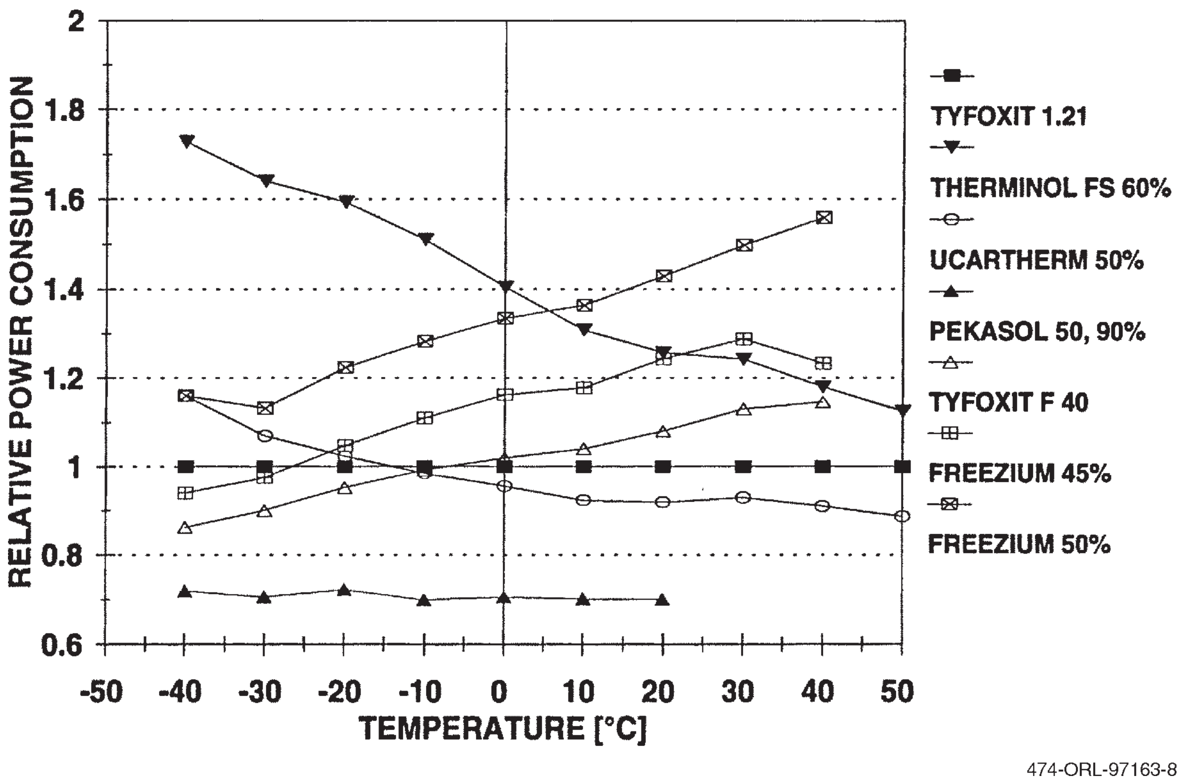

Figure 3-6. Pump power comparison of heat fluids relative to Tyfoxit 1.21 (3-5)

The self-contained system approach could be attractive for reduction of refrigerant charge. It has been estimated (3-6) that the total charge for a supermarket could be reduced to 100 to 300 lb of refrigerant if self-contained systems were used for all display cases.

An advanced self-contained system could be formulated, which used water-cooled condensers and a fluid loop for heat rejection. The fluid loop would be similar to that employed with the distributed system. The use of the fluid loop would eliminate concerns of heat rejection in the store sales area.

The compressor noise issue is still a factor that can be addressed by the use of scroll compressors. Until recently, scroll compressors were available only in a vertical configuration, which was not suitable for placement in display cases. Now, horizontal scroll compressors have been introduced, which could be employed for this purpose. These horizontal scrolls are capable of continuous unloading for capacity control and maintenance of a suction pressure set point. These scroll compressors are capable of unloading to as low as 25 percent of their full load capacity (3-7).

Figure 3-7 shows the relation between capacity control and compressor power required. The use of compressor unloading also allows the condensing temperature to vary as lower temperature heat rejection is possible with lower ambient temperature. For analysis, the minimum condensing temperature was set at 40 and 60°F for low and medium temperature refrigeration, respectively. This may or may not be practical, since 2 glycol loops are needed in order to have two different minimum condensing temperature values.

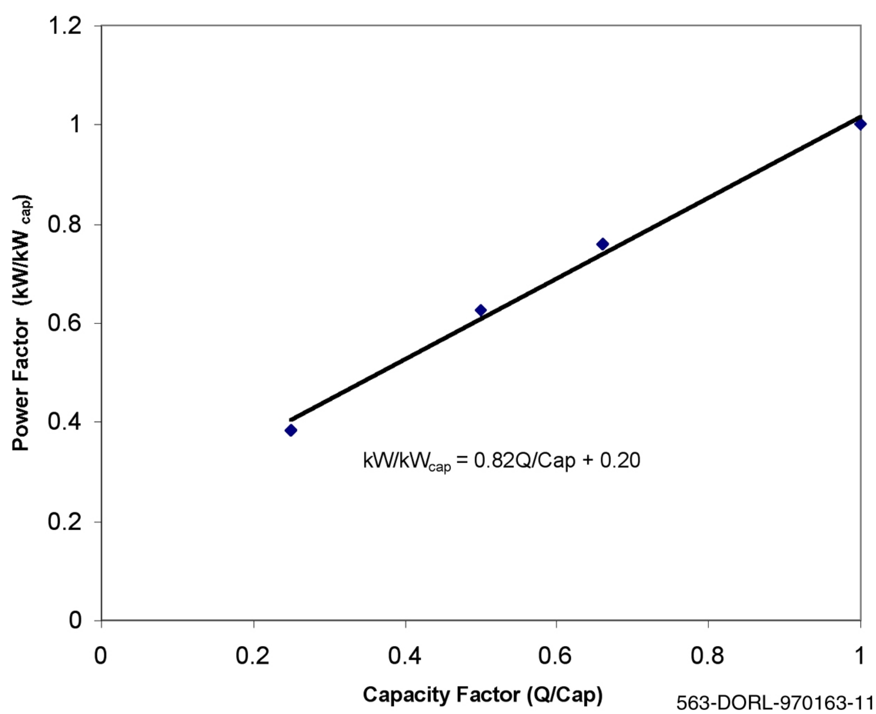

Figure 3-7. Relation between capacity and power used for modeling unloading scroll compressors

The close coupling of the compressor to the case evaporator seen in self-contained systems reduces the pressure drop at the compressor suction and also minimizes the heat gain to the suction gas. Both of these effects will result in more efficient operation and were included in the analysis.

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4. SUPERMARKET HVAC

4.1 HVAC Load Characteristics

Supermarkets have unique HVAC characteristics because of the large amount of refrigerated fixtures operated within the store. The refrigerated display cases remove a large quantity of heat and moisture from the store, which has several impacts on the HVAC requirements.

For space cooling, the cooling loads tend to have a small sensible portion and a large dehumidification need. The display cases are designed to operate at conditions of a 75°F dry-bulb and 55% RH, which is a lower temperature and humidity than normally seen in a commercial building. Humidity in the sales area will tend to increase the refrigeration load on the display cases, and will cause frost formation on the evaporator coils. Defrost of the coils is necessary in order to maintain the capability of providing refrigeration. Defrost increases energy consumption due to the use of electric heaters for defrost, if electric defrost is employed, and also increased compressor run time to lower the display case temperature after defrost. Frozen food door cases are equipped with anti-sweat heaters to prevent fogging of the glass doors. The amount of on time needed by these heaters is directly related to the store humidity level.

Store space cooling systems are often controlled using both a thermostat and humidistat. With this arrangement, the cooling system can be operated for dehumidification when the sensible cooling load has been satisfied. The resulting supply air is colder than desired. Either the store is allowed to cool to a temperature below the set point value, or reheat of the supply air is done to maintain the correct sales area temperature.

Space heating requirements tend to be greater for supermarkets because of the large amount of installed refrigeration. Figure 4-1 compares the heating requirement of a supermarket to that of other commercial buildings. The sensible cooling provided by the refrigeration tends to negate heat gains from store lighting and other electric uses. The heating season for supermarkets is longer than is seen in most commercial buildings for this reason.

To estimate HVAC loads in supermarkets, a so-called “case credit” is calculated to determine the effect of the refrigerated cases. The portion of the refrigeration load that impacts the store consists of the sensible and latent loads removed from the sales area, which is roughly 80 percent of the total case refrigeration load. The remainder of the case refrigeration load is associated with the removal of heat due to the operation of case fans, lights, and anti-sweat heaters. The latent portion of the load falls between 12 and 19 percent of the total case load, depending upon the type of case and evaporator temperature. The highest latent loads occur with either low or

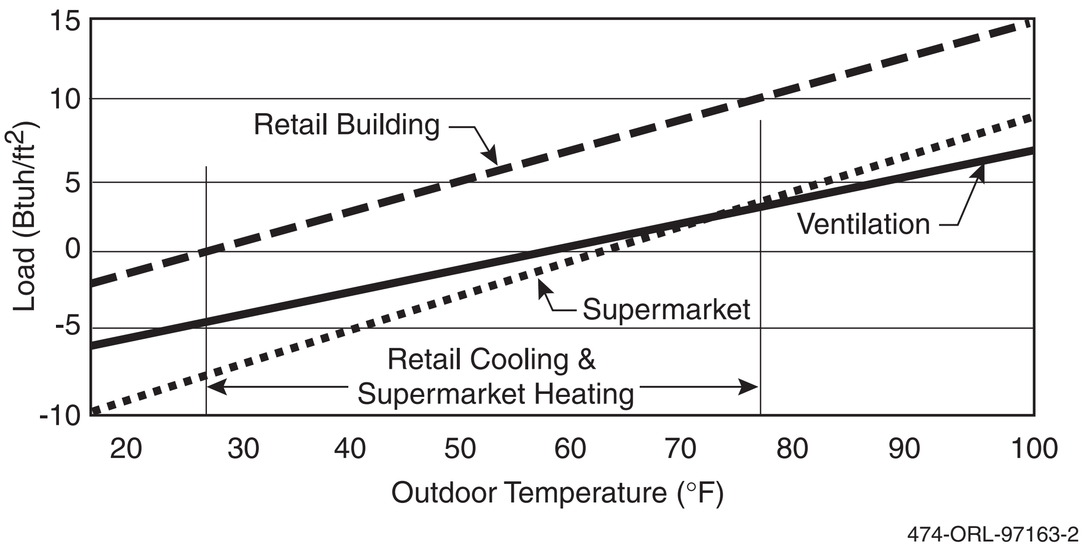

Figure 4-1. HVAC load characteristics for supermarkets and other retail buildings (4-1)

medium temperature multi-deck cases. Latent loads are reduced significantly when door or tub-type cases are employed.

Airflow rate for store air circulation is typically set at 1 cfm/ft^2 of sales floor area. Ventilation flow is set at 10 percent of total circulation, which corresponds to an air change rate of 4.5 per hour. Some stores are investigating the use of higher ventilation rates for improved indoor air quality. To meet the ASHRAE Standard 62-89, the ventilation flow rate to the store must increase to 30 percent of total circulation. This increase in ventilation flow has a significant impact on store HVAC requirements.

Supermarket air systems attempt to maintain the store under a positive pressure to eliminate infiltration. No exhaust flow is used in the HVAC other than that associated with cooking hoods, etc.

The refrigeration system is also a potential source of heat for space heating. Heat reclaim systems have been used in many stores where a portion of the reject heat from the refrigeration system is sent to a coil in the air ducting and is used to heat store supply air.

4.2 HVAC System Configurations

The layout of HVAC supermarket equipment falls into two configurations. The first consists of one or two air handlers located in the back of the store, with each air handler and its ducting operating as its own HVAC zone. The air handler (Figure 4-2) consists of a blower, and heating and cooling coils. Either one or two heating coils are employed with one being for refrigeration heat reclaim. The second coil consists of a duct heater that is either electric or gas fired. The cooling coil is placed upstream of the heating coils so that reheat of the supply air can be

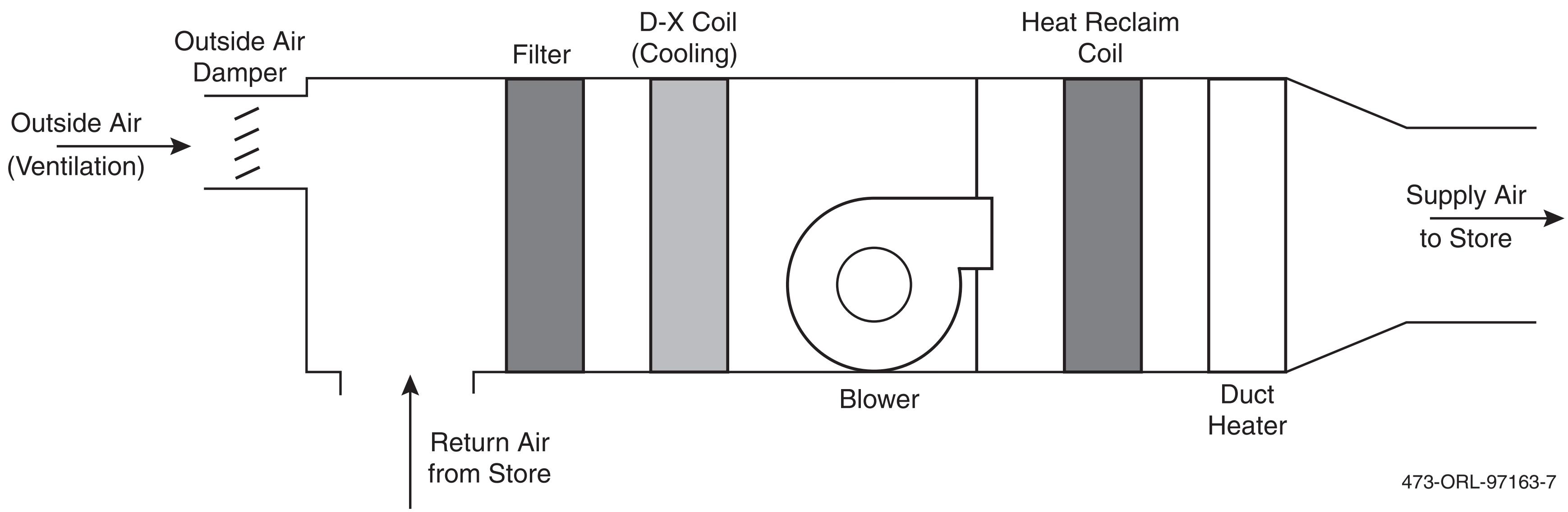

Figure 4-2. ***Elements of a supermarket air handler***

performed if desired. Ducting is provided to the air handler so that most of the airflow consists of recirculated air from the sales area. Ventilation air ducting is connected to the suction side of the air handler and a manual damper is employed to set the amount of ventilation air supplied. The air conditioning condensing unit for the cooling coil is located on the roof above the air handler. Supermarkets will have the air handlers mounted indoors in the duct systems with the heating and cooling equipment mounted on the roof above the air handlers. The alternate location for HVAC equipment is in a "penthouse" machine room located on the roof of the store. These penthouse systems are factory assembled and placed on the store roof where they are mated to the supply and return ducting. The penthouse will contain all elements of the HVAC system, including air handler elements and heating and cooling coils.

The second configuration, which has become more common, is the use of rooftop units (RTUs) at 4 to 8 locations on the roof above the sales area. Figure 4-3 shows a typical RTU used for commercial HVAC. Each RTU is equipped with an air handler consisting of a blower and ducting for supply, return, and ventilation airflows. The supply air duct contains heating and cooling coils. The RTU also has an air conditioning condensing unit with 2 to 4 compressors and an air-cooled condenser. Space heating is provided by a gas-fired duct burner.

4.3 Airflow Path Configurations

The most common airflow path used in supermarket HVAC is the single-path which is shown in Figure 4-4. This airflow configuration is used in most central and rooftop systems. The two airflows for return and ventilation air are joined at the suction side of the blower and then passed through the HVAC coils.

An alternate method now used in some installations is the dual-path approach, which is shown in Figure 4-5. Here the ventilation air stream is conditioned prior to mixing with the return air by a cooling coil located in the ventilation ducting. After mixing, the airflow is then directed through a second set of coils where final conditioning of the air is performed. The advantage of the dual-path method is that the latent load fraction associated with the ventilation air is often much higher than that of the return air. By dehumidifying the ventilation air and then mixing the amount of final conditioning needed is greatly reduced, particularly in terms of reheat. The use of dual path can eliminated the need for reheat in most applications. This method can be used with either vapor compression cooling or with desiccant dehumidification.

4.4 HVAC System Equipment

Standard supermarket HVAC systems consist of vapor compression air conditioners for space cooling, and gas duct heaters for space heating. These components are the same as those employed for most commercial and retail applications.

4.5 Desiccant Dehumidification

The use of desiccant dehumidification has been investigated for supermarkets. The advantage offered by desiccant systems is that they can address the latent load on the store

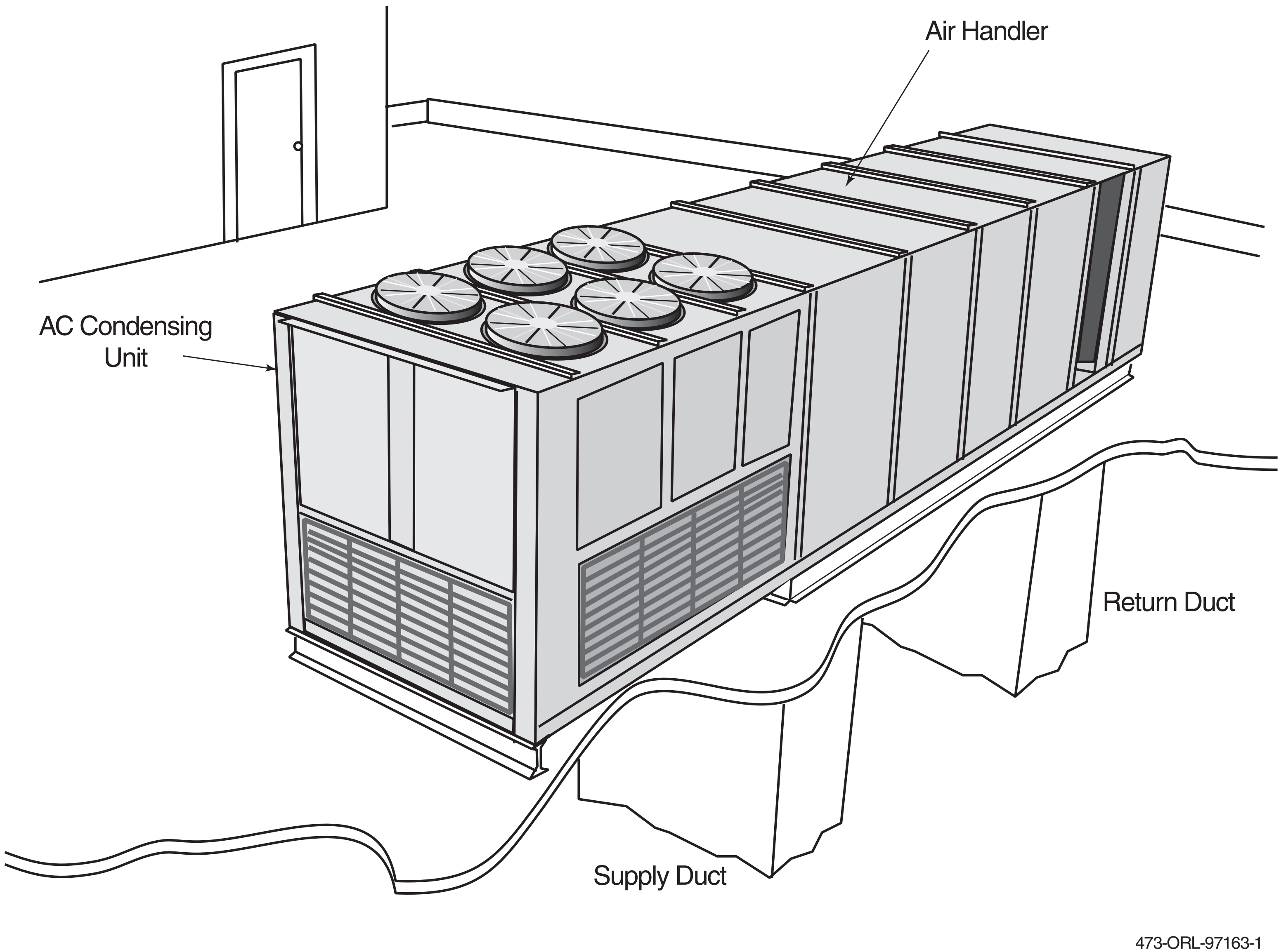

Figure 4-3. *Rooftop unit for HVAC*

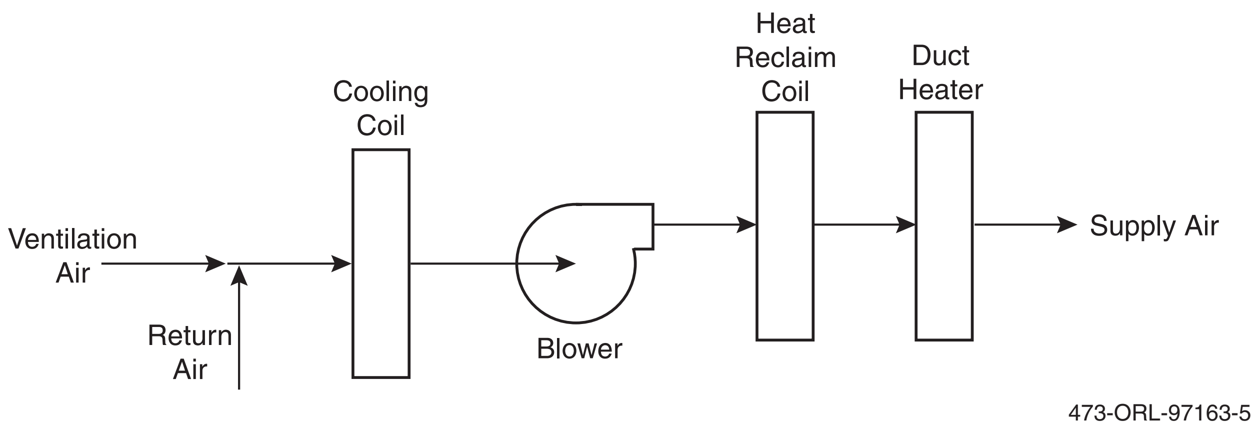

Figure 4-4. Single-path HVAC

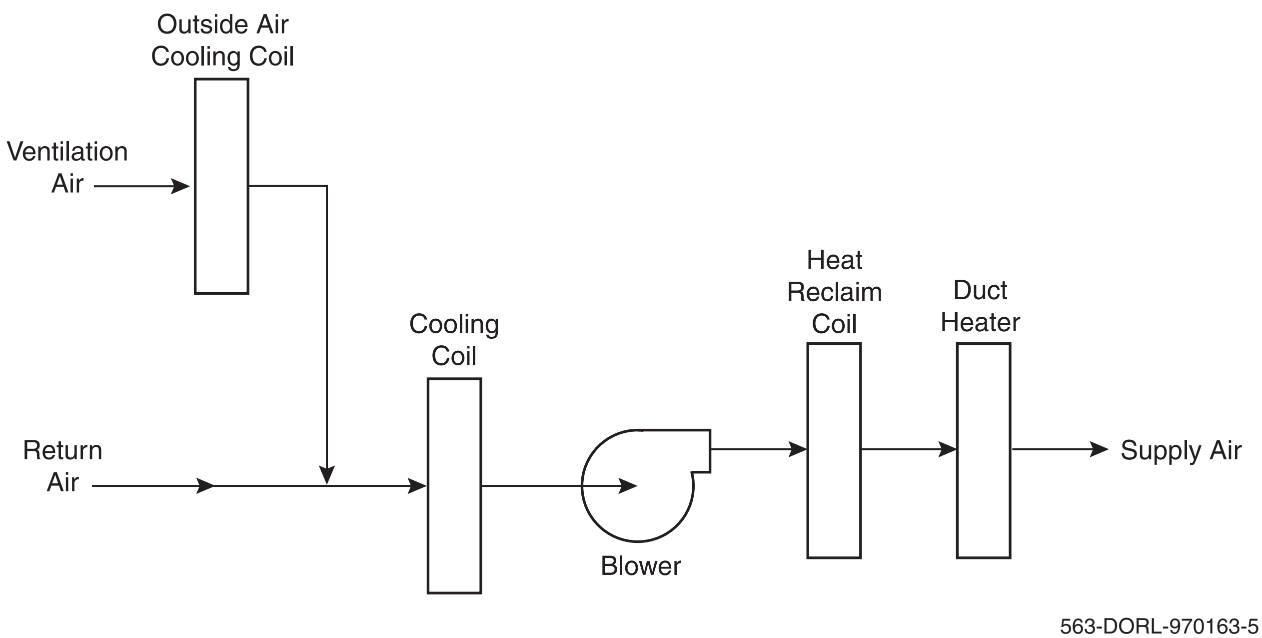

Figure 4-5. Dual-path HVAC

separately and convert the latent heat into a sensible cooling load that can be addressed as needed by a vapor compression cooling system.

The elements of a desiccant dehumidification system are shown is Figure 4-6. The air to be conditioned is passed through the desiccant wheel where moisture is adsorbed and the latent heat is released back to the air. Most desiccant systems also have a second wheel that is used as a rotary heat exchanger to cool the dried air and to preheat the regeneration air, which passes through the desiccant wheel in order to removes the moisture from the wheel. The regeneration air is passed through a gas-fired heater where it is heated to 180 to 220°F. Refrigeration heat

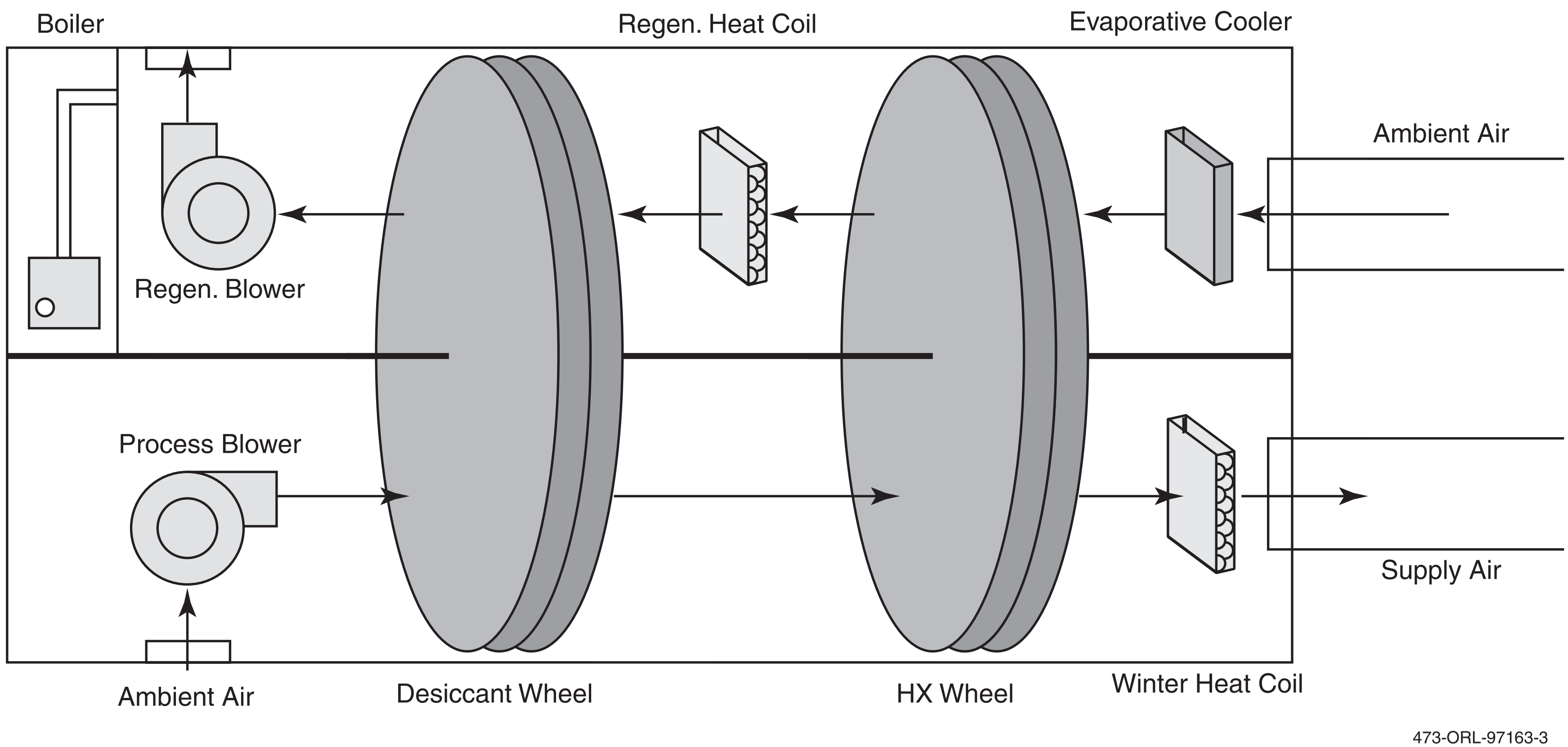

Figure 4-6. Elements of a desiccant dehumidification system

reclaim is employed on some systems to provide heat to the regeneration air, which helps reduce the amount of gas needed for operation.

Desiccant systems are used in a dual path arrangement where the desiccant system conditions outside air only. The air is dried thoroughly and is then mixed with the recirculation air. The air mixture is normally passed through a cooling coil to remove any excess sensible heat. One major advantage to this approach is that the need for reheat of the conditioned air is avoided.

A number of analyses have been carried out to evaluate desiccant systems in supermarkets (4-2 - 4-6). The results are extremely varied and totally dependent on local rates for electricity and natural gas. The general findings are that the amount of energy needed to dehumidify the entire store to a lower level is difficult to justify. While the refrigeration loads are reduced when humidity levels in the store are lowered, the reduction in refrigeration energy is not large enough to offset the costs associated with the added HVAC operation.

A case study done by the HEB supermarket chain compares desiccant and conventional HVAC systems in two operating supermarkets in the San Antonio, TX area (4-6). Table 4-1 gives a summary of the findings. In general, the desiccant system was more costly to install than the conventional system. Savings achieved by the desiccant system were negated by increased maintenance costs. Savings seen in refrigeration energy were on the order of 3 percent. The payback on the cost premium for the desiccant system was estimated to be 27.3 years.

The more accepted practice now used in a large number of supermarkets is to limit the use of desiccant systems (or any other low humidity system, such as dual path air conditioning) to areas in the store where lower humidity is very advantageous. The best example is the frozen food aisle of the store. The large amount of sensible cooling performed by the frozen food cases tends to cool the aisle too much, making it uncomfortable for customers. The desiccant system works well in this application, because the air from the desiccant system can be discharged directly into

Table 4-1. Comparison of first and operating costs for conventional and desiccant HVAC systems (4-6)

	Conventional Packaged Rooftop Unit	Desiccant
Store Conditions	75ºF, 45% RH	75ºF, 40%RH
Store Airflow (cfm/ft^2)	0.65	0.65
HVAC System First Cost ($)		
Components	48,000	107,000
Installation	12,400	15,640
Duct System (installed)	112,261	116,741
Total Installed Cost	172,661	239,381
Annual Expenses ($)		
Utility Cost		
HVAC	33,880	30,210
Refrigeration	55,366	53,740
Maintenance	1,041	3,895
Total Annual Expenses	90,287	87,845

the frozen food aisle, which helps offset the overcooling done by the cases. The drying of the air is limited to the vicinity of the low temperature refrigerated cases, where maximum benefit to the refrigerated cases is obtained. The frozen food cases are very susceptible to humidity in terms of frost deposition on the cases and product. The door case anti-sweat heaters are normally operated in response to the humidity level in the aisle and the drier air helps to limit heater operation.

4.6 Water-Source Heat Pumps

A water-source heat pump employs a vapor compression cycle to provide either space heating or cooling. Figure 4-7 shows a diagram of the operation of the heat pump. A water loop is used as the heat source for space heating and the heat sink for cooling. The vapor compression cycle is equipped with a valve that controls the flow of refrigerant depending upon the heat pump's mode of operation. For space heating, the water coil is used as the evaporator of the heat pump to remove heat energy from the water loop. The refrigerant is then pumped by the compressor to a higher temperature and sent to the indoor air coil, which acts as the refrigerant condenser. The condensing of the refrigerant provides air heating which is used for space heating. For space cooling operation, the flow of refrigerant is reversed and the indoor air coil serves as the heat pump evaporator. The air is cooled and dehumidified as it passes through the air coil by evaporating the refrigerant. The compressor moves the refrigerant from the evaporator and increases its temperature and pressure. The high-pressure refrigerant flows to the water coil, where the refrigerant is condensed and heat is rejected to the water loop.

The heating performance of the water-source heat pump is characterized by its heating coefficient of performance (COP_H) which is defined as

$$COP_H = \frac{\text{Heat Provided}}{\text{Compressor Work}} = \frac{Q + W}{W}$$

where

Q = the heat absorbed from the water loop
Tw = the work of compression

Values of COP_H for water-source heat pumps suitable for supermarket application fall in the range of 4.5 to 5.0 at an inlet water temperature of 70°F

Space cooling performance is specified by an energy efficiency ratio (EER) which is the ratio of the amount of cooling provided to the compressor power. The rated value of EER for water-source heat pumps is 10.0 Btu/hr/Watt at an entering water temperature of 85°F.

Water-source heat pumps are presently constructed in either single- or dual-path configurations. For the dual-path units, separate heat pumps are employed for the outside and circulated air flows. Each heat pump has dedicated compressors, and water and air coils. A

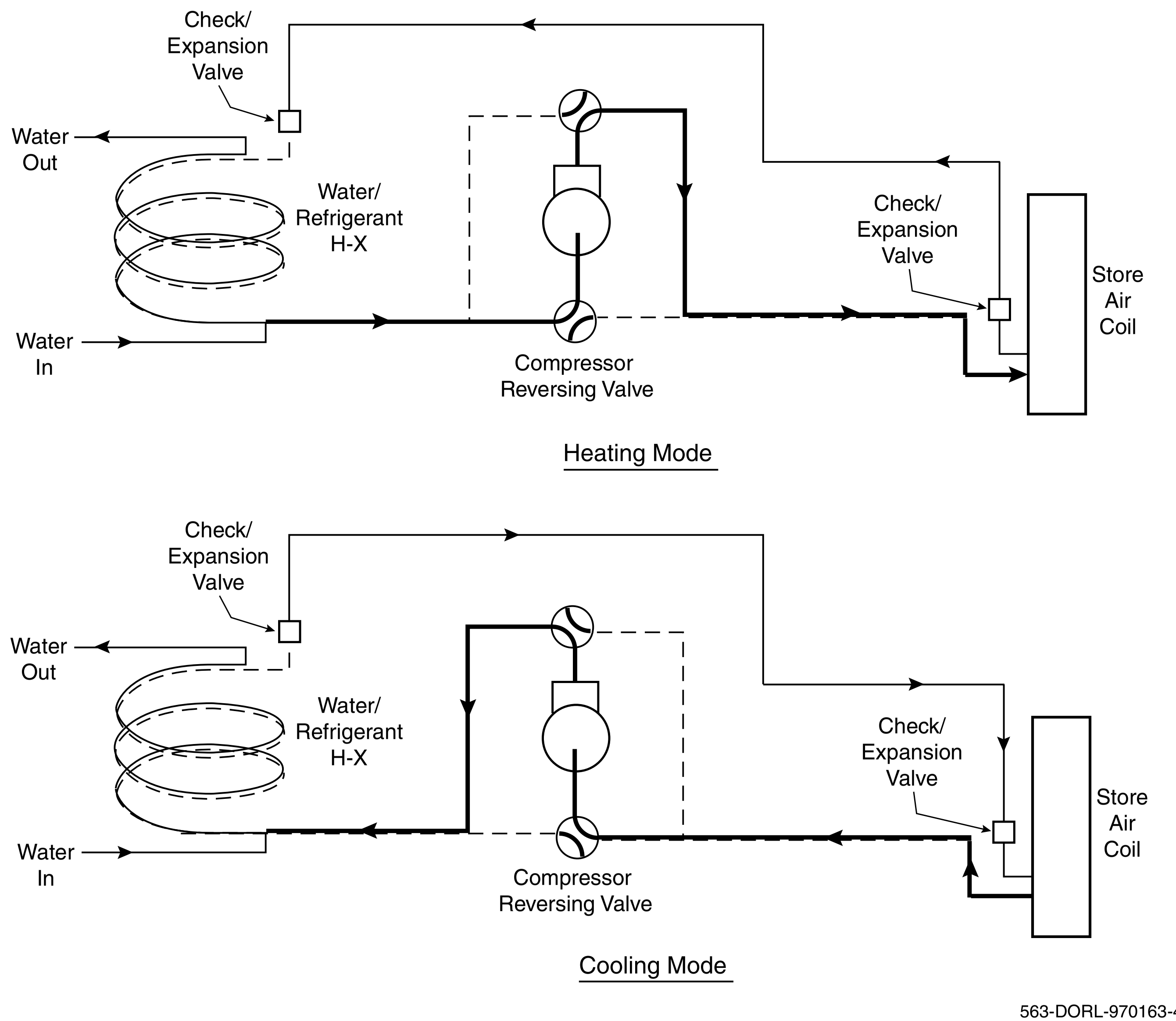

Figure 4-7. Detailed diagram of the water-source heat pump

single fan is used for air circulation and ventilation flows. The flow rate of ventilation air is controlled by a damper.

Water-source heat pumps can be used with supermarket refrigeration systems that employ a fluid loop for heat rejection. The integration of these two systems is shown in Figure 4-8. Figure 4-9 shows a detailed diagram of the water-source heat pump. The water-source heat pump is operated from a separate parallel branch of the fluid loop. In this arrangement, the fluid loop can be used to either provide heat to the heat pump during space heating, or provide heat rejection during space cooling. During heating operation, a control valve is used as a bypass to the fluid cooler to maintain a minimum temperature in the fluid loop.

Section 4 References

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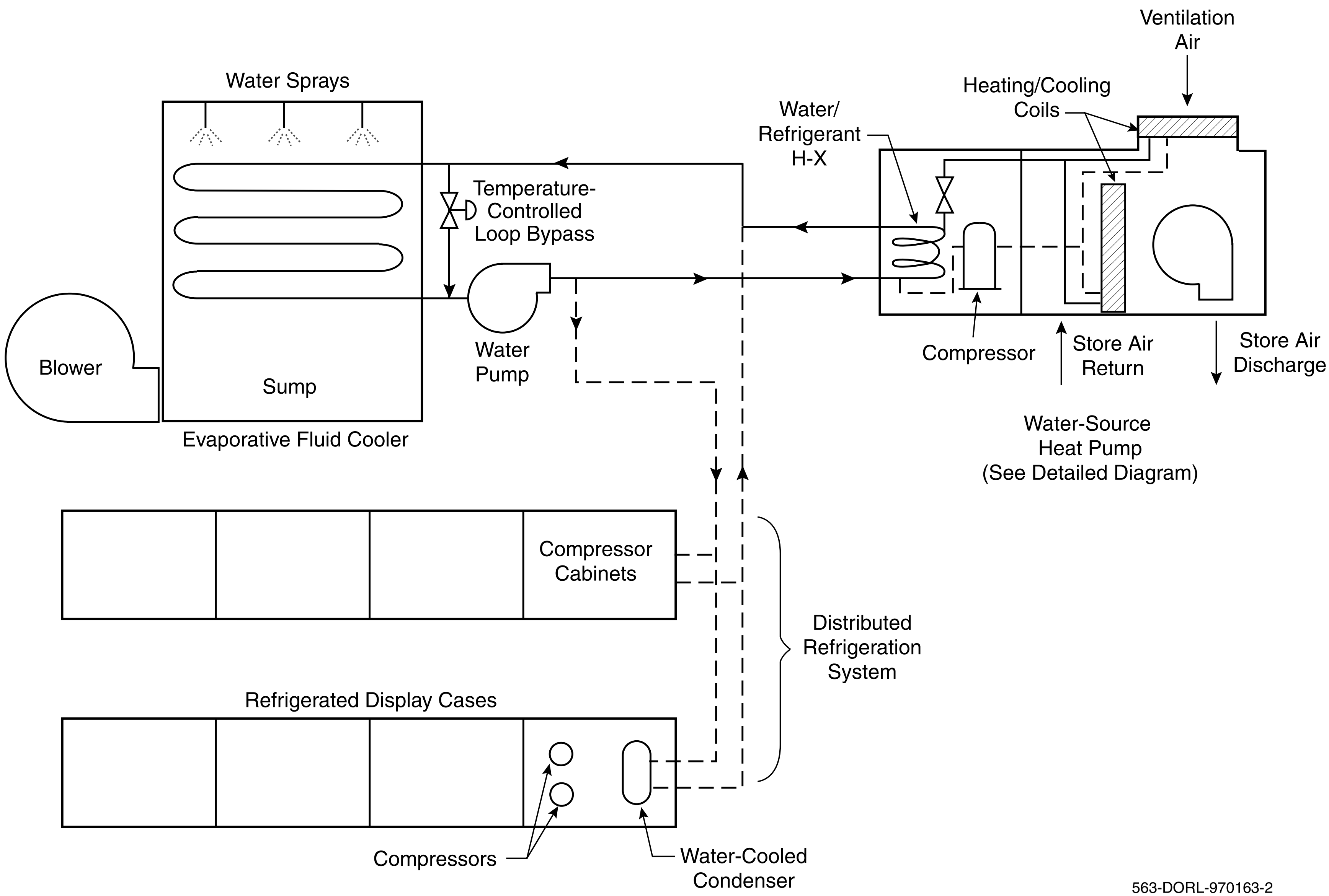

Figure 4-8. Integration of supermarket refrigeration and HVAC systems

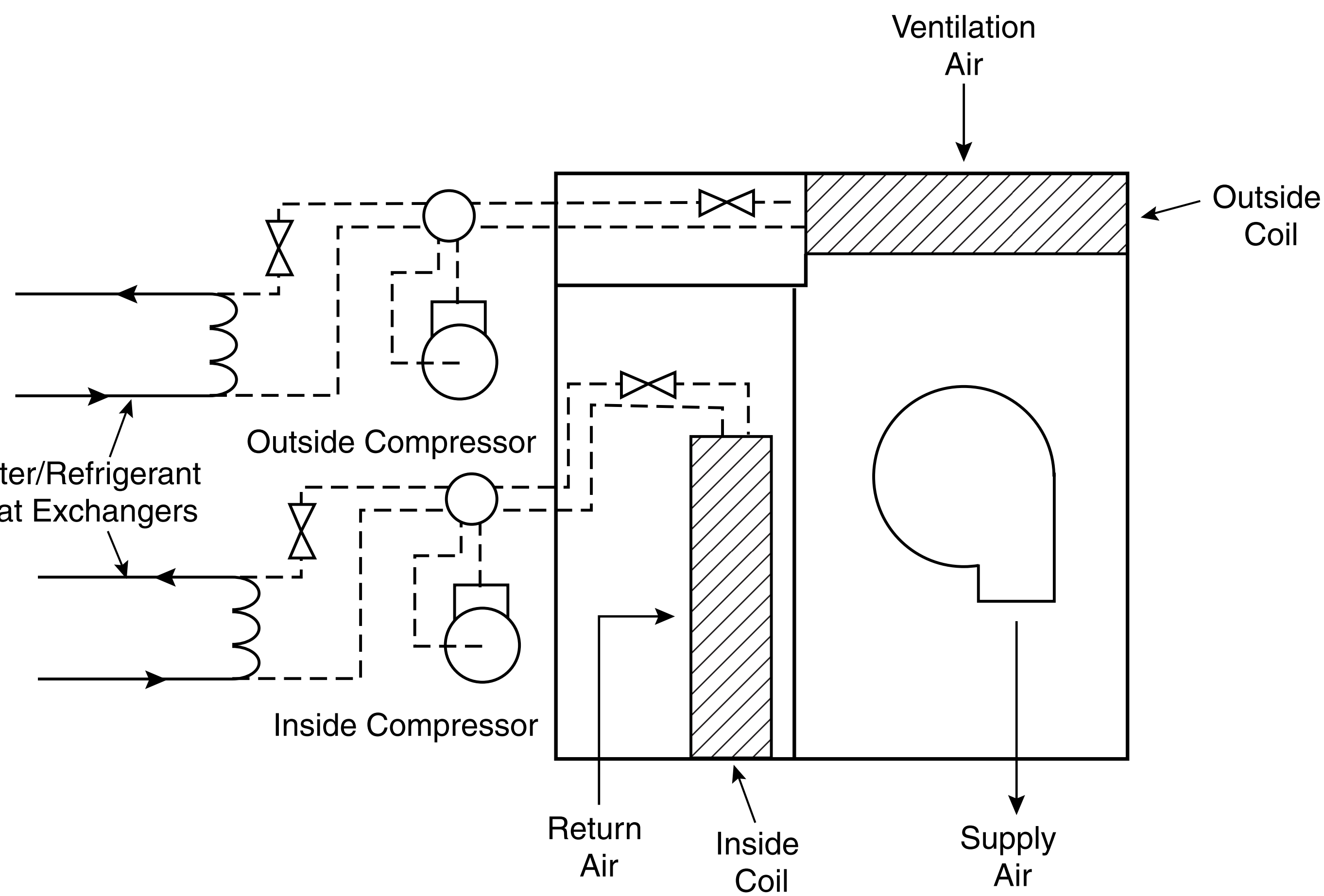

Figure 4-9. ***Detailed diagram of water-source heat pump***

5. ANALYSIS OF SUPERMARKET REFRIGERATION AND HVAC

5.1 Multiplex Refrigeration

Figure 5-1 shows the flow chart of the model employed to analyze refrigeration system performance. Some variations exist in the model, depending upon the type of system examined (i.e., multiplex, distributed, secondary loop, etc.), but the overall procedure and methodology is the same for all. The annual performance is calculated on the basis of ambient dry-bulb temperature bins, where each bin specifies: an ambient dry-bulb temperature value; the coincident value of the wet-bulb temperature; and the number of hours at which the ambient temperature occurs during the year. The general procedure is, for each temperature bin, to calculate the power needed by the refrigeration system and apply that power to the number of hours at each ambient temperature. The procedure is repeated for all ambient temperatures seen at the site.

The bins can be constructed in any temperature increment desired, depending upon the weather data available. For all results presented here, a temperature bin size of 5°F, was employed, and the temperature values and hours observed were obtained from ref. (5-1).

A refrigeration configuration must also be specified, where each refrigeration system employed in the supermarket is described in detail. Table 5-1 lists the information that must be called out for the configuration specification of each refrigeration system. System information identifies the type of system, operating temperature level - low or medium, the design refrigeration load, lowest display case evaporator temperature in the system, minimum condenser temperature, type of condenser (and heat rejection for fluid loops), and the refrigerant employed. Optional information that can be specified include saturated temperature change between the display cases and the suction of the compressors, which is representative of the suction pressure drop, and the temperature rise in the return gas.

The first step in the analysis is to determine the refrigeration load on the system. Past experience (5-2) has shown that the refrigeration load will vary with outside ambient temperature, decreasing as the ambient temperature decreases. The rate of decrease in load is greater for medium temperature than low temperature refrigeration, and a minimum is reached, which is due to the space heating of the store. Store temperature will not fall below 68°F. The minimum store temperature is normally seen at ambient temperatures that are less than 40°F. The maximum values of store temperature and humidity are also constrained by the use of air conditioning, which tends to maintain the temperature at 75°F with a corresponding relative humidity of 55 percent. Maintaining the indoor conditions at these levels can be expected at outdoor ambient temperatures of greater than 85°F. Based on these constraints, a load factor can

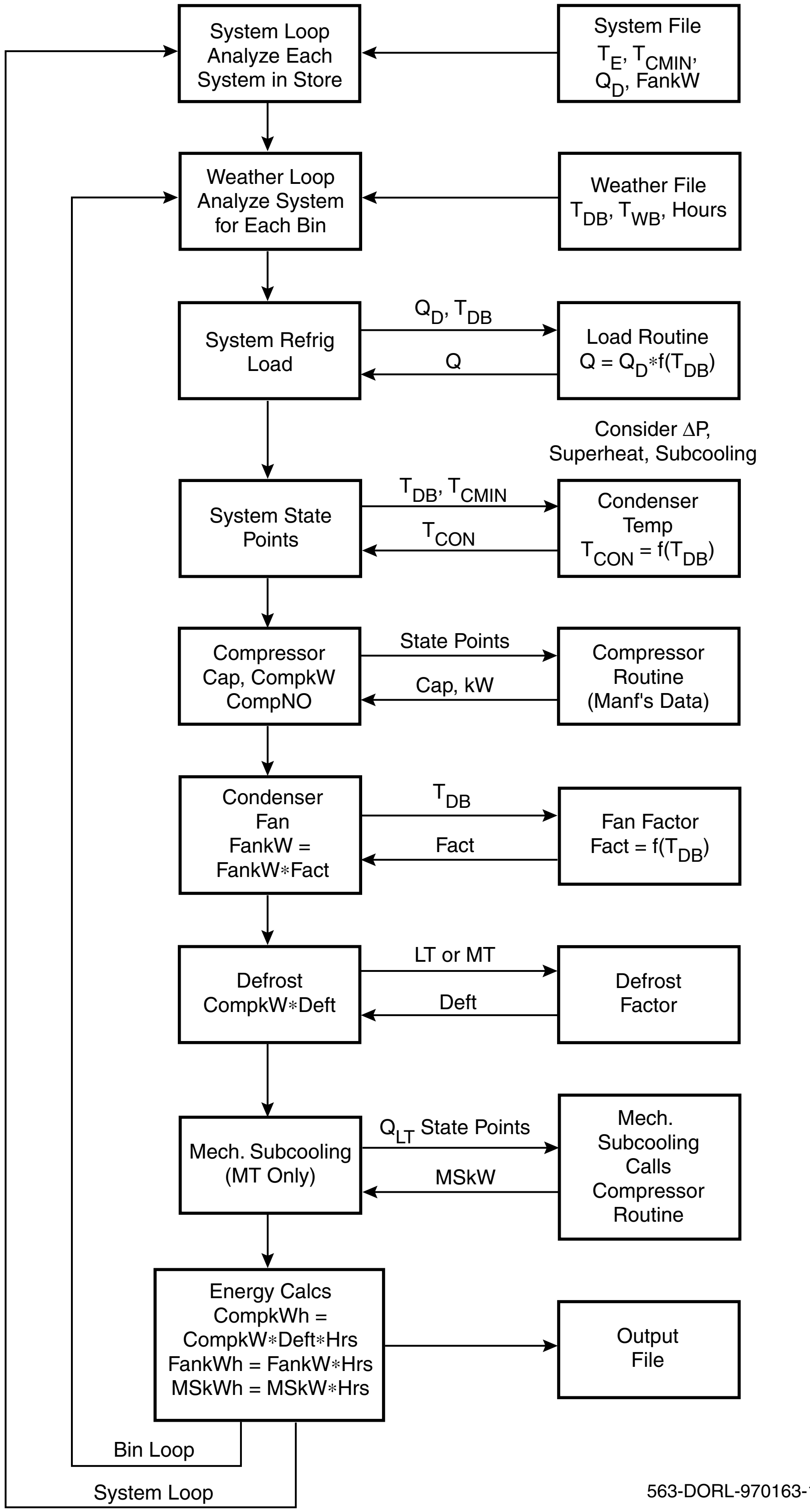

Figure 5-1. Supermarket refrigeration analysis model

Table 5-1. Refrigeration system characteristics needed for analysis model input

Specified Item	Description	Range of Values
Type of System	Type of refrigeration system being examined	Multiplex (incl. Low-Charge) Distributed Secondary Loop Adv. Self-Contained
Operating Temperature Level	Either low or medium temperature	LT – evap temps of –30 to -10°F MT – evap temps of 10 to 35°F
Subcooling	Denotes if mechanical subcooling is employed with this system	SUB – subcooled NS – no subcooling
Refrigeration Load (Q_D)	Design refrigeration load for this particular system	Total of refrigeration loads specified by the case and walk-in cooler manufacturers
Evaporator Temperature (T_E)	Lowest case or cooler evap. temperature associated with the system	
Minimum Condenser Temperature (T_{CMIN})	Lowest condensing temperature employed with the system	Dependent on compressor type Reciprocating Screw Scroll
Condenser Type	Type of condenser employed	AIR – air-cooled EVAP – evaporative WATER – water-cooled
Fluid Cooler Type (Water-Cooled Condensing Only)	Type of fluid cooler employed with water-cooled condensers	DRY – air-cooled WET – evaporative
Refrigerant	Refrigerant employed in system	22,134a,502,404A,507, 407A, 407B
Optional Inputs		
Saturated Suction Temperature Drop	Change in saturated temperature between the case evaporator and the compressor suction (i.e., suction pressure drop)	Specified in °F
Return Gas Temperature Rise	Increase in return gas temperature	Specified in °F

be calculated which is applied to the design load to determine the refrigeration load for each temperature bin. The refrigeration load factor is determined from the following relations:

- For both low and medium temperature refrigeration, the value of the refrigeration load is set at design for ambient temperatures of 85°F or greater.

- At ambient temperatures of 40°F or less, the refrigeration load is at its smallest fraction of design value, which is 66 percent for medium temperature and 80 percent for low temperature.

- For ambient temperatures between 40 and 85°F, the load factor is found from

$$\text{Load factor} = \left(1 - (1 - \text{min})\frac{(85 - T_{amb})}{(85 - 40)}\right)$$

where

min = the minimum fraction of design load (0.66 for medium temperature and 0.8 for low temperature)

T_{amb} = ambient dry-bulb temperature

The state points for the refrigeration system are then determined. The system configuration specifies the desired evaporator temperature for the display cases. In operation, pressure drop will occur between the case evaporators and the compressor suction. This drop is reflected in a lower saturated temperature value at the compressor suction. Heat gain to the return gas will also take place, which affects the mass flow rate of refrigerant seen by the compressor. Both of these factors tend to decrease the capacity of the compressors and increase the run time need to meet the refrigeration load. The amount of pressure drop and superheating is a strong function of the distance between the display cases and the compressors, increasing with increased distance. In the analysis, these factors are taken into account by values included with the system configuration description.

The condenser temperature is also determined at this time. The most significant parameter in determining condensing temperature is the ambient temperature, since heat is rejected by heat transfer to the ambient. The operation of the condenser can be characterized by the temperature difference (TD) between the condensing refrigerant and the ambient. The condensing temperature is allowed to vary with the ambient until a certain minimum condensing temperature is reached. At that point, control of the condenser fans or a liquid pressure regulator maintains the condensing temperature at the minimum value. The model compares the ambient temperature with the characteristic TD of the condenser type specified and calculates a condensing temperature. The calculated value is compared with the specified minimum condensing temperature. If the calculated temperature is less than the minimum, the minimum value is used to set the state point of the refrigeration system.

The condenser TD is determined by the type of condenser modeled. A summary of the characteristics of each condenser and heat rejection system examined here is listed in Table 5-2. For air-cooled condensers, the TD refers to the difference between the condensing and ambient dry-bulb temperatures. The standard values of TD for air-cooled condensers in supermarket refrigeration are 10 and 15°F for low and medium temperature, respectively. The TD of an evaporative condenser refers to the difference between condensing and ambient wet-bulb temperatures. Evaporative condensers are often sized to produce a condensing temperature of 100°F at the design wet-bulb, however, analysis and field measurements by Foster-Miller (5-3) showed that the lowest combined compressor and condenser fan power is achieved at a TD value

Table 5-2. Heat rejection system specifications for refrigeration modeling

System	Temperature Difference ($T_{cond} – T_{amb}$)	Design Load/kW Fan Power (kBtuh/kW)
Air-Cooled Condenser	TD = 10ºF Low temperature TD = 15ºF Medium temperature	34 Low temperature 65 Medium temperature
Evaporative Condenser	TD = 12ºF based on wet-bulb temperature	70.5 Condenser fan and spray pumps
Water-Cooled Condenser Wet Fluid Cooler	TD = 10ºF based on water temperature Tower approach = 12ºF based on wet-bulb temperature	70.5 tower fan and spray pumps
Water-Cooled Condenser Dry Fluid Cooler	TD = 10ºF based on water temperature Tower approach = 12ºF based on dry-bulb temperature	65 tower fan only

of approximately 12°F. This lower value was used for all relevant analysis presented here. The TD of a water-cooled condenser is defined as condenser temperature – inlet water temperature, which is typically about 10°F. The inlet water temperature is affected by the ambient, and whether dry or evaporative heat rejection is employed at the fluid cooler. For dry rejection, the outlet water temperature is 10 to 15°F higher than the ambient dry-bulb temperature. For evaporative rejection the water temperature is about 10 to 15°F higher than the wet-bulb temperature. Water-cooled condensers are specified with either wet or dry heat rejection. The wet system will have a condensing temperature either the same or slightly less than that seen with air-cooled condensing. Use of the dry tower results in higher condenser temperatures, because of the added temperature difference seen across the water loop.

The refrigerant liquid temperature is also determined to set the state points of operation. For non-subcooled systems, the liquid temperature is taken at 10 deg less than the condensing temperature. When mechanical subcooling is employed in multiplex systems, the liquid refrigerant temperature leaving the subcooler heat exchanger is typically 40°F. Some warming of the liquid occurs as the liquid is piped to the display cases such that the temperature of the liquid entering the cases is set at 42°F. The refrigeration needed for mechanical subcooling is normally provided by the highest temperature refrigeration system in operation, typically at a SST of 25 to 35°F. The size of the mechanical subcooling load varies with the load of the subcooled system, normally the low temperature refrigeration. For each temperature interval the low temperature refrigeration load is first determined, and the liquid refrigerant flow required for this load is then determined. The subcooling load is the amount of cooling needed to lower the temperature of the liquid flow from 10°F less than the condensing temperature to 40°F. The subcooling load is added to the refrigeration load of the medium temperature system and is used to determine the energy consumption for that system.

Once the state points are determined, the capacity and power of the compressors is found. These calculations are made using the compressor performance data supplied by the manufacturers. Performance data consists of refrigeration capacities (Btuh), refrigerant mass flow rate (lb/hr), and input power (Watts) as functions of saturated suction temperature (SST) and saturated discharge temperature (SDT). These data are obtained from the ARI equation

$$\begin{aligned}\text{Capacity, Mass Flow, or Power} = {} & C_0 + C_1{*}SST + C_2{*}SDT + C_3{*}SST^2 \\ & + C_4{*}SST{*}SDT + C_5{*}SDT^2 + C_6{*}SST^3 \\ & + C_7{*}SDT{*}SST^2 + C_8{*}SST{*}SDT^2 \\ & + C_9{*}SDT^3\end{aligned}$$

where

$C_0 \ldots C_9$ = Performance equation coefficients
SST = Saturated suction temperature (°F)
SDT = Saturated discharge temperature (°F)

Compressor manufacturers provide three sets of coefficients for each compressor, where each set is to determine either cooling capacity, mass flow rate, or compressor input power.

The compressor cooling capacity and mass flow rate given by the above equations are determined at particular rating conditions. One such condition commonly seen is a return gas temperature of 65°F and 0°F of liquid subcooling. Corrections are made to account for the values of superheat and refrigerant liquid temperature. The superheat correction takes into account the density and enthalpy change, while change in liquid temperature affects the enthalpy of the refrigerant entering the evaporator. The correction applied to the compressor capacity or mass flow rate is

$$\textit{Capacity}\ \text{correction} = \frac{v_r}{v} * \frac{(h_{out} - h_{in})}{(h_{rout} - h_{rin})}$$

where

v = the specific volume of the refrigerant (ft^3/lb)
h_{in} = the enthalpy of the refrigerant entering the evaporator
h_{out} = the enthalpy of the refrigerant leaving the evaporator
the subscript, r, designates that the property is at the rating conditions

The capacity value and refrigeration load are then used to find the number of compressors operating by taking the ratio of refrigeration load to capacity. Typically 3 or 4 compressors are needed to meet the load at design conditions. At other conditions less than this number is required. Fractional values represent compressor on/off cycling. The analysis does not use specific compressor models, but instead uses a single generic size for each type of compressor. The generic size is based upon the most commonly compressor, which is a 7.5 HP unit for the reciprocating compressor and a 6 HP unit for a scroll compressor.

Compressor energy consumption for the temperature bin is found by first determining the power needed by the compressor at the state point and load conditions found. The compressor power is multiplied by the number of compressors operating and the number of hours associated with the ambient temperature.

The fan power for remote condensers or fluid coolers is based upon the type of condenser or cooler being examined. Table 5-2 gives the values of fan power required as a function of the design refrigeration load. Air-cooled condensers for low temperature refrigeration are sized for a smaller TD and require more fan power than condensers employed with medium temperature refrigeration. Fan requirements are less for evaporative heat rejection than is needed for dry rejection, because less air flow is needed. The power value listed in the table for the evaporative units includes the sump pump that is employed to spray water over the heat exchanger coil. Installed fan power is found by dividing the design refrigeration load by the appropriate value found in the table.

The condenser fans operate continuously as long as the resulting condensing temperature is greater than the specified minimum value. Fan cycling is employed with both the condensers and fluid coolers to regulate the condensing temperature when full operation of the fans reduces condensing temperature below the minimum value. Fan energy is estimated by multiplying the installed fan power by a fan factor, that represents the amount of fan on-time needed to maintain the condensing temperature at the minimum. For air-cooled condensers and dry heat rejection, the analysis sets the fan factor at 1.0 when the sum of the ambient dry-bulb temperature and the condenser TD are greater than the minimum condensing temperature. The fan factor is set at 0.25 when the ambient dry-bulb temperature is less than 30°F. For ambient temperatures greater than 30°F where continuous fan operation is not needed, the fan factor is calculated from

$$\text{Fan Factor} = \left(1 - 0.75 * \frac{(T_{con} - TD - T_{amb})}{(T_{con} - TD - 30)}\right)$$

where

T_{con} = the minimum condensing temperature (°F)
TD = the temperature difference (°F)
T_{amb} = the ambient dry-bulb temperature (°F)

For evaporative rejection, the above relation is also employed, but the ambient wet-bulb is used instead of the dry-bulb temperature. The fan energy for the temperature bin is determined from the product of the installed fan power, the fan factor, and the number of hours in the bin.

The systems employing fluid loops for heat rejection also require a circulation pump to circulate the water/glycol between the water-cooled condensers and the fluid cooler. For the analysis, the power requirement for this pump is 6 kW. Pump operation is continuous, so that the annual energy consumption for the pump is 52,560 kWh. This amount is added to the fan energy to determine the total energy consumption for heat rejection for the fluid loop heat rejection.

Defrost must also be accounted for in the energy calculation. Refrigeration systems considered here do not employ electric defrost. Hot gas and off-cycle defrost are used for low and medium temperature systems, respectively. Added compressor energy consumption is seen with either of these defrost methods, due to added cooling that must be provided to return the cases to operating temperature after defrost. From previous field tests conducted by Foster-Miller (5-4), the added energy seen is approximately 2 and 3 percent of compressor energy consumption for medium and low temperature refrigeration, respectively. These factors are applied to each temperature bin result.

The energy for mechanical subcooling of the low temperature refrigeration is addressed by the medium temperature system with the highest SST. As mentioned previously, the subcooling load is calculated for each temperature bin and is added to the refrigeration load of the appropriate medium temperature system. The compressor run time for the medium temperature system is calculated on the basis of this combined load.

The bin calculation is repeated until energy values are set for each temperature bin for the refrigeration system configuration specified. Once the bin loop is completed, the model then obtains the next system description and the bin loop is repeated for this next system. The procedure continues until all systems are analyzed.

5.1 Analysis Variations for Distributed and Secondary Loop Refrigeration

5.2.1 Modeling of the Distributed Refrigeration System

The modeling and analysis of the performance of the distributed refrigeration system follows the same procedure described above. The compressor cabinets contain multiple scroll compressors that are piped in parallel. These compressors are on/off cycled in the same fashion as the multiplex compressors to provide control of suction pressure. The same procedures to determine the refrigeration load and state points are followed. The energy consumption of the compressors is found by first comparing the refrigeration load to the available capacity to determine the number of compressors needed. The power requirement of each compressor is then determined from the state points and performance data. The energy is the product of the number of compressors, the power, and the number of hours in the temperature bin. The exceptions to the analysis procedure are for liquid refrigerant subcooling and heat rejection, using the glycol/water fluid loop.

The scroll compressors are capable of mid-scroll injection of refrigerant vapor, which can be used to subcool liquid refrigerant. The subcooling is done using a heat exchanger in the liquid line that is mounted in the compressor cabinet. A portion of the liquid is taken from the liquid line and is expanded into the exchanger to provide liquid subcooling. The vapor generated at the heat exchanger is piped to the scroll compressor at their injection port. The performance obtained by mid-scroll injection subcooling is determined from manufacturer's performance data for the scroll compressor when subcooling is applied. The resulting subcooled liquid temperature is taken at 50°F.

The heat rejection system for the distributed refrigeration system consists of water-cooled condensers mounted in each of the compressor cabinets. A glycol/water loop is pumped between the condensers and a fluid cooler. Either an evaporative or dry fluid cooler can be used in the analysis. The condensing temperature of the compressor cabinet is set at the inlet fluid temperature plus a temperature difference of 10°F. The water temperature is set by the type of fluid cooler employed. For evaporative units the water temperature is the ambient wet-bulb temperature plus 15°F. For the dry cooler, the water temperature is the ambient dry-bulb temperature plus 15°F. A minimum water temperature exists that produces the lowest allowable condensing temperature. The analysis does not allow water temperature to drop below this minimum. Fan and pump energy are determined using the same approach described above for heat rejection.

5.2.2 Modeling of Secondary Loop Refrigeration

The major difference in the analysis of the secondary loop refrigeration system is the operation of the secondary loop. The loop consists of brine that is pumped between a central chiller and the display cases. Several secondary loops are employed in a supermarket, depending upon the refrigeration loads required at each loop temperature. For the supermarket modeling, four loops were considered operating at brine temperatures of -20 for low temperature, and 10, 20 and 30°F for medium temperature refrigeration, respectively. The analysis is, therefore conducted separately for each of these loops at each ambient temperature. Energy results are combined at the completion of the analysis to determine total energy consumption.

The system configuration specifies the design refrigeration load for each secondary loop. The analysis first considers the variation on the refrigeration load seen at each ambient temperature, using the method described previously. The model assumes that the refrigeration load at the display cases is handled by a constant temperature change of the fluid, while the flow through the cases is varied as the refrigeration load varies. This flow arrangement is an attempt to simulate the operation of a temperature control valve that maintains constant fluid outlet temperature from the display case heat exchanger. Since all loads on the loop behave in this fashion, the estimated total fluid flow can be found from

$$\dot{M}_{brine} = \frac{\Delta T_{brine}}{C_{brine}}$$

where

$\dot{M}_{brine}$ = the mass flow rate of brine (lb/hr)
ΔT_{brine} = the temperature change in the brine seen at the cases
C_{brine} = the specific heat of the brine

The secondary fluid loop will experience some heat gain while flowing between the cases and the central chiller. The most significant of these gains is the addition of energy do to operation of the secondary fluid loop pump. The pump power is based on the maximum fluid flow needed to meet the design refrigeration load, which is found with the above relation. The required pump head is set at 75 ft of water column (WC) for low temperature refrigeration loop and 50 ft (WC) for the medium temperature loop. While the flow to the cases varies as the load changes, the total fluid flow through the pump remains constant, since a bypass is used to regulate the operation of the pump in the loop. The power input to the pump is calculated as the ideal power for the maximum fluid flow and head divided by a pump efficiency of 55 percent. The power input to the pump is converted into heat in the fluid, which must be removed by the chiller system. The rise in temperature is calculated from

$$\Delta T_{pump} = \frac{\text{Pump Power}}{\dot{M}_{brine}\, C_{brine}}$$

where

Pump Power = the power input to the secondary fluid pump

Some line heat gain is also expected and was set at 0.25°F for the supply and return lines of the loop.

Once the total temperature rise of the secondary fluid loop is determined, the load on the chiller evaporator can be found by

$$Q_{evap} = \dot{M}_{brine}\, C_{brine} \Delta T_{brine}$$

where

ΔT_{brine} = the total temperature gain seen in the fluid

Mechanical subcooling is used in the secondary loop system for the low temperature chiller system in the same fashion as is seen in multiplex systems. A portion of the medium temperature system provides the subcooling. The refrigeration load associated with the subcooling must be added to the total load of the medium temperature system. The subcooling load is calculated based upon the refrigeration load of the low temperature system, which sets the flow rate of refrigerant needed. The subcooling reduces the temperature of the refrigerant from the temperature leaving the condenser to 40°F. The subcooled liquid temperature is factored into the available capacity of the low temperature system in meeting its refrigeration load.

The state points of the chiller system are then determined. The evaporator temperature of the chiller heat exchanger is set at 7°F below the outlet temperature of the fluid loop (either –20 or 20°F). The outlet temperature of the refrigerant is set at 8°F higher than the evaporator temperature. The temperature rise is due primarily to the control action of the thermostatic expansion valve of the chiller heat exchanger, which regulates the outlet temperature of the refrigerant at a superheated condition. Pressure drop of the refrigerant vapor to the suction of the compressors is negligible due to the close-coupling of the compressors and the heat exchanger. The SDT of the compressor system is determined from the condensing temperature, which is calculated depending upon the type of heat rejection system analyzed. The secondary loop refrigeration system can be modeled with air-cooled, water-cooled, or evaporative condensing. The method of determining the condensing temperature for each heat rejection type is the same as described previously.

The central chiller uses multiple parallel compressors to address the chiller heat exchanger load. The types of compressors now employed in presently installed systems are either screw or reciprocating. Scroll compressors can be analyzed if desired. The procedure for determining the number of compressors operating is the same as that used for the multiplex and distributed systems. Manufacturer's performance data are used to determine compressor cooling capacity and power. The number of compressors operating is found from the ratio of the chiller cooling load to the total available compressor capacity. Compressor energy is the product of the compressor power, number of compressor operating, and the number of hours in the temperature bin being examined.

The energy consumption of the secondary fluid pump is determined using the method outlined previously. While the fluid flow to the display cases varies with changing refrigeration load, the total fluid flow across the pump is constant. Variation in flow is achieved by flow bypassing around the pump. The head addressed by the pump was estimated based upon the temperature of the secondary fluid loop, the secondary fluid employed, the length and diameter of pipe between the cases and chiller, and the pressure drop at the display cases and through the chiller heat exchanger. The secondary fluids examined here consist of propylene glycol/water for the medium temperature loop, and Pekasol 50 for the low temperature loop. Loop piping diameter was sized to maintain a velocity of 4 to 6 ft/sec, while the typical length of piping was estimated at 250 ft. Values of the pressure drop for the display cases range from 5 to 7 ft (WC), while the pressure drop of the chiller heat exchanger was set at 20 ft (WC). The head requirements calculated for the secondary fluid loop pump were 50 and 75 ft (WC) for the medium and low temperature systems, respectively. The pump and motor efficiencies were taken at 55 and 85 percent, respectively.

The energy requirement for heat rejection is dependent upon the type of heat rejection employed. The energy is calculated using the procedures described previously.

5.3 Heat Reclaim for Multiplex Refrigeration

Heat reclaim refers to the use of the reject heat from the refrigeration system for store space heating. Heat reclaim is accomplished by routing the discharge gas from the compressors to a coil located in one or more of the HVAC air handlers. In the coils heat is removed from the refrigerant and is transferred to the store circulation air to provide heat.

The amount of heat that can be recovered in this fashion is determined by the amount of condensing allowed to take place in the heat reclaim coils. Previous attempts to recover a large portion of the heat by complete condensing during heat reclaim resulted in the need for a very large refrigerant charge, since refrigerant was needed to fill both the condenser and the piping between the heat reclaim coils and the condenser. The so-called "winter" refrigerant charge was often larger than could be held in the system during non-heating periods and venting of refrigerant was necessary when the heating season had concluded.

To avoid this excessive amount of refrigerant, the heat reclaim coils are purposely undersized to limit the amount of heat removal. The coils are often sized at a large temperature difference (condensing temperature – inlet air temperature) so that only desuperheating or a very limited amount of condensing occurs.

The amount of heat reclaimable through desuperheating can be estimated from rated compressor data and a cycle calculation as is illustrated in Figure 5-2. The total heat rejection is a combination of the refrigeration capacity and the compressor power. The cycle calculation allows the refrigeration effect and refrigerant mass flow to be found for the rated cooling capacity. The discharge enthalpy, h_{disch}, from the compressor is then determined from

$$h_{disch} = h_{ret} + \frac{\dot{W}_{comp}}{\dot{m}}$$

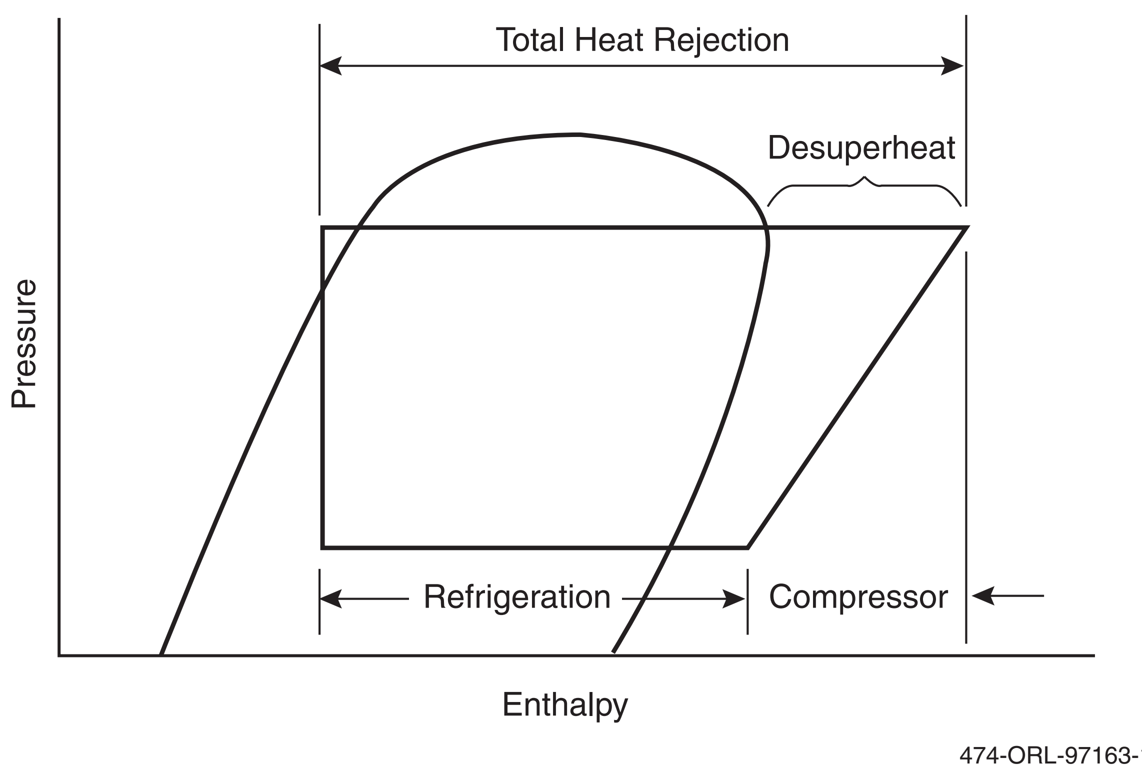

Figure 5-2. Cycle analysis for heat reclaim

where

h_{ret} = the enthalpy of the return gas entering the compressor
$\dot{W}_{comp}$ = the power input to the compressor
$\dot{m}$ = the mass flow rate of refrigerant

The amount of heat recovered, Qrec, by desuperheating is the heat energy contained in the refrigerant greater than that associated with the saturated gas enthalpy.

$$Q_{rec} = \dot{m}\left(h_{disch} - h_g\right)$$

where

h_g = the saturated gas enthalpy of the refrigerant (Btu/lb)

The fraction of heat that is recoverable from desuperheating is

$$\frac{Q_{rec}}{THR}$$

where

THR = total heat rejection
= Refrigeration Capacity + Power

The recovery fractions were evaluated for medium and low temperature refrigeration as a function of evaporator and condenser temperatures, and the return gas temperature was set at 65°F. The refrigeration capacity and compressor work were found from representative compressor performance curves. The following relations were found for the heat recovery fraction as a function of condensing temperature.

For medium temperature refrigeration, the recoverable fraction =

0.23 + (2.0e-05*SST – 0.001)*SST + (2.27e-05*SDT - 0.003)*SDT

For low temperature refrigeration, the recoverable fraction =

0.29 + (0.0001*SST +0.002)*SST + (4.36e-05*SDT – 0.03)*SDT

where

SST = the saturated suction temperature (°F)
SDT = the saturated discharge temperature (°F)

Based on this calculation, it can be found that the amount of heat recovered is small compared to the total heat of rejection. For low temperature refrigeration the fraction of heat recovered ranges from 0.31 to 0.38 for condensing temperatures of 70 and 90°F, respectively. The fraction for medium temperature refrigeration is smaller at 0.16 to 0.24 for these same condensing temperatures.

The standard method for heat reclaim coil sizing consists of allowing a maximum heat recovery of 50 percent of the total heat rejection at a temperature difference (condensing temperature – inlet air temperature) of 35 to 40°F. This sizing is made at the design conditions so that the amount of heat recovered during operation is actually substantially less primarily because the temperature difference seen at the heat reclaim coil ranges from 5 to 30°F, depending upon the value of minimum condensing temperature applied to the refrigeration. The highest value of minimum condensing temperature normally seen is on the order of 95°F. At this condition, partial condensing of the refrigerant will occur to a quality no less than 90 percent. For analysis of heat reclaim performance, the amount of heat recovered can be estimated as being proportional to the temperature difference seen at the heat reclaim coil.

Fan power requirements associated with the heat reclaim coils are also significant in terms of total annual energy consumed. Air must flow through coils at all times, therefore, fan power is consumed regardless whether space heating is needed or not. A typical pressure drop for the heat reclaim coil is on the order of 0.2 in. WC. The flow rate of air passing through the coil must be capable of maintaining a coil face velocity of 500 ft/min. For the application considered here, the required flow is 31,250 cfm. Using a fan efficiency of 0.2, yields a fan power of 3.4 kW. Annual energy consumption for airflow through the heat reclaim coil is 29,784 kWh.

Energy analysis of heat reclaim consists of first determining the annual energy consumption of the refrigeration system at the minimum condensing temperature chosen for heat reclaim operation. At each ambient temperature the amount of heat recovered is calculated based on the expected TD of the heat reclaim coil. For this calculation, the amount of heat recovered is expected to be proportional to the TD. The recovered heat is then compared to the expected building heating load at this same ambient temperature. Only a fraction of the recovered heat will be utilized if the space heat load is less than the amount of heat reclaim. It is also possible that the heating load will exceed the amount of heat recovered, which requires the use of auxiliary heating to satisfy the heating load. A comparison of the combined energy consumption for both space heating and refrigeration can be made to determine the benefit derived from heat reclaim.

5.4 Supermarket HVAC Analysis

5.4.1 HVAC Model Description

The model used for the analysis of supermarket HVAC employs the same ambient temperature bin method as the refrigeration model to determine annual energy use. Figure 5-3 shows a diagram of the model and major steps followed.

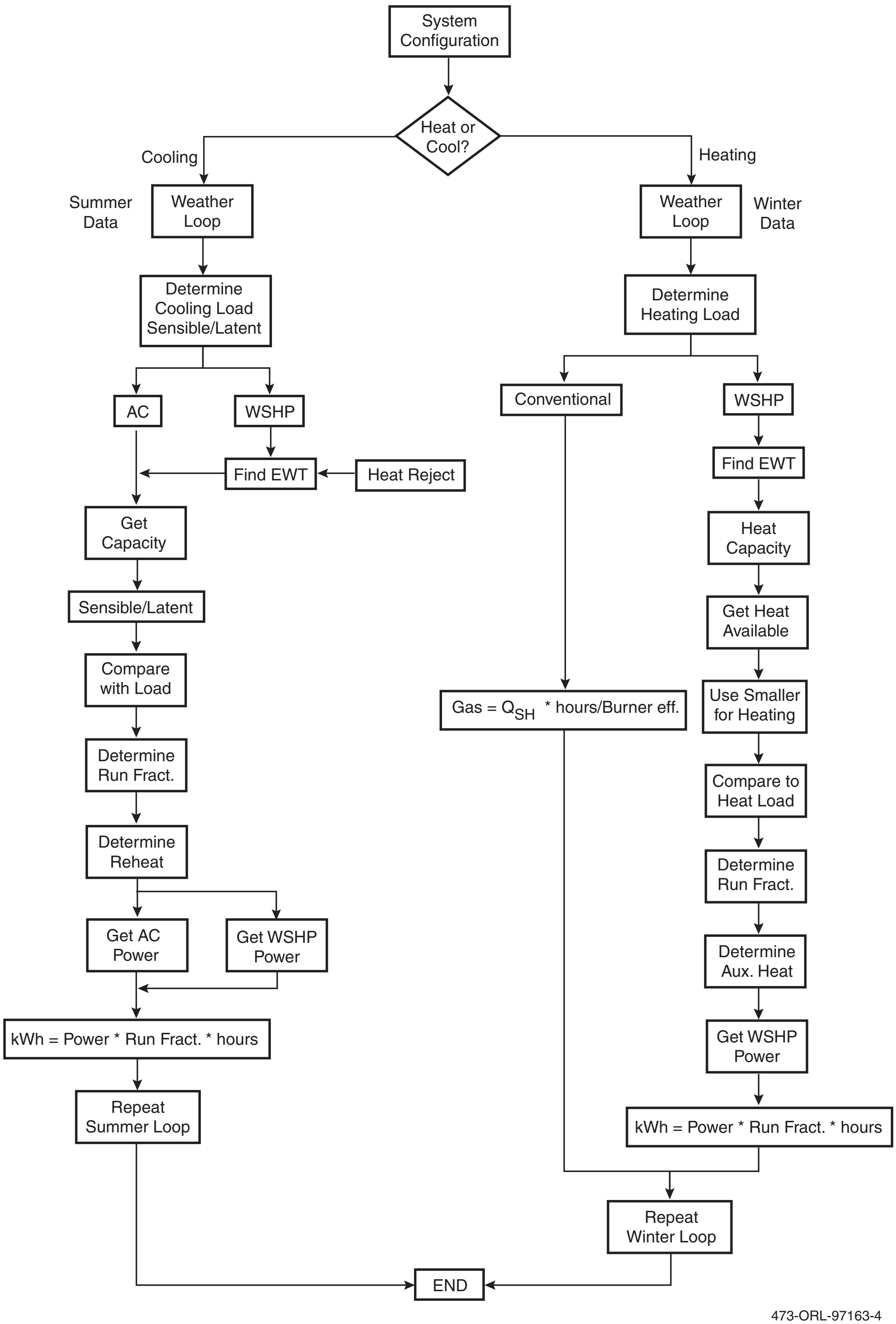

Figure 5-3. HVAC model flow chart

The HVAC model uses a configuration file to initiate the analysis. The configuration file provides the following information:

- Type of HVAC system to be analyzed – conventional or water-source heat pump.
- Space heating or cooling analysis.
- Temperature set point for the store.
- Humidity set point for the store (specified as %RH).
- Installed capacity for space cooling (tons).
- Ventilation flow rate (specified as percent of total store air flow).

The installed space cooling capacity relates to the evaluation of either standard air conditioning or water-source heat pump. The cooling capacity also determines the heating capacity of the water-source heat pump.

The model then determines if space heating or cooling analysis is desired. The type of weather data employed for the analysis is also set by this parameter. For space heating, weather data occurring in the six months of November through April (winter data) are considered. The space cooling analysis examines weather data for May through October (summer data).

The next step in the space cooling analysis is to determine the space cooling load of the store, including both the sensible and latent components. The procedure used is presented in detail in the description of the modeled supermarket to find the design cooling load. Several exceptions to this procedure are used when determining the cooling load during non-design conditions. The sensible and latent loads due to people are reduced to reflect an occupancy of 70, rather than 200 (design value), people. Solar loads through roof and glass store fronts are also reduced to account for lower solar radiation levels and nighttime operation.

The sensible and latent cooling capacities of the installed air conditioning system for the ambient conditions of the bin are now determined. For the conventional air conditioner, capacity is determined based upon the ambient air temperature and the air dry-bulb temperature entering the cooling coil (T_{edb}). T_{edb} is found by considering the mix temperature of the return and ventilation air flows. For the water-source heat pump, total cooling capacity is related to the water temperature entering the heat pump condenser (T_{we}), and the T_{edb} at the cooling coil. T_{we} is determined by the heat rejection employed, which is normally an evaporative fluid cooler. The typical water temperature approach to the ambient wet-bulb is 15°F. The latent component of the cooling capacity is found based upon the entering wet-bulb temperature (T_{ewb}) to the cooling coil. The sensible component is taken as the difference between the total and latent capacities.

The run fraction for the space cooling system is found by comparing the latent and sensible parts of the cooling load to the corresponding components of the cooling capacity. The load-to-capacity ratios for sensible and latent are compared and the larger of the two is considered the run fraction of the cooling system.

In the instance when the run fraction is set by the latent load, excess sensible cooling of the store will occur and reheat is applied to maintain the set point temperature. The amount of reheat needed is determined from the difference between the amount of sensible cooling supplied and the sensible cooling load. The amount of cooling applied is the product of the sensible cooling capacity and the system run fraction. The analysis assumes that the reheat is supplied by a gas duct heater. The gas consumption is determined based on a burner efficiency of 80 percent.

The power and energy needed for space cooling is found from the energy efficiency ratio (EER) of the cooling system. Both the conventional air conditioner and the water-source heat pump have EERs at standard rating conditions. The rated EER can be modified for non-rated conditions. For the conventional air conditioner, the EER is a function of the ambient air dry-bulb temperature. The EER of the water-source heat pump is determined from a relation with the entering water temperature. The power requirement for the temperature bin is the product of the EER and the total cooling capacity. The energy used is the product of the power, the system run fraction, and the number of hours in the bin.

For space heating analysis, winter weather data are used to determine the space heating load for each temperature bin. The procedure is the same as that used for finding the design space heating load, but using the bin's dry-bulb temperature, and allowing for the average solar load and building occupancy.

For conventional system, space heating is provided by a gas-fired duct heater. The gas consumption is the product of the space heating load, the number of hours in the bin, and the burner efficiency, which is set at 80 percent.

For analysis of space heating with the water-source heat pump, the initial step is to determine the amount of heat that can be removed from the water loop. The heat absorbing capacity can be estimated from the rated cooling capacity and is approximately 97 percent of the rated cooling at an entering water temperature of 85°F. The analysis uses the installed cooling tonnage to determine the rated cooling by applying a correction for the design entering water temperature. The absorbed heat capacity, Q_{abs}, at rating is then corrected for the actual entering water temperature for the temperature bin. This corrected value is checked against the amount of rejected heat produced by the refrigeration system, which is transferred to the water loop. If the absorbed heat capacity is larger than the refrigeration reject heat, the absorbed capacity is reduced to match the reject heat and the entering water temperature is recalculated to the value corresponding this heat absorption. The total heat capacity, Q_{tot}, provided by the heat pump for space heating is

$$Q_{tot} = Q_{abs} * \left(1 + \frac{1}{(COP_H - 1)}\right)$$

where

Q_{abs} = absorbed heat capacity
COP_H = heating coefficient of performance

A rated heat pump COP_H is specified for an entering water temperature of 85°F. A correction factor is applied for the entering water temperature calculated for the temperature bin.

The total heating capacity of the heat pump is compared to the space heating load. If the capacity is greater than or equal to the heating load, a run fraction for the heat pump is found from the ratio of the space heat load to the heating capacity. If the space heating load is greater than the heating capacity, the run fraction of the heat pump is set at 1 and the amount of auxiliary heat that must be supplied is determined. The auxiliary heating is equal to the difference between the space heating load and the total heating capacity.

The power used by the heat pump is found from the relation

$$\text{Heat Pump Power} = \frac{Q_{abs}}{(COP_H - 1)}$$

The energy consumed by the water-source heat pump is the product of the power, the heat pump run fraction, and the number of hours in the temperature bin.

The auxiliary heat can be supplied by either gas or electric duct heaters. For the gas heaters, a burner efficiency of 0.8 is used to determine the amount of gas consumed.

The calculations are repeated for all temperature bins contained in the weather data file.

5.4.2 Analysis of Conventional Air Conditioning Systems

Modeling and performance estimates of air conditioning systems are required to evaluate the energy consumption of supermarket HVAC systems. Data from air conditioner manufacturers are available to allow such predictions. The information employed here was found in the Carrier Product Catalog (5-5) for standard rooftop packages, which contain all components needed for store HVAC. The rooftop package will have a standard vapor compression air conditioning system with compressor, condenser and evaporator. The rooftop package also has a blower and air handler to pass store air through the air conditioner evaporator. The air handler provides a duct and damper to control the addition of ventilation air from the outside, which is mixed with the return air from the store at the suction side of the blower. The air is then pumped by the blower through the evaporator coil and discharged to the store. The rooftop unit also provides store heating by adding a gas-fired heater to the supply-side ducting.

For the supermarket air conditioning system, the ability to address the sensible and latent components of the cooling load must be considered. The load split of the air conditioner can be characterized through the concept of the bypass factor (BF) of the evaporator coil. The bypass factor represents the fraction of air that passes through the coil that is unaffected by the coil's cooling. The remainder of the air is cooled to the surface temperature of the coil, which is referred to as the apparatus dew point (ADP) temperature. The BF of a coil is controlled by the amount of air passing through the coil with the BF increasing as the air flow rate is increased.

Lower air flow produces lower values of BF, and larger latent cooling capacity. Higher values of BF increase the total cooling capacity of the air conditioner. The value of BF typically ranges between 0.03 to 0.13.

The split between sensible and latent cooling is also influenced by the entering wet-bulb (T_{ewb}) and dry-bulb (T_{edb}) temperatures. Rating data shows values of total and sensible cooling capacities at three values of T_{ewb} of 62, 67, and 72°F, and at a T_{edb} of 80°F.

Ambient air temperature also has some effect on the cooling capacity of the air conditioner and a strong effect on compressor power, due to changes in condensing temperature.

The rated data for a typical air conditioner were used in a multi-variable regression analysis to determine the values of total and sensible cooling capacities, and compressor power as functions of BF, T_{ewb}, and ambient temperature (T_{amb}). The following relations were determined:

$$\text{Total Cooling Capacity, } Cap_{tot} \text{ (kBtuh)} = 82.897 + 971.83*BF + 5.6357*T_{ewb} - 1.6578*T_{amb}$$

$$\text{Sensible Cooling Capacity, } Cap_{sens} \text{ (kBtuh)} = 974.72 + 1376.8*BF - 8.8357*T_{ewb} - 1.993*T_{amb}$$

$$\text{Compressor Power (kW)} = -12.711 + 50.694*BF + 0.2931*T_{ewb} + 0.238*T_{amb}$$

The sensible cooling capacity is also affected by T_{edb}. The catalog gives the following relation to correct the sensible cooling capacity for T_{edb} values other than 80°F.

$$\text{Sensible Cooling Correction Factor} = 1.10*(1 - BF)*(T_{edb} - 80)$$

The sensible cooling correction factor is added to the above sensible cooling capacity to determine the sensible cooling capacity at the actual value of T_{edb}.

Once the total and sensible cooling capacities are found, the latent capacity is determined from

$$\text{Latent Cooling Capacity, } Cap_{lat} = Cap_{tot} - Cap_{sens}$$

The above relations can be used to estimate the energy consumption of the air conditioning through the following procedure. For a given ambient condition (dry-bulb and coincident wet-bulb temperatures), the space cooling load for the supermarket can be estimated, along with the sensible and latent load components. Given the store set points for temperature and humidity, and the ventilation rate, the temperature and humidity of the air entering the air conditioner coil can be found. The above performance equations are used to find the sensible and latent cooling capacities of the air conditioner. The required run fraction of the air conditioner is found by

$$\text{Sensible Run Fraction, } RF_{sens} = \frac{Q_{sens}}{Cap_{sens}}$$

$$\text{Latent Run Fraction, } RF_{lat} = \frac{Q_{lat}}{Cap_{lat}}$$

where

Q and Cap represent the load and capacity, respectively
sens and lat refer to the sensible and latent components

The run fraction used for energy consumption estimate is the larger of the two and the compressor power is found from the relation shown above

Air Conditioner Energy = RF * Air Conditioner Power * hours

where

Air Conditioner Power = Compressor Power + Condenser Fan Power (1.7 kW)
Hours = number of hours at the ambient conditions specified

The above procedure also allows the estimate of reheat requirements. Reheat will occur when the latent run fraction is greater than the sensible. In this situation, more sensible cooling will be provided than is needed to satisfy the sensible load. The amount of reheat needed can be estimated from

Reheat required = $Cap_{sens} * RF_{lat} - Q_{sens}$

5.4.3 Water-Source Heat Pump

The water-source heat pump (WSHP) can supply either heating or cooling to the supermarket sales area and works in conjunction with the glycol/water loop used for refrigeration system heat rejection. The fluid loop provides heat rejection when the WSHP is space cooling, or can be the source of heat for the heat pump when store space heating is called for.

Analysis of WSHP requires that the heating and cooling performance of the heat pump be known in relation to ambient conditions inside and outside the store. The heating and cooling capacity of the heat pump are given in manufacturer's data in terms of the entering water temperature (T_{we}). Power requirement for the heat pump is found from the value of the EER, for cooling, and COP_H, for heating. Both of these quantities are given in the literature as functions of T_{we}.

Cooling performance of the WSHP is also characterized by the inlet air temperature and humidity entering the cooling coil, and the bypass factor of the coil. These relations are similar to those presented previously for the conventional air conditioner and are used here to determine the split between latent and sensible cooling for space cooling analysis.

Manufacturer's data (5-5) were used to establish the relation between the entering water temperature and the heating and cooling capacity of the WSHP. These data were correlated using standard linear regression procedures. In order to use these relations for any size heat pump examined, the resulting equations were normalized in terms of the rated heating and cooling capacities of the heat pump. The rated heating capacity of the heat pump occurs at a T_{we} of 70°F, while the rated cooling capacity is at a T_{we} of 85°F.

The relation for space cooling capacity of the WSHP with change in T_{we} is the following

$$\frac{Cap}{Cap_r} = -0.0061T_{we} + 1.5214$$

where

Cap = the total cooling capacity
Cap_r = the total cooling capacity at the rated condition

The EER of the heat pump in cooling mode can be expressed as follows

$$\frac{EER}{EER_r} = -0.0127T_{we} + 2.0936$$

where

EER = the energy efficiency ratio of the heat pump
EER_r = the energy efficiency ratio at the rated condition

The space heating capacity of the heat pump is described in terms of the total heat absorbed (Q_{abs}) from the water loop and the COP. The total amount of heat delivered to the conditioned space, Q_{tot} is the sum of Q_{abs} and the power consumed by the compressor, which is found from

$$Q_{tot} = Q_{abs}\left(1 + \frac{1}{(COP_H - 1)}\right)$$

The relation found for the ratio of $\frac{Q_{abs}}{Q_{absr}}$ and T_{we} is

$$\frac{Q_{abs}}{Q_{absr}} = 0.004T_{we} + 0.6789$$

The heat pump COP_H can be found from

$$\frac{COP_H}{COP_{Hr}} = 0.0022T_{we} + 0.8467$$

Section 5 References

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5-4. Borton, D.N., and D.H. Walker, "Development of a Demand Defrost Controller," New York State Energy Research & Development Authority, Contract No. 800-CON-BCS-86, March, 1992.

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6. DESCRIPTION OF THE MODELED SUPERMARKET

A description of a typical, modern supermarket was constructed for the analysis of various refrigeration and HVAC systems. The conditioned sales area for this modeled market was set at 40,000 ft^2, which is a representative size for the majority of stores now in operation or recently constructed.

6.1 Refrigeration Description

A typical refrigeration load schedule was devised for modeling and comparison of the refrigeration systems examined. The refrigeration schedule describes the connected refrigerated fixtures, made up of display cases and walk-in storage coolers, and gives the desired discharge air and evaporator temperatures, and the refrigeration load at design conditions (75°F, 55% RH). Table 6-1 provides a listing of all refrigerated fixtures in the modeled supermarket.

The refrigeration loads were then assigned to the multiplex, distributed, and secondary loop systems so that a refrigeration system description was formulated for each type of refrigeration system examined. The resulting configurations are shown in Tables 6-2, 6-3, and 6-4 for the multiplex, distributed, and secondary loop systems, respectively. For the multiplex system, the refrigeration loads are assigned to the compressor racks based upon the evaporator temperature of each load and saturated suction temperature (SST) of the racks. The load is assigned to the rack with the SST closest to, and less than the evaporator temperature of the load. Table 6-2 shows that several of the compressor racks employ two SST levels, which signifies that a split suction manifold is used with a common discharge for both suction levels. System performance analysis is done at each SST value.

Table 6-3 shows the refrigeration load configuration for the distributed refrigeration system. The compressor cabinets are matched to the refrigeration loads in the same fashion as is used for the multiplex system. Because the compressor cabinets are located throughout the store, less refrigeration circuits are assigned to each cabinet, and usually one compressor cabinet is used for refrigeration of a particular product, such as meat, produce, or frozen food. For this reason, the match between the SST of the compressor cabinets and the evaporator temperature can be kept closer than is seen in multiplex systems. If more than one SST is needed in a particular locality in the store, a split suction manifold can also be used in a compressor cabinet. Analysis of the distributed system is also done for each SST value seen in the supermarket.

Table 6-4 shows the arrangement of refrigeration loads for the secondary loop system. The secondary loop system operates at four secondary fluid loop temperatures of –20, 10, 20 and

Table 6-1. Description of modeled cases and coolers (low temperature)

System No.	Case or Cooler Description	Length No. of Doors or Floor Size (ft)	Discharge Air Temperature (°F)	Evap Temperature (°F)	Design Load (Btuh)
1	Ice Cream Door Cases	17 doors	-12	-19	26,180
2	Ice Cream Door Cases	17 doors	-12	-19	26,180
3	Ice Cream Door Cases	19 doors	-12	-19	29,260
4	Ice Cream Door Cases	20 doors	-12	-19	30,800
5	Frozen Meat	28	-10	-20	12,100
6	Grocery Freezer	42 x 15	-10	-20	39,900
7	Frozen Fish	12	-12	-19	3,300
9	Frozen Food Door Cases	15 doors	-5	-11	21,375
10	Frozen Food Door Cases	16 doors	-5	-11	29,300
11	Frozen Food Door Cases	15 doors	-5	-11	21,375
12	Frozen Food Door Cases	16 doors	-5	-11	22,800
13	Bakery Freezer	12 x 10	-5	-11	9,800
14	Deli Freezer	8 x 10	-5	-11	7,000
16	Meat Cases	40	24	15	54,800
17A	Meat Cases	36	22	17	15,120
17B	Fish Cases	12	22	17	12,660
18	Meat Cases	30	24	18	12,580
19A	Bakery Cooler	6 x 8	28	18	10,560
19B	Deli Cases	20	32	18	29,900
19C	Deli Cases	8	30	25	3,360
19D	Deli Cases	16	26	20	7,040
19E	Deli Cooler	12 x 11	36	29	9,100
19F	Cooler	6 x 8	25	15	10,560
19G	Cheese Cooler	6 x 14	25	15	18,480
20A	Deli Cooler	6 x 8	25	15	10,560
20B	Deli/Meat Cases	32	32	18	46,880
21	Meat Cooler	40 x 15	30	22	35,800
22	Meat Box	15 x 10	30	22	11,400
24A	Dairy/Deli Cases	36	32	21	54,000
24B	Dairy/Deli Cases	40	32	21	60,000
25A	Dairy	12	32	18	18,000
25B	Beverage	36	32	18	53,820
25C	Beverage	24	32	18	36,000
26A	Produce	24	32	21	35,880
26B	Produce	56	39	21	43,120
26C	Produce	82	39	21	62,440
26D	Floral Cooler	12 x 11	38	32	11,675
26E	Floral Case	13		21	12,350
27A	Bakery Retarder	12 x 8	36	28	7,300
27B	Dairy Cooler	40 x 20	36	28	54,125
28A	Meat Prep	40 x 15	50	35	74,250
28B	Produce Cooler	36 x 14	38	30	27,200

Table 6-2. Multiplex refrigeration system configuration for the modeled supermarket

	SST (°F)	Design Refrigerant Load (Btuh)	Refrigerant
Low Temperature			
Rack A	-23	167,720	R404A
Rack B	-14	111,650	R404A
Total Low Temperature Load		279,370	
Medium Temperature			
Rack C	13	222,100	R-22
	18	66,700	
Rack D	19	375,610	R-22
	26	174,550	
Medium Temperature Load		838,960	
Design Subcooling		53,000	
Subcooling Load varies with change in low temperature refrigeration load			

Table 6-3. Distributed refrigeration system configuration for the modeled supermarket

	SST (°F)	Design Refrigerant Load (Btuh)	Refrigerant
Low Temperature			
Cabinet A	-14	15,400	R-404A
Cabinet B	-23	152,320	
Cabinet C	-14	111,650	
Total Low Temperature Load		279,370	
Medium Temperature			
Cabinet D	19	105,720	R-404A
Cabinet E	14	35,490	
Cabinet F	13	39,600	
Cabinet G	19	262,120	
Cabinet H	19	212,380	
Cabinet I	26	70,525	
Cabinet J	29	113,125	
Total Medium Temperature Load		838,960	

30°F. Four chiller systems are employed, one for each fluid loop. These four loop temperatures are capable of addressing all display case loads, including ice cream and meats (6-4).

The multiplex and secondary loop systems employ mechanical subcooling for the low temperature refrigeration that is supplied by the medium temperature refrigeration. A design subcooling load is shown in the table for both these refrigeration systems. During operation, the amount of subcooling required by the low temperature systems varies with the size of the low temperature load. As part of the analysis, the subcooling load is calculated for each temperature bin and is added to the medium temperature refrigeration load. The subcooling load is not included in the total medium temperature refrigeration shown for the multiplex and secondary loop systems so that the loads can be compared with the medium temperature load of the distributed refrigeration. Subcooling for the low temperature refrigeration in the distributed

Table 6-4. Secondary loop refrigeration system configuration for the modeled supermarket

	Loop Temperature (°F)	Design Refrigerant Load (Btuh)	Refrigerant
Low Temperature			
Total for Low Temperature Loop	-20	279,370	R-507
Medium Temperature			
MT-1	10	75,090	R-507
MT-2	20	580,220	
MT-3	30	183,650	
Total for Medium Temperature Loop		838,960	R-507
Design Subcooling Load		53,000	
Subcooling load varies with change in low temperature refrigeration load			

systems is provided by vapor injection at the low temperature compressors. The capacity and power changes seen at the compressors are included as part of the manufacturer performance data, and is therefore, not specified as a particular load.

Table 6-5 lists the heat rejection methods employed for each system. The choice of heat rejection was made on the basis of most common practice for each system type. The multiplex and secondary loop systems are modeled with air-cooled condensers that are sized at TDs (condenser – dry bulb) of 10 and 15 °F, for low and medium temperature refrigeration, respectively. The distributed refrigeration uses water-cooled condensers located in each of the compressor cabinets. The TD (condenser – inlet water) for the water-cooled condenser is 10°F.

Table 6-5. Heat rejection for modeled refrigeration systems

			Refrigerant System Applied		
Heat Rejection	Temperature Difference (TD)	TD Value (°F)	Mult.	Dist.	Sec. Loop
Air-Cooled Condenser	Condenser – Ambient Dry-bulb	Low temperature - 10 Medium temperature – 15	x		x
Evaporative Condenser	Condenser – Ambient Wet-bulb	12	x		x
Water-Cooled Condenser	Condenser – Entering water	10	x	x	x
Evaporative Fluid Cooler	Leaving water – Ambient Wet-bulb	12	x	x	x
Dry Fluid Cooler	Leaving water – Ambient Dry-bulb	12	x	x	x

The fluid cooler used to reject heat from the water loop is evaporative and operates at an approach temperature (leaving water – wet-bulb) of 12°F.

The use of the evaporative fluid cooler with the distributed refrigeration is currently the most prevalent method for heat rejection with these systems, since rejection to the wet-bulb temperature helps to overcome the added temperature difference associated with the fluid loop. Dry fluid coolers can also be employed, in which case, a representative approach temperature (leaving water – dry-bulb) would be 12°F. The resulting difference between the refrigeration condensing and ambient dry-bulb temperatures will be higher than is seen with air-cooled condensing, on the order of 22°F, which will hurt the performance of the distributed system. Air-cooled condensers can also be used with the distributed system and were modeled here. Condensing temperatures for a distributed system with air-cooled condensers would be similar to these seen with multiplex with air-cooled condensers.

Operation of the multiplex and secondary loop systems with either evaporative or water-cooled condensers is also possible, but is not considered standard practice. For evaporative condensers, best operation occurs at a TD (condenser – wet-bulb) of 12°F. A water-cooled rejection system could employ either an evaporative or a dry fluid cooler. The condenser and ambient temperature differences would be the same as seen for the distributed refrigeration system.

Table 6-6 lists the remaining system parameters specified for the refrigeration system analysis. The significant parameters are:

- Pressure drop between the evaporators and the compressor suction – expressed as a difference between the saturated evaporator and suction temperatures. The largest drop is seen in the multiplex refrigeration because of the long suction pipe runs between the display cases and the compressor racks. The close coupling of the compressor cabinets and the display cases seen with the distributed system helps to reduce this loss to 2°F or less. For the secondary loop refrigeration, the chiller evaporator is mounted on the same skid as the compressors, which reduces this pressure drop to a very small value; a value of 0 was used in this analysis.

Table 6-6. Parameters used for refrigeration system analysis

System	System Pressure Drop (SET-SST) (ºF)	Return Gas Temperature (ºF)	Minimum Condensing Temperature (ºF)	Liquid Refrigerant Temperature (ºF)	
				No Subcool	Subcooled
Multiplex	3	45	70	Tcon - 10	45
Distributed	2	Tevap + 15	60	Tcon - 10	Variable
Secondary Loop	0	Tevap +10	70	Tcon - 10	40

- Return gas temperature – the rise in suction gas temperature due to ambient heating reduces the mass flow capability of the compressors, which results in longer run times to satisfy the refrigeration load. The return gas temperature was set at 45°F for both low and medium temperature for the multiplex system. For the distributed and secondary loop systems, the value of the return gas temperature was set at 15 and 10°F higher than the saturated suction temperature, respectively.

- Minimum condensing temperature – the lowest condenser temperature value the system is allowed to operate. A value of 70°F was used for the multiplex and secondary loop refrigeration systems, because both employed reciprocating compressors in the present analysis and this is the lowest condensing temperature recommended for these compressors. For the distributed system, the minimum condensing temperature was set at 60°F, because of the use of scroll compressors, which can be operated at lower condensing temperatures than reciprocating units. For the low-charge multiplex system, the minimum condensing temperatures were set at 40 and 60°F for low and medium temperature refrigeration, respectively. The condenser control scheme of the low-charge multiplex allows the compressors to operate at these lower temperature values (6-5).

- Liquid refrigerant temperature – for non-subcooled systems, the liquid temperature was set at 10°F less than the condensing temperature. This value is typical in systems where fan cycling is used for head pressure control. For subcooled systems, the liquid temperature varied according to system type. A value of 45°F was used for multiplex refrigeration to account for temperature rise seen between the subcooler heat exchanger and the display case inlet. The temperature of the liquid leaving the subcooler was estimated at 40°F, which is a typical set point value for mechanical subcooling. The subcooled liquid temperature for the distributed system is variable depending upon the vapor injection flow rate seen at the scroll compressors. In general, the subcooling provided by the vapor injection will reduce the liquid temperature leaving the condenser by approximately 15 to 20°F. Minimal temperature rise is anticipated because of the close proximity of the compressor cabinets to the display cases. No temperature rise is expected for the secondary loop refrigeration because the subcooled refrigerant liquid is used in the low temperature chiller.

Table 6-7 lists system parameters unique to the operation of the secondary fluid loops for secondary loop refrigeration. The fluid temperature difference across the display cases was set at

Table 6-7. System parameters used for the analysis of secondary loop refrigeration

Parameter	Value
Fluid Temperature Difference (ºF)	7
Chiller Approach (Fluid leaving – Evaporator) (ºF)	5
Secondary Fluids	
Medium Temperature	Propylene Glycol/Water
Low Temperature	Pekasol 50

a nominal 7°F. The approach temperature difference for the chillers was set at 5°F. The secondary fluids selected for modeling were a mixture of propylene glycol and water for medium temperature and a potassium formate/water mixture that is marketed under the product name Pekasol 50. The secondary loop refrigeration system was modeled as employing reciprocating rather than screw compressors. Energy related performance of the reciprocating compressors is better than that seen with the screw compressors and the results presented here represent the lowest energy consumption expected from the secondary loop refrigeration.

The effect of defrost on refrigeration system performance was also considered in the analysis. For the multiplex and distributed refrigeration systems, hot gas defrost is employed for low temperature refrigeration. Defrost of the medium temperature system is done through off-cycle defrost. The secondary loop system uses warm glycol for both low and medium temperature refrigeration defrosts.

To evaluate the effect of ambient temperature on refrigeration system performance, four locations were chosen and are listed in Table 6-8 according to the average number of degree-days of heating seen at each location. These sites represent a wide range of ambient conditions with the highest number of degree-days occurring in Worcester, MA and the lowest in Los Angeles, CA.

6.2 HVAC Description for the Modeled Supermarket

Space cooling and heating loads were determined for each of the sites selected using standard methods to calculate these loads. A description of the methods used is provided below.

6.2.1 Space Cooling Load

The air conditioning load for a supermarket is unique because of the large amount of refrigerated fixtures installed, which greatly influence both the sensible and latent components of the cooling load. Internal sensible loads such as lights and other equipment normally dominate air conditioning loads for commercial buildings. Air conditioning in most commercial buildings tends to operate over an extended portion of the year, despite ambient temperatures, which are less than the building's temperature set point. For supermarkets, the sensible heating of the internal loads are negated by the large amount of heat removed by the refrigerated fixtures. The sensible portion of the air conditioning load is substantially smaller, and the total air conditioning load of the supermarket has a large latent load component. Latent loads are also important to the operation of the refrigerated display cases, because ambient moisture is captured by the cases and deposited as frost on the evaporator. Frost loading can reduce air flow through the evaporator, reducing the refrigerating capability of the case. The maximum ambient conditions at which the display case is rated are a dry-bulb temperature of 75°F and a relative

Table 6-8. Locations chosen for supermarket refrigeration analysis

Location	Average Heating Degree-Days
Worcester, MA	6,848
Washington, DC	4,550
Memphis, TN	3,227
Los Angeles, CA	1,819

humidity of 55 percent. These values are often used as the control points for the store air conditioning. Because of the large latent load seen in a supermarket, both a thermostat and humidistat are employed to control the air conditioning. In some locations, it is possible to have a latent load large enough to cause the store air conditioning to operate after the sensible cooling demand has been met. In these situations, reheat of the air may be required to maintain a condition in the store that is acceptable to customers and store employees. Reheat can be done using heat reclaim from the refrigeration if the store's HVAC system is equipped with reclaim coils at the air handlers.

The sizing of the air conditioning system is based upon the design air conditioning load, which can be calculated using the procedures outline in ref. (6-1). Additional specific information concerning HVAC loads for supermarkets can be found in ref. (6-2).

The air conditioning load for a supermarket consists of sensible and latent portions. The sensible cooling load is derived from the following elements:

- Conduction through the building walls and roof – The conduction load can be determined through methods outlined in ASHRAE. For analysis, wall construction was characterized as 8 in. concrete block with insulation. The heat transfer coefficient, U, for this wall type is listed as 0.103 Btu/hr-ft^2-°F. The roof construction was considered to be steel sheet with 2 in. of insulation and a U of 0.092 Btu/hr-ft^2-°F. One wall of the supermarket was taken to be essentially all glass with a U value of 1.0 Btu/hr-ft^2-°F.

- Roof solar loading – Solar insolation, particularly on the roof, can add significantly to the cooling load of the building. The procedure used to determine the roof solar load involved the Cooling Load Temperature Difference (CLTD), described in ASHRAE, which determines an equivalent temperature difference to take into account the solar loading. For each location examined, peak and average CLTD values were calculated. The peak value was used to determine the design cooling load, while the average value was used in energy analysis calculations.

- Glass solar transmittance – A sensible load was included to account for solar gain through the glass front of the supermarket. Peak and average values of 36,000 and 18,000 Btuh were used for design and energy calculations, respectively.

- Fan heating – The power associated with the HVAC fans add to the sensible cooling load. The value of the fan load was taken as a constant 61,000 Btuh.

- Ventilation – The ventilation air flow into the supermarket will have both latent and sensible loads. The sensible load is found from the change in air temperature from outside ambient to store. The flow rate of ventilation air was set at 10 percent of the total store circulation.

- Infiltration – The amount of ambient air entering the store through infiltration was set at 5 percent of total store circulation.

- People – People provide a total cooling load of 450 Btuh, of which 250 Btuh is sensible. The peak number of people in the store was taken to be 200, while the average number of people was set at 70.

- Lighting and miscellaneous heat loads – The lighting level in the store was taken as 3.0 Watts/ft^2, which is a typical value for supermarkets. The remainder of the load caused by installed equipment, such as electrical appliances, ovens, etc., was set at 0.8 Watts/ft^2.

The latent portion of the cooling load can be described in terms of the following components:

- Ventilation – Moisture must be removed from the ventilation air to lower the humidity from ambient to store level.

- Infiltration – Infiltration air will also increase store humidity and is addressed in the same fashion as the ventilation air.

- People – The latent portion of the cooling load is 200 Btuh/person.

- Miscellaneous – A constant moisture load of 70 lb/hr was used to account for all remaining latent load addition to the supermarket.

The remaining elements of the cooling load are the cooling credits assigned to the refrigerated display cases. The cooling credits represent the sensible and latent loads removed from the sales area by the operation of the refrigeration. The cooling credits were calculated based upon the refrigeration schedule used for the analysis of the refrigeration compressor systems. The total display case refrigeration load is 768,270 Btuh. The displays fans, lights, and heaters require a cooling load of 141,533 Btuh for the electric input associated with these items, which leaves a net refrigeration load of 626,737 Btuh. The latent portion of the load accounts for approximately 18 percent of the total case load, which amounts to 138,289 Btub. The remaining sensible load is 488,448 Btuh. A correction was applied to the latent load credit since the air conditioning does not have to freeze the moisture associated with this load. The resulting latent credit is 121,966 Btuh.

The design ambient condition for each site examined consisted of the 1 percent values for the dry-bulb and wet-bulb temperatures. Table 6-9 gives these temperature values for each of the sites considered.

Tables 6-10 and 6-11 give the sensible load elements calculated for each site. Roof and glass loads include both solar and conduction loads. The glass load also contains a solar transmittance component.

Table 6-12 describes the latent load elements estimated for the design air conditioning load.

Table 6-9. Design ambient conditions for the supermarket air conditioning load calculation

Location	Design Point		
	Dry-bulb (ºF)	Wet-bulb (ºF)	Specific Humidity (lb/lb)
Worcester, MA	88	77	0.0176
Washington, DC	93	78	0.0174
Memphis, TN	98	80	0.018
Los Angeles, CA	93	72	0.012

Table 6-10. Sensible cooling load analysis for supermarket air conditioning

Location	Conduction and Solar			
	Roof	Walls	Glass	Fans
Worcester, MA	267,737	20,414	102,066	61,000
Washington, DC	271,169	28,266	127,476	61,000
Memphis, TN	271,169	36,118	152,886	61,000
Los Angeles, CA	271,169	28,266	127,476	61,000

Table 6-11. Sensible load analysis for supermarket air conditioning

Location	Ventilation	Infiltration	Case Credit	Lighting and Miscellaneous	People
Worcester, MA	56,394	28,197	(488,449)	518,776	50,000
Washington, DC	78,084	39,042	(488,449)	518,776	50,000
Memphis, TN	99,774	49,887	(488,449)	518,776	50,000
Los Angeles, CA	78,084	39,042	(488,449)	518,776	50,000

Table 6-12. Analysis of latent loads for supermarket air conditioning

Location	People	Miscellaneous	Ventilation	Infiltration	Case Credit
Worcester, MA	40,000	75,320	147,197	73,598	(121,966)
Washington, DC	40,000	75,320	143,323	71,662	(121,966)
Memphis, TN	40,000	75,320	154,944	77,472	(121,966)
Los Angeles, CA	40,000	75,320	38,736	19,368	(121,966)

Table 6-13 gives the total estimated cooling loads for design of the air conditioning system. For the final sizing, an extra 20 percent cooling capacity was added to ensure that adequate cooling is available at each of the sites. The use of a safety factor like this is standard procedure.

6.2.2 Space Heating Load

The space heating load for a supermarket is made up of two types of load elements, which are fixed, essentially constant, and ambient-dependent, change as ambient dry-bulb temperature changes.

Table 6-13. Air conditioning unit sizing for supermarkets

	Total Cooling Loads (Btuh)					
Location	Sensible	Latent	Total	Sense/Total	Tons	Added 20% Capacity
Worcester, MA	616,135	214,149	830,284	0.74	69.2	83.0
Washington, DC	685,364	208,339	893,703	0.77	74.5	89.4
Memphis, TN	751,161	225,770	976,931	0.77	81.4	97.7
Los Angeles, CA	685,364	51,458	736,822	0.93	61.4	73.7

Space Heating Load = Fixed Load Elements + Ambient-dependent Load Elements

The fixed load elements consist of the following:

- Lighting and miscellaneous heat loads.
- Fan heating.
- People.

The same values for these quantities applied to space cooling were used for space heating analysis with the exception of the people loading. Only the sensible load portion of 250 Btuh/person is considered for space heating. All of these load elements generate heat in the store and reduce the amount of heating to the supplied by the HVAC.

The ambient dependent portion of the space heating load consists of:

- Ventilation –The heating load is found from the change in air temperature from outside ambient to store. The flow rate of ventilation air was set at 10 percent of the total store circulation.

- Infiltration – The amount of ambient air entering the store through infiltration was set at 5 percent of total store circulation.

- Wall and roof conduction – The same approach and heat transfer coefficients are used to determine these portions of the space heat load.

The impact of the refrigerated display cases on space heating is to increase the amount of heat that must be provided by the HVAC. The value of this increase was the same as the sensible cooling credit of 488,448 Btuh used for determining the space cooling load.

Table 6-14 lists the 99 percent design dry-bulb temperature values for each of the sites examined (6-3). A design space heating load was found for each site based on these design temperatures. The values of the design loads are also listed in Table 6-14.

Table 6-14. Space heating design ambient temperatures and loads for modeled sites

Location	Winter Design Dry-Bulb Temperature (ºF)	Design Space Heating Load (Btuh)
Worcester, MA	0	1,181,055
Washington, DC	10	1,000,435
Memphis, TN	13	946,249
Los Angeles, CA	37	512,761

6.3 Utility Rates

Table 6-15 provides the utility rates for electric, gas, and water used in the analysis. These rates were obtained from the respective local utility of each location considered.

Section 6 References

6.1. *1989 ASHRAE Handbook, Fundamentals,* Chapter 26, "Air-Conditioning Cooling Load," American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc., Atlanta, GA.

6.2. *1987 ASHRAE Handbook, HVAC Systems and Applications,* Chapter 18, "Retail Facilities," American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc., Atlanta, GA.

6.3. *1989 ASHRAE Handbook, Fundamentals,* Chapter 24, "Weather Data," American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc., Atlanta, GA.

6.4. Personal communications with Mr. Yakov Arshansky, Hill-Phoenix Refrigeration Corporation.

6.5. Enviroguard & Enviroguard II, System Technical Brochure, Tyler Refrigeration Corporation, Niles, MI 49120.

Table 6-15. Utility rates used for the supermarket refrigeration and HVAC analysis

	Local Utility Rates		
Location	Electric ($/kWh)	Gas ($/MMBTU)	Water ($/Mgal)
Worcester, MA	0.092	7.34	2.59
Washington, DC	0.071	7.38	1.37
Memphis, TN	0.064	6.11	2.04
Los Angeles, CA	0.092	6.43	2.27

7. ANALYSIS RESULTS

7.1 Refrigeration Analysis Results

Figure 7-1 and Table 7-1 show the estimated annual electric energy consumption for the refrigeration systems analyzed at each of the locations chosen. The multiplex system with air-cooled condensing is considered the baseline, since it is the most commonly installed configuration now used in supermarkets. The results for the low refrigerant charge systems are shown for different methods of heat rejection. The heat rejection method chosen for each system was based upon lowest energy consumption obtained for that particular system. The analysis results show that the lowest energy consumptions were achieved by the distributed systems for operation in Worcester, MA and Washington, DC. Lowest energy consumption was seen for the secondary loop system with evaporative condensing in Memphis, TN and Los Angeles, CA. The low-charge multiplex air-cooled had lower energy consumption than the baseline multiplex system, but also had an energy consumption greater than distributed or secondary loop for all locations. The use of evaporative condensing with the low-charge multiplex lowered the energy consumption, particularly for Memphis, TN and Los Angeles, CA, where this configuration

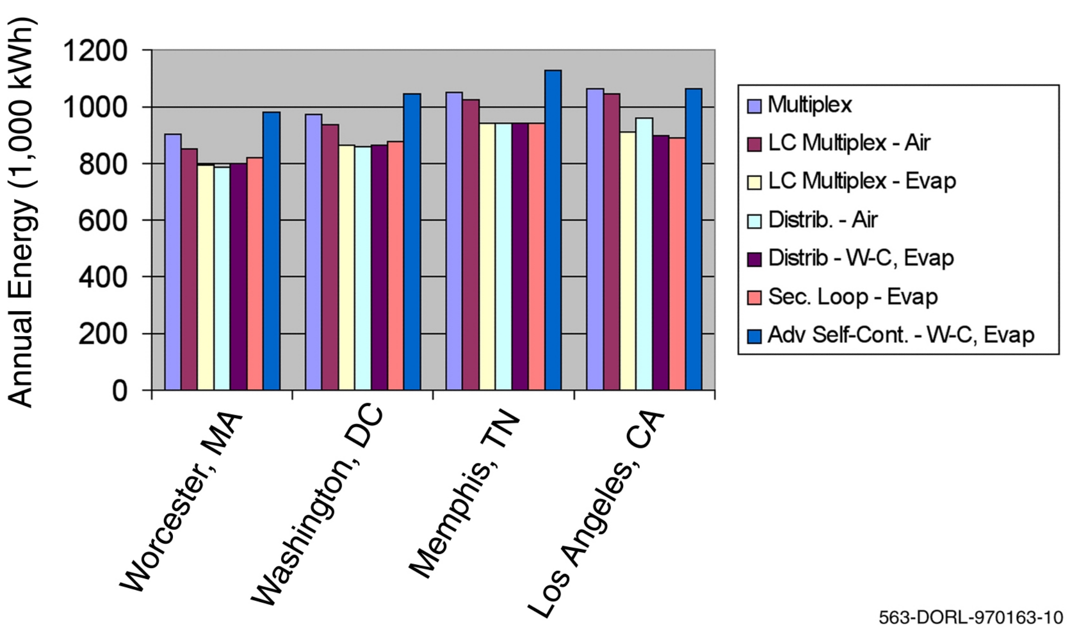

Figure 7-1. Annual energy consumption for low-charge supermarket refrigeration systems

Table 7-1. Annual energy consumption (kWh) for low-charge refrigeration systems for selected locations

Location	Multiplex Air-Cooled	Low-Charge Multiplex Air-Cooled	Low-Charge Mutiplex Evaporative Condenser	Distributed Air-Cooled	Distributed Water-Cooled, Evaporative	Secondary Loop Evaporative Condenser	Advanced Self-Contained Water-Cooled, Evaporative
Worcester, MA	904,500	850,000	791,600	785,700	802,200	821,600	983,700
Washington, DC	976,800	935,200	863,600	860,500	866,100	875,200	1,048,300
Memphis, TN	1,050,200	1,027,100	941,500	942,800	943,200	940,400	1,126,800
Los Angeles, CA	1,067,200	1,042,600	911,300	958,432	894,400	892,400	1,066,800

produced the lowest energy consumption. The advanced self-contained system produced the highest energy consumption for all locations.

Water consumption for evaporative heat rejection was also estimated. The annual consumption of water at each site is given in Table 7-2. Similar amounts of water consumption were seen for each of the refrigeration systems when either evaporative condensing or closed-loop water cooling with evaporative rejection was employed.

Table 7-2. Annual water consumption for heat rejection distributed refrigeration with an evaporative fluid cooler

Location	Annual Water Consumption (gal)
Worcester, MA	1,128,067
Washington, DC	1,179,911
Memphis, TN	1,241,584
Los Angeles, CA	1,225,884

Table 7-3 gives the annual energy savings achieved by each of the low-charge refrigeration systems when compared to the multiplex refrigeration system for operation in Washington, DC. The largest energy savings were achieved by the distributed system operating with air-cooled condensing at 11.9 percent. Similar savings were also seen with the distributed system employing water-cooled condensing and evaporative heat rejection, and with the low-charge

Table 7-3. Energy savings achieved by low-charge refrigeration systems

System	Heat Rejection	Annual Energy (kWh)	Energy Savings versus Multiplex (kWh)	% Savings versus Multiplex
Multiplex	Air-Cooled Condenser	976,800	-	-
Low-Charge Multiplex	Air-Cooled Condenser	935,200	41,600	4.3
Low-Charge Multiplex	Evaporative Condenser	863,600	113,100	11.6
Distributed	Air-Cooled Condenser	860,500	116,300	11.9
Distributed	Water-Cooled Condenser, Evaporative Rejection	866,100	110,700	11.3
Secondary Loop	Evaporative Condenser	875,200	101,600	10.4
Advanced Self-Contained	Water-Cooled Condenser, Evaporative Rejection	1,048,300	-	-
Secondary Loop	Water-Cooled Condenser, Evaporative Rejection	959,700	17,100	1.8

Results for supermarket at Washington, DC location multiplex system using evaporative condensing. Energy savings achieved by these two systems were 11.3 and 11.6 percent, respectively. The secondary loop system with evaporative condensing showed savings of 10.4 percent. Secondary loop refrigeration with water-cooled condensing and evaporative heat rejection, and the advanced self-contained system showed energy consumptions greater than that of the multiplex baseline system.

Table 7-4 gives a breakdown of the annual energy consumption of the refrigeration systems for operation in Washington, DC. Compressor energy consumption is lower for low-charge multiplex, distributed and secondary loop systems. The savings can be attributed to the close-coupling employed by the distributed and secondary loop systems and the lower condensing temperature at which the low-charge multiplex can operate. The distributed and secondary loop systems can also operate at lower condensing temperature when evaporative heat rejection is employed. Energy consumption for the secondary loop system is higher than that of the distributed system, because of the added energy needed for secondary loop pumping. Compressor energy consumption for the advanced self-contained system is significantly higher than that of the multiplex system, despite the close proximity of the compressors to the case evaporators and the use of a minimum condensing temperature of 60°F. The possible explanation for this increase is the use of unloading capacity control for the scroll compressors.

7.1.1 Impact of Heat Rejection on Energy Consumption

The initial results presented showed the refrigeration system energy consumption for each system in its most energy-efficient configuration. Additional analysis was performed to determine the impact of different heat rejection approaches on the energy consumption of the each refrigeration system type. Annual energy consumption estimates are presented in Tables 7-5, 7-6, and 7-7 for multiplex, distributed, and secondary loop refrigeration, respectively. For the multiplex and secondary loop systems air-cooled, evaporative, and water-cooled

Table 7-4. Breakdown of refrigeration annual energy consumption (kWh)

System	Compressors	Secondary Loop Pumps	Heat Rejection*	Total
Multiplex, Air-Cooled	809,400	-	167,400	976,800
Low Charge Multiplex, Air-Cooled	767,700	-	167,400	935,100
Low-Charge Multiplex, Evap Condenser	737,000	-	126,600	863,600
Distributed, Air-Cooled	690,700	-	169,800	860,500
Distributed, Water-Cooled, Evap	697,500	-	168,600	866,100
Secondary Loop, Evap Condenser	684,500	74,500	116,200	875,200
Advanced Self-Contained, Water-Cooled, Evap	867,700	-	180,600	1,048,300

Results for supermarket at Washington, DC location
*Heat rejection includes energy consumed by fans and pumps (for water loops and evaporative cooling)

Table 7-5. Impact of heat rejection on multiplex refrigeration

Location	Air-Cooled	Water-Cooled, Evaporative Tower	Water-Cooled, Dry Tower	Evaporative
Worcester, MA	904,500	916,100	1,025,800	836,100
Washington, DC	976,800	975,700	1,102,700	896,400
Memphis, TN	1,050,200	1,050,400	1,192,200	964,800
Los Angeles, CA	1,067,200	1,003,800	1,196,400	928,700

Table 7-6. Impact of heat rejection on distributed refrigeration

Location	Air-Cooled	Water-Cooled, Evaporative Tower	Water-Cooled, Dry Tower
Worcester, MA	785,700	802,200	898,600
Washington, DC	860,500	866,100	977,900
Memphis, TN	942,800	943,700	1,077,900
Los Angeles, CA	958,400	896,400	1,064,800

Table 7-7. Impact of heat rejection on secondary loop refrigeration

Location	Evaporative Condenser	Air-Cooled	Water-Cooled, Evaporative Tower	Water-Cooled, Dry Tower
Worcester, MA	821,600	880,500	900,000	1,266,700
Washington, DC	875,200	951,400	959,700	1,327,200
Memphis, TN	940,400	1,032,600	1,034,800	1,339,500
Los Angeles, CA	892,300	1,039,500	979,200	1,370,700

condensing were considered. Two types of heat rejection for the water-cooled condensing were evaluated; either evaporative or dry fluid coolers were used for final rejection. For the distributed refrigeration, air-cooled condensing, and water-cooled condensing using either evaporative or dry heat rejection were analyzed.

Table 7-8 shows the analysis results for low-charge multiplex refrigeration where air-cooled and evaporative condensing were examined. Energy savings are seen for low-charge multiplex for all locations. Savings are increased when evaporative condensing is employed. Evaporative condensing requires lower fan energy to maintain low condensing temperature operation. This difference in fan energy is a major portion of the savings increase. The remainder can be attributed to lower condensing temperature achieved through the use of evaporative condensing during warm weather operation.

Table 7-9 compares the energy consumption of multiplex with air-cooled condensing to distributed with dry heat rejection. The use of dry rejection increases the energy consumption of the distributed system significantly. The reason for this is the added temperature difference incurred in heat rejection, which raises the refrigeration condensing temperature. This increase has the largest impact during the summer months, raising the condensing temperature of the

Table 7-8. Estimated annual energy consumption for standard and low-charge multiplex refrigeration

Location	Multiplex System Type	Annual System Energy (kWh)	Low-Charge Savings	Percent Savings
Worcester, MA	Standard	904,600		
	Low-Charge Air-Cooled	850,000	54,600	6.0
	Low-Charge Evaporative Condenser	791,600	112,900	12.5
Washington, DC	Standard	976,800		
	Low-Charge Air-Cooled	935,200	41,400	4.3
	Low-Charge Evaporative Condenser	863,600	113,200	11.6
Memphis, TN	Standard	1,058,000		
	Low-Charge Air-Cooled	1,027,100	25,600	2.4
	Low-Charge Evaporative Condenser	941,500	116,400	11.0
Los Angeles, CA	Standard	1,067,200		
	Low-Charge Air-Cooled	1,042,600	24,600	2.3
	Low-Charge Evaporative Condenser	911,300	156,000	14.6

Table 7-9. Annual energy consumption comparison between multiplex with air-cooled condensing and distributed with dry heat rejection

	Refrigeration System		
Location	Multiplex (kWh)	Distributed (kWh)	Savings Achieved by Distributed (kWh)
Worcester, MA	904,600	898,600	6,000 (0.7%)
Washington, DC	976,800	977,900	-1,100
Memphis, TN	1,058,200	1,077,900	-19,700
Los Angeles, CA	1,067,200	1,064,800	2,400 (0.2%)

Multiplex employs air-cooled condensing.
Distributed employs water-cooled condensing with a dry fluid cooler.

distributed system above that of the multiplex. The higher condensing temperature also decreases the number of hours that the distributed system can operate at the minimum condensing temperature during winter months. Some energy savings were seen for the distributed system at two locations, Worcester, MA and Los Angeles, CA; but negative energy savings were predicted for operation in Washington, DC and Memphis, TN. These results show the value of evaporative heat rejection for close-loop cooling systems.

Table 7-10 compares energy consumption values of multiplex with evaporative condensing and distributed with evaporative heat rejection. An increased condensing temperature can still be expected for the distributed system because of the added temperature difference of the fluid loop. Some added energy consumption for the distributed system can be anticipated for summer operation because of increased condenser temperature. Winter operation is less likely to be impacted, and operation at minimum condensing temperature can be expected for most of the time. Energy savings range from 2.2 to 4.0 percent for this configuration. The largest savings are seen for operation in Worcester, MA, which has the coldest climate.

Table 7-11 compares the performance of multiplex and distributed refrigeration when both employ water-cooled condensing and evaporative heat rejection. In this situation, the temperature differences incurred in heat rejection are the same for both systems. Energy savings achieved by the distributed system ranged from 10.2 to 12.4 percent for the sites examined.

7.1.2 Analysis of Refrigeration Heat Reclaim for Space Heating

The initial analysis work done for refrigeration heat reclaim for space heating was to determine the correct value of minimum condensing temperature to be used to maximize savings obtained. Minimum condensing temperature value affects the amount of heat reclaimed, which increases as the condensing temperature increases. It also impacts the energy consumption of the refrigeration system, which will also increase as the condensing temperature is increased. In order to determine the best operating point, multiple analyses were conducted in which different values of minimum condensing temperature were employed. The annual energy consumption of the refrigeration system and the amount of heat reclaim were calculated at each of the

Table 7-10. Annual energy consumption comparison between multiplex with evaporative condensing and distributed with evaporative heat rejection

Location	Refrigeration System Multiplex (kWh)	Distributed (kWh)	Savings Achieved by Distributed (kWh)
Worcester, MA	836,100	802,200	33,900 (4.0%)
Washington, DC	896,400	866,100	30,300 (3.4%)
Memphis, TN	964,800	943,700	21,100 (2.2%)
Los Angeles, CA	928,700	896,400	32,300 (3.5%)

Multiplex employs evaporative condensing.
Distributed employs water-cooled condensing with an evaporative fluid cooler.

Table 7-11. ***Annual energy consumption comparison between multiplex and distributed refrigeration with water-cooled condensing and evaporative heat rejection***

	Annual Energy Consumption (kWh) Refrigeration System		
Location	Multiplex	Distributed	Savings Achieved by Distributed (kWh)
Worcester, MA	916,100	802,200	113,900 (12.4%)
Washington, DC	975,700	866,100	109,600 (11.2%)
Memphis, TN	1,050,400	943,700	106,700 (10.2%)
Los Angeles, CA	1,003,800	896,400	107,400 (10.7%)

Multiplex employs water-cooled condensing with an evaporative fluid cooler.
Distributed employs water-cooled condensing with an evaporative fluid cooler.

temperatures. A comparison of operating cost could then be made which includes the cost of electric energy to operate the refrigeration and the value of natural gas, which is displaced by the reclaimed heat for space heating.

Table 7-12 and Figure 7-2 give the results of such an analysis for a supermarket located in Worcester, MA. The results also include the no heat reclaim situation where the minimum condensing temperature is maintained at 70°F. The analysis showed that the amount of heat recovered is strongly influenced by the condensing temperature. The amount of heat reclaim seen at a 70°F condensing temperature is small and limited to desuperheating of the refrigerant only, due to the small temperature difference existing between the refrigerant and the circulated air. Heat recovery at 70°F is often at less than full desuperheating because of this small temperature difference. The combined operating costs for the heat reclaim and no heat reclaim systems are very close with a difference of only $783. At a minimum condensing temperature of 80°F, the amount of heat recovered is substantially more due to the increase in temperature difference at the heat reclaim coil. Full desuperheating is achieved at this condensing temperature. Savings achieved with heat reclaim at 80°F condensing are close to the largest savings seen. The graph in Figure 7-2 suggests that the optimum condensing temperature is approximately 80°F, where the lowest combined operating cost for refrigeration and heating is achieved. At higher condensing temperatures, more heat is reclaimed, but the amount of refrigeration energy consumed increases dramatically, so that no savings are seen at these higher condensing temperatures.

The same analysis was conducted for a supermarket operating in Los Angeles, CA. The results are shown in Table 7-13 and Figure 7-3. The results are similar to that seen for Worcester, where the lowest cost of operation is at a condensing temperature of 80°F. The savings achieved are considerably less than were seen at the Worcester site, making heat reclaim questionable for warm climates where space heating is of less significance.

Table 7-12. Heat reclaim performance multiplex refrigeration with air-cooled condensing Worcester, MA location

Minimum Condensing Temperature (ºF)	Annual Energy Consumption: Refrigeration (kWh)	Annual Energy Consumption: Heating Gas (MMBTU)	Annual Energy Consumption: Added Fan Energy (kWh)	Annual Operating Cost ($): Refrigeration	Annual Operating Cost ($): Heating	Annual Operating Cost ($): Total
70 (No Heat Reclaim)	904,600	3,155		83,223	23,158	106,381
70	904,600	2,675	29,800	83,223	22,376	105,599
80	954,600	1,940	29,800	87,823	16,981	104,804
90	1,006,700	1,803	29,800	92,616	15,976	108,592
100	1,090,700	1,626	29,800	100,344	14,676	115,021

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Operating Costs based upon an electric cost of $0.092/kWh and a gas cost of $7.34/MMBTU.

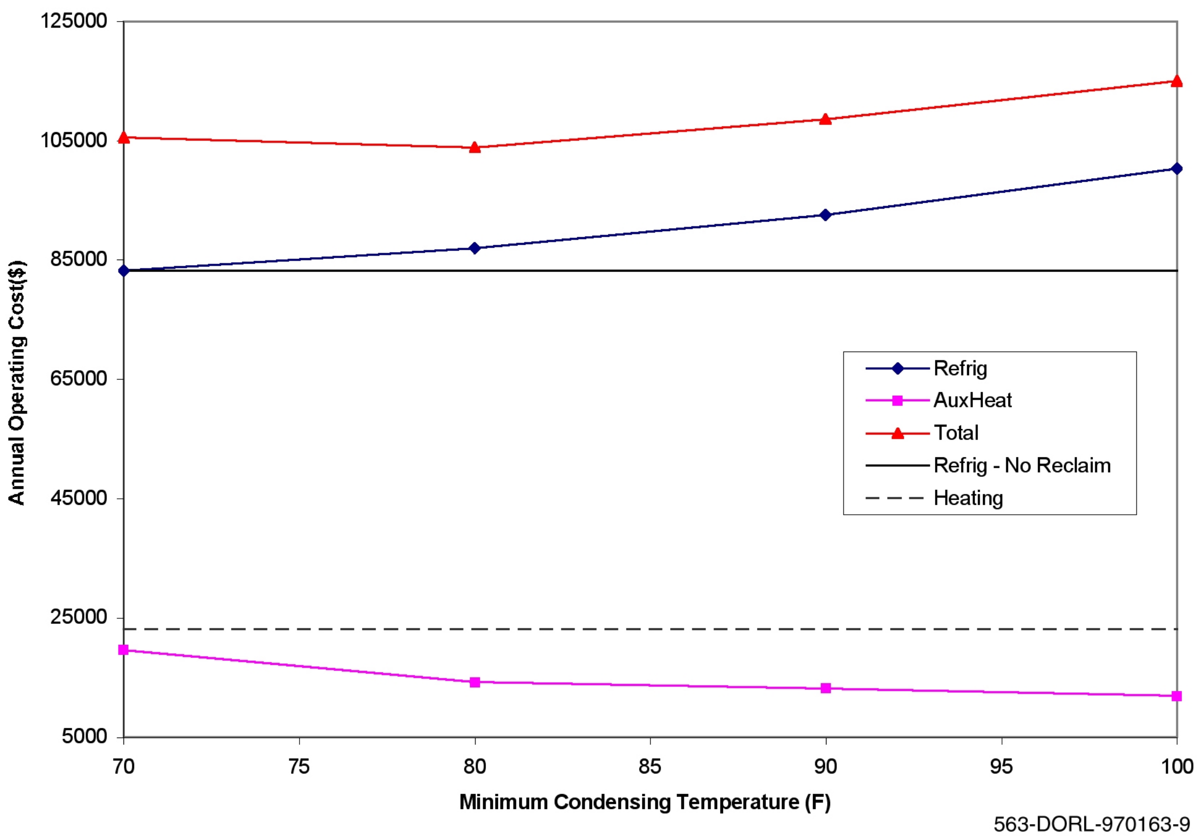

Figure 7-2. Analysis of heat reclaim for space heating, multiplex refrigeration with air-cooled condensing, Worcester, MA location

Table 7-13. Heat reclaim performance multiplex refrigeration with air-cooled condensing Los Angeles, CA location

Minimum Condensing Temperature (ºF)	Annual Energy Consumption: Refrigeration (kWh)	Heating Gas (MMBTU)	Added Fan Energy (kWh)	Annual Operating Cost ($): Refrigeration	Heating	Total
70 (No Heat Reclaim)	1,067,200	882		98,182	5,671	103,854
70	1,067,200	610	29,784	98,182	6,662	104,845
80	1,098,400	79	29,784	101,053	3,248	104,301
90	1,141,300	55	29,784	105,000	3,094	108,093
100	1,217,600	29	29,784	112,019	2,927	114,946

Operating Costs based upon an electric cost of $0.092/kWh and a gas cost of $6.43/MMBTU.

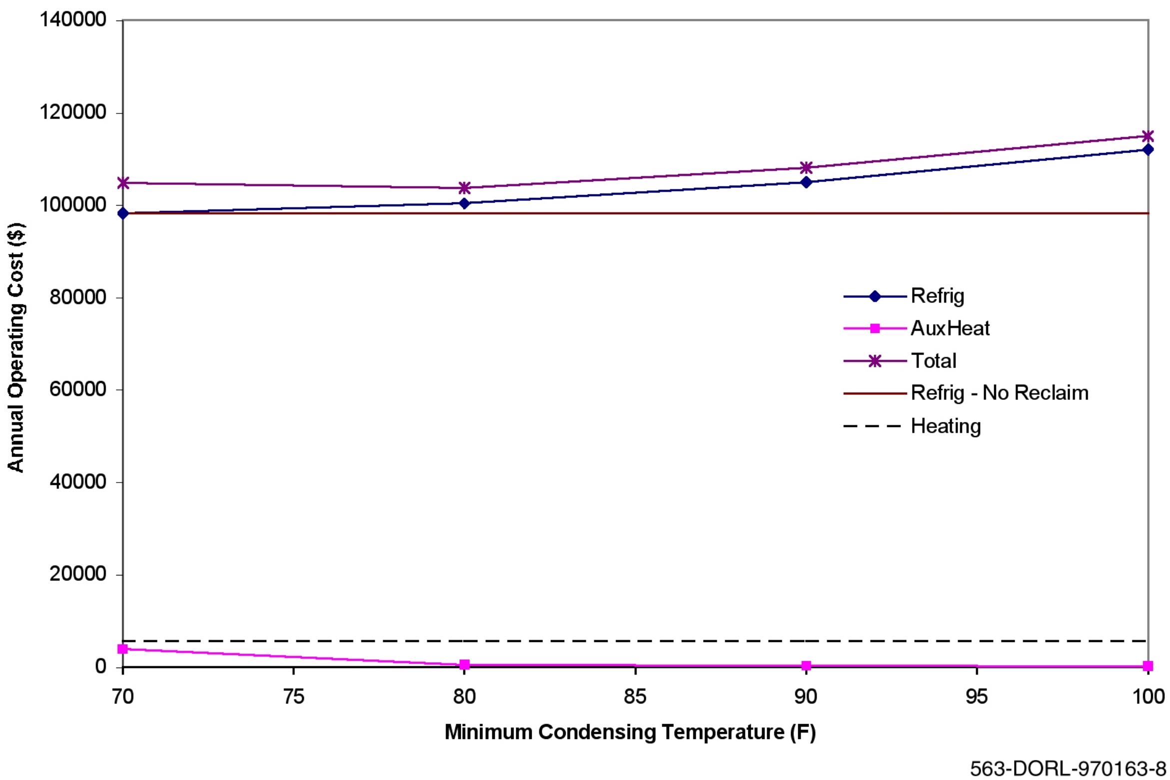

Figure 7-3. Analysis of heat reclaim for space heating, multiplex refrigeration with air-cooled condensing, Los Angeles, CA location

7.2 Evaluation of Environmental Impact of Low Charge Refrigeration

The environmental benefit that can be derived by the use of advanced, low charge refrigeration is a significant reduction in the amount of halogenated refrigerants now used in supermarkets. Present supermarkets employ as much as 3,000 lb of refrigerant, most of which is HCFC-22. This refrigerant has an ozone depletion potential (ODP) of 0.055 and a global warming potential (GWP) of 1700. The latest replacement refrigerants are HFCs, such as R-134a, R-404A, and R-507, which have an ODP of 0, but have high GWP values, in the range of 1300, 3260, and 3300, respectively (7-1).

All refrigeration systems considered here offer better approaches in terms of reduction and containment of refrigerant. Some variation in charge requirement is seen depending upon the type of heat rejection employed. Lowest charge is required by systems employing a fluid loop for heat rejection. The charge requirement for close-coupled systems, such as the distributed, secondary loop, and advanced self-contained is less due to the reduction in suction-side piping. The estimated charge for a distributed refrigeration system with fluid-loop heat rejection is approximately 900 lb, which is based on a charge requirement for each compressor cabinet of 90 lb, and, typically, 10 cabinets are needed in a supermarket. The charge associated with the secondary loop refrigeration system employing evaporative condensing is approximately 500 lb, which is split between two chiller systems. The charge requirement of the secondary loop sysstem can be reduced further to approximately 200 lb if water-cooled condensers and a fluid loop are used for heat rejection. For either system, the significant reduction in refrigerant piping drastically reduces the annual leakage rate to no more than 5 percent of total charge annually.

The environmental impact of the supermarket refrigeration system, including both the refrigerant charge and energy consumption can be determined through the use of the total equivalent warming impact (TEWI), which is a measure of the direct impact of refrigerant emissions and the indirect impact of electric generation on global warming. The direct portion of the TEWI shows the effect of refrigerant emissions on global warming due to the atmospheric lifetime of the refrigerant. The indirect portion shows the emission of carbon dioxide during generation of electric energy to drive the refrigeration system.

The TEWI can be calculated on an annual basis from the relation:

$$TEWI = Mass_{ref} * GWP_{ref} + E_{annual} * C$$

where

$Mass_{ref}$ = the amount of refrigerant leaking from the system annually (kg)
GWP_{ref} = Global Warming Potential of the refrigerant
E_{annual} = the annual electric energy consumption of the refrigeration system
C = the emission rate of CO_2 associated with electric generation. For North America, the accepted value is 0.65 kg of CO_2 per kWh

Table 7-14 shows the results of the TEWI calculations for the refrigeration systems considered here. The estimates are given for a system life of 15 years at a location in Washington, DC. The system leak rates were taken from TEWI investigation conducted by Oak Ridge National Laboratory (7-2).

The results show that the TEWI values for the distributed and secondary loop systems are significantly lower than that of the multiplex system. The lowest TEWI value is achieved by the distributed system employing a fluid loop for heat rejection. The low-charge multiplex system shows some reduction in TEWI versus the baseline multiplex, due to the reductions in system charge and energy use. The advanced self-contained systems has the lowest direct TEWI value, but has the largest indirect value due to high energy consumption.

Table 7-15 gives the estimated operating savings for the low-charge systems due to reduced refrigerant leakage. For this analysis, refrigerant costs of $1.75/lb for R-22, and $7.75/lb for R-404A and R-507, were used, respectively.

7.3 Analysis Results for HVAC

The results for the analysis of supermarket HVAC systems are shown in Tables 7-16 and 7-17. Table 7-16 shows the annual electric and gas consumption for a conventional HVAC

Table 7-14. Total Equivalent Warming Impact (TEWI) for supermarket refrigeration

						TEWI (million kg of CO_2)		
System	Condensing	Charge (lb)	Refrigerant	Leak (%)	Annual Energy (kWh)	Direct	Indirect	Total
Multiplex	Air-Cooled	3,000	R404A/	30	976,800	13.62	9.52	23.15
	Evaporative	3,000	R-22	30	896,400	13.62	8.74	22.36
Low-Charge	Air-Cooled	2,000	R404A/	15	935,200	4.54	9.12	13.66
Multiplex	Evaporative	2,000	R-22	15	863,600	4.54	8.42	12.96
Distributed	Air-Cooled	1,500	R404A	10	860,500	3.33	8.38	11.71
Distributed	Water-Cooled, Evaporative	900	R404A	5	866,100	1.00	8.44	9.44
Secondary Loop	Evaporative	500	R507	10	875,200	1.13	8.54	9.67
Secondary Loop	Water-Cooled, Evaporative	200	R507	5	959,700	0.23	9.36	9.58
Advanced Self-Contained	Water-Cooled, Evaporative	100	R404A	1	1,048,300	0.02	10.22	10.24

Results for site in Washington, DC – 15 year service life.
Conversion factor = 0.65 kg CO_2/kWh.
Multiplex – 33.3% R404A (low temperature), GWP = 3260; 66.7% R22 (medium temperature), GWP = 1700.
Distributed and Advanced Self-Contained – 100% R404A, GWP = 3260.
Secondary Loop – 100% R507, GWP = 3300.

Table 7-15. Estimated operating cost savings for reduced refrigerant leakage

System	Annual Leakage (lb) R-404A	R-507	R-22	Savings ($)
Multiplex - (R-404A/R-22)	300		600	
Multiplex - Low Charge	100		200	2,250
Multiplex - Low Charge Evap Cond	100		200	2,250
Distributed Air-Cooled	150			2,213
Distributed Water-Cooled, Evap	45			3,026
Secondary Loop Evap Condensing		50		2,988
Secondary Loop Water-cooled, Evap		10		3,298
Advanced Self-Contained	1			3,367

Table 7-16. Annual energy consumption for conventional supermarket HVAC

Location	System	Electric Consumption (kWh) Reclaim	HVAC Fans	Cooling	Total	Gas Consumption (MMBTU)
Worcester, MA	Conventional	0	160,307	45,498	205,805	3,155
	Heat Reclaim	40,200	190,100	45,498	275,798	1,894
Washington, DC	Conventional	0	160,307	68,085	228,392	2,392
	Heat Reclaim	30,200	190,100	68,085	288,385	1,215
Memphis, TN	Conventional	0	160,307	119,458	279,765	1,636
	Heat Reclaim	31,600	190,100	119,458	341,158	637
Los Angeles, CA	Conventional	0	160,307	43,207	203,514	882
	Heat Reclaim	24,000	190,100	43,207	257,307	72

Table 7-17. Annual energy consumption for water-source heat pump HVAC

Location	Electric Energy Consumption (kWh) HVAC Fans	Water-source Heat Pump Cooling	Heating	Loop Pump	Total	Gas Consumption (MMBTU)
Worcester, MA	160,307	35,920	120,004	26,280	342,511	12
Washington, DC	160,307	54,429	91,095	26,280	332,112	0
Memphis, TN	160,307	96,656	60,998	26,280	344,241	0
Los Angeles, CA	160,307	35,321	32,920	26,280	254,829	0

system, consisting of rooftop units alone, and for rooftop units with refrigeration heat reclaim for space heating. The electric consumption is divided into the following categories:

- Reclaim – added energy used by the refrigeration system for heat reclaim.
- HVAC fans – energy consumed for air circulation through the store, including fan energy associated with the heat reclaim heating coil.
- Cooling – energy consumed by the condensing units used to provide air conditioning.
- Gas consumption - consists of gas used by duct heaters to provide space heating.

Table 7-17 gives the annual energy consumption for HVAC provided by water-source heat pumps. Energy consumption consists of fan energy for air circulation (same as conventional system), operation of the heat pump compressors, and for water pumping. Fan energy for heat rejection is accounted for in the consumption of the refrigeration system, since the same water loop is used for both systems. Gas consumption represents heating needed to supplement the output of the water-source heat pumps.

Table 7-18 gives the operating cost associated with the energy use described above. Local utility rates were used for each location to estimate the cost. Water cost for evaporative heat rejection was included in the cooling costs associated with the water-source heat pumps. A breakdown of the operating cost is also provided. The largest element of the HVAC operating cost is the cost of energy for the circulation fans which accounts for 35 to 67.9 percent of the total cost. Heating also represents are large fraction of the cost in two locations, Worcester and Washington. In Memphis and Los Angeles, the heating and cooling costs are similar in magnitude.

Figure 7-4 shows these cost breakdowns graphically. The results indicate that the water-source heat pump system operates with the lowest annual cost in all locations considered. Table 7-19 gives the savings obtained by the water-source heat pumps when compared to conventional HVAC and conventional HVAC with heat reclaim. Savings range from 2.2 to 30.7 percent versus conventional HVAC alone, and from 3.8 to 22.1 percent versus conventional HVAC with heat reclaim for space heating. The largest savings were seen for a Washington, DC location, which had a combination of large heating load and favorable utility rates. The smallest savings versus heat reclaim were seen for a Memphis, TN location, which was due to unfavorable utility rates, primarily a low rate for natural gas. HVAC savings for a Los Angeles location were small for either heat reclaim or water-source heat pumps, because the space heating requirement for this location is smallest of all sites considered.

7.4 Analysis of Integrated Operation of Refrigeration and HVAC

An analysis was performed to determine the best integrated system approach to refrigeration and HVAC. The systems that were compared were:

Table 7-18. Annual operating costs for supermarket HVAC

Location	System	Reclaim $	Reclaim %	Fans $	Fans %	Cooling $	Cooling %	Heating $	Heating %	Total $
Worcester, MA	Conventional	0	0.0	14,748	35.0	4,186	9.9	23,158	55.0	42,092
	Heat Reclaim	4,609	11.5	17,488	43.5	4,186	10.4	13,902	34.6	40,185
	WSHP	0	0.0	14,748	46.7	5,975	14.3	12,337	39.0	33,060
Washington, DC	Conventional	0	0.0	11,382	33.6	4,834	14.3	17,653	52.1	33,869
	Heat Reclaim	2,833	9.4	13,496	44.8	4,834	16.0	8,967	29.8	30,130
	WSHP	0	0.0	11,382	50.2	5,605	21.2	6,490	28.6	23,477
Memphis, TN	Conventional	0	0.0	10,260	36.8	7,645	27.4	9,996	35.8	27,901
	Heat Reclaim	2,534	9.4	12,166	45.3	7,645	28.5	4,503	16.8	26,849
	WSHP	0	0.0	10,260	42.2	8,293	28.9	7,027	28.9	25,580
Los Angeles, CA	Conventional	0	0.0	14,748	60.5	3,975	16.3	5,671	23.2	24,395
	Heat Reclaim	2,870	11.6	17,488	70.5	3,975	16.0	463	1.9	24,797
	WSHP	0	0.0	14,748	65.7	4,641	14.5	4,458	19.9	23,848

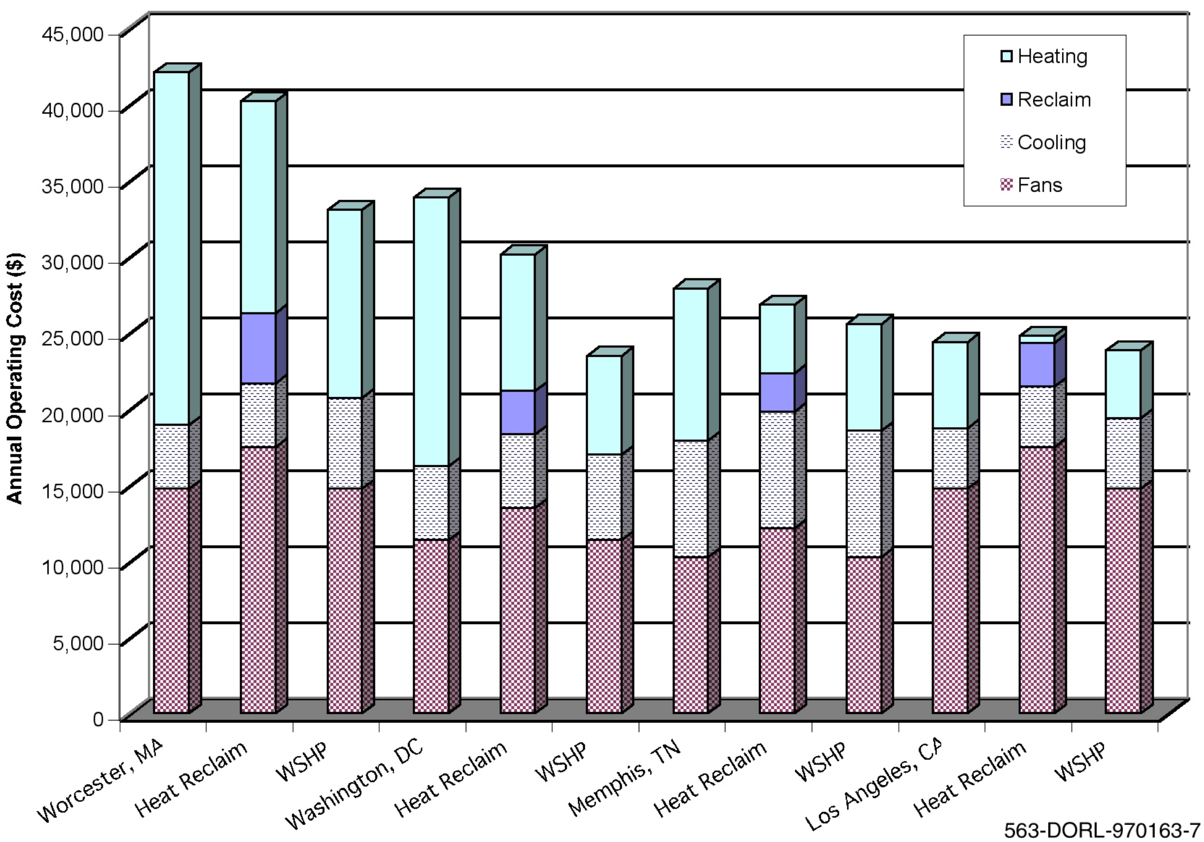

Figure 7-4. Comparison of supermarket HVAC systems

Table 7-19. HVAC operating cost savings achieved by water-source heat pumps

	Annual Savings ($)	
Location	versus Conventional	vesus Heat Reclaim
Worcester, MA	9,032 (21.4%)	7,125 (17.7%)
Washington, DC	10,391 (30.7%)	6,653 (22.1%)
Memphis, TN	2,321 (8.3%)	1,269 (4.7%)
Los Angeles, CA	547 (2.2%)	949 (3.8%)

- Multiplex refrigeration with conventional rooftop units for HVAC.
- Multiplex refrigeration with heat reclaim and conventional HVAC.
- Distributed refrigeration with water-source heat pumps for HVAC.

Table 7-20 and Figure 7-5 show the estimated operating costs for refrigeration and HVAC for the sites examined. The lowest cost approach was the combination of distributed refrigeration and water-source heat pumps for all four locations. Table 7-21 gives the annual savings achieved for each site. The savings ranged from a low of 11.1 percent for Memphis, TN to a high of 19.2 percent for Washington, DC. Table 7-21 also contains a comparison of multiplex refrigeration and conventional HVAC with and without heat reclaim. Savings were seen for heat reclaim operation at all locations except Los Angeles. Savings achieved by heat

Table 7-20. Estimated annual operating cost for supermarket refrigeration and HVAC

		Annual Operating Cost ($)				
		Energy				
Location	System	Refrigerant	HVAC	Refrigerant	Water	Total
Worcester, MA	Multiplex	83,214	42,092	3,375	0	128,681
	Multiplex and Heat Reclaim	86,912	35,576	3,375	0	125,864
	Distributed and WSHP	73,802	31,599	349	2,922	108,672
Washington, DC	Multiplex	69,353	33,869	3,375	0	106,597
	Multiplex and Heat Reclaim	72,193	27,297	3,375	0	102,865
	Distributed and WSHP	61,493	22,669	349	1,616	86,127
Memphis, TN	Multiplex	67,213	27,901	3,375	0	98,489
	Multiplex and Heat Reclaim	69,231	24,314	3,375	0	96,921
	Distributed and WSHP	60,365	24,314	349	2,533	87,560
Los Angeles, CA	Multiplex	98,182	24,395	3,375	0	125,952
	Multiplex and Heat Reclaim	101,047	21,926	3,375	0	126,348
	Distributed and WSHP	82,285	22,456	349	2,783	107,873

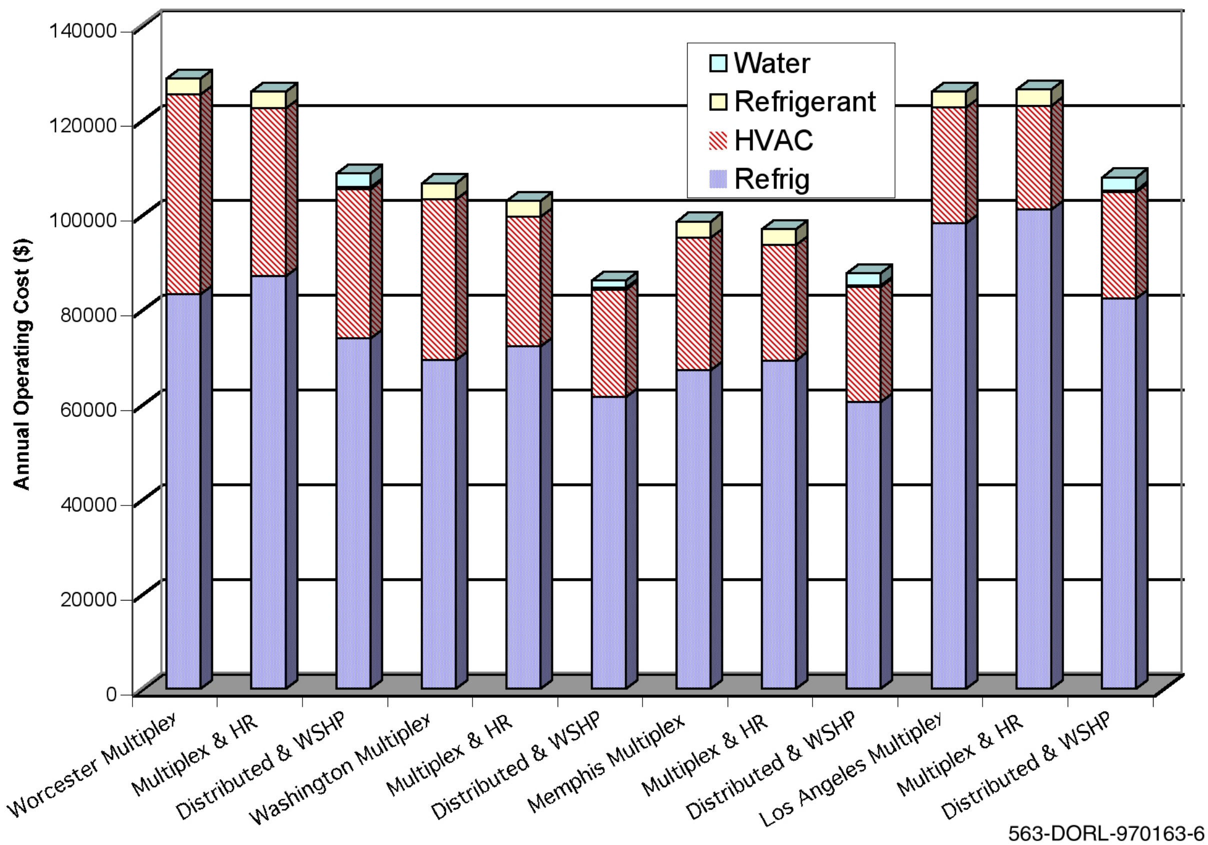

Figure 7-5. Operating cost of supermarket refrigeration and HVAC

Table 7-21. Annual cost savings achieved by distributed refrigeration and water-source heat pumps versus multiplex refrigeration and conventional HVAC

	Annual Operating Savings			
	with Heat Reclaim		Distributed Refrigeration and WS Heat Pumps	
Location	$	%	$	%
Worcester, MA	2,817	2.2	20,009	15.5
Washington, DC	3,732	3.5	20,469	19.2
Memphis, TN	1,568	1.6	10,929	11.1
Los Angeles, CA	-397	-0.3	18,079	14.4

reclaim were considerably less than seen with distributed refrigeration and water-source heat pumps, ranging from -0.3 to 3.5 percent.

7.5 Payback Analysis for Advanced Systems

Table 7-22 gives the estimated installed cost premiums for distributed, secondary loop, and low-charge multiplex refrigeration systems. Estimates are based on actual construction budgets

Table 7-22. Estimated installed cost premiums for low-charge supermarket refrigeration systems

System	Installed Cost Premium ($) Equipment	Installation	Total
Multiplex	Baseline		
Low-charge Multiplex	0	0	0
Distributed, Water-cooled	53,000	7,000	60,000
Secondary Loop, Evaporative	70,000	77,000	147,000

supplied by the engineering departments of two supermarket chains. For purposes of confidentiality between the chains and their vendors, the identities of these supermarkets will not be given in this report. It should be noted that the actual installed cost of any refrigeration system will vary greatly, depending upon many factors, such as: purchasing arrangements between the supermarket and refrigeration equipment vendors; whether or not display cases are purchased in conjunction with the refrigeration system; special system features or configuration requested by the supermarket; and unique installation requirements of each site. The lowest cost premium is seen with the low-charge multiplex system, which was estimated to have the same installed cost as the baseline multiplex system. The distributed system showed higher equipment cost, but only a small increase in installation cost. This can be attributed to reduced refrigeration piping cost, but also increased electrical and fluid loop costs. The equipment and installation cost premiums of the secondary loop system are similar in magnitude.

The cost premium for the water-source heat pumps is shown in Table 7-23 and is estimated at $25,000, which consists of $15,000 in added equipment cost and $10,000 in installation cost. The extra installation cost includes water piping for the heat pumps and over-sizing of the refrigeration heat rejection to allow heat pump heat rejection during space cooling.

Table 7-24 gives the estimated simple paybacks for each of the low-charge systems at each of the sites examined. Operating cost savings for the refrigeration include the savings obtained for reduced energy use and refrigerant leakage. Water costs are included for all systems employing evaporative heat rejection. The low-charge multiplex achieves immediate payback, since no installed cost premium exists. Highest savings for the low-charge multiplex are seen when evaporative condensing is employed. Paybacks for the distributed system ranged from 3.4 to 7.0 years, while the payback for the secondary loop system ranged from 8.3 to 16.8 years.

Table 7-23. Estimated installed cost premiums for water-source heat pumps

HVAC (Water-source heat pumps)	Cost ($)
Equipment	15,000
Installation	10,000
Total Installed Cost Premium for HVAC	25,000

Table 7-25 shows the payback associated with the use of distributed refrigeration in combination with water-source heat pumps for store HVAC. The combined payback for refrigeration and

Table 7-24. Estimated energy savings and payback for low-charge supermarket refrigeration (versus multiplex with air-cooled condensing)

Location	Low-Charge Multiplex, Air-Cooled		Low-Charge Multiplex, Evaporative		Distributed, Water-Cooled, Evaporative		Secondary Loop, Evaporative	
	$	Year	$	Year	$	Year	$	Year
Worcester, MA	7,264	0	11,176	0	10,977	5.5	9,153	16.1
Washington, DC	5,204	0	9,479	0	10,078	6.0	9,393	15.6
Memphis, TN	3,728	0	7,940	0	8,608	7.0	8,748	16.8
Los Angeles, CA	4,513	0	15,201	0	17,532	3.4	17,678	8.3

Table 7-25. Estimated payback for distributed refrigeration and water-source heat pumps

Location	Savings ($)		Payback (Year)	
	Refrigeration	Combined	Refrigeration	Combined
Worcester, MA	10,977	20,009	5.5	4.2
Washington, DC	10,078	20,469	6.0	4.2
Memphis, TN	8,608	10,929	7.0	7.8
Los Angeles, CA	17,532	18,079	3.4	4.7

HVAC savings was less than that for refrigeration savings alone for the Worcester and Washington sites, because of increased space heating savings. Combined paybacks for the Memphis and Los Angeles sites were longer, because space heating is not as significant at these locations.

Section 7 References

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8. CONCLUSIONS AND RECOMMENDATIONS

The investigation of low charge supermarket refrigeration showed that at present, four system approaches are available, consisting of:

- Distributed refrigeration – compressors cabinets located throughout the store and a fluid loop for heat rejection.
- Secondary loop refrigeration – central chiller is used to cool a secondary fluid that is pumped to the display cases to provide refrigeration.
- Advanced self-contained - each display case is equipped with a water-cooled condensing unit. A fluid loop is connected at all condensing units and is used for heat rejection.
- Low-charge multiplex - the multiplex refrigeration system is equipped with control piping and valves to allow operation at close to critical charge, greatly reducing the amount of refrigerant needed.

All of these advanced systems provide substantial reductions in refrigerant charge, and energy savings versus multiplex systems were shown for all systems with the exception of the advanced self-contained.

Further investigation and analysis was performed to predict the energy consumption for these low-charge systems and compare their performances to multiplex refrigeration with air-cooled condensing and mechanical subcooling for low temperature refrigeration, which is now the most commonly installed supermarket refrigeration system. Results from this analysis showed that the largest energy savings were achieved by the distributed and secondary loop refrigeration systems. The distributed system produced the energy savings, ranging from 10.2 to 16.2 percent of multiplex consumption. Secondary loop refrigeration reductions in energy of 9.2 to 16.4 percent for the locations investigated. The secondary loop system had higher energy savings than the distributed system for the Memphis and Los Angeles sites, while the distributed showed lower energy consumption for the Worcester and Washington sites. The low-charge multiplex system showed less energy use than the multiplex baseline for all locations. Savings ranged from 2.2 to 6.0 percent and 10.4 to 14.6 percent for the low-charge multiplex with air-cooled and evaporative condensing, respectively.

The energy savings achieved by the distributed refrigeration system can be attributed to close-coupling of the compressors to the display case evaporators, operation of the scroll compressors at 60°F minimum condensing temperature, and the use of evaporative heat rejection with the fluid loop. Savings seen with the secondary loop system are due to close-coupling of the compressors and the chiller evaporator, subcooling produced by brine heating for defrost, and the use of evaporative condensing. The refrigeration energy of the secondary loop system was found to be less than that of the distributed system, but the added energy associated with brine pumping negated some of this advantage. The energy savings seen with the low-charge multiplex system were due to the ability of this system to operate at very low minimum condensing temperatures. The minimum condensing temperatures were 40 and 60°F for low and medium temperature refrigeration, respectively.

A TEWI analysis of the low-charge refrigeration systems showed that both the distributed and secondary loop systems produced TEWI values that were significantly smaller than that estimated for the multiplex. Lowest TEWI values were achieved when a fluid loop with evaporative heat rejection was employed.

Substantial operating savings were obtained by the low-charge systems through reduced refrigerant leakage. The largest refrigerant savings were seen with the advanced self-contained system which was credited with a savings of $3,367. Distributed and secondary loop systems produced refrigerant savings of $3,026 and $2,988, respectively. The smallest refrigerant savings were seen with the low-charge multiplex system at $2,250.

A payback analysis of operating costs (electric, refrigerant, and water) for the low-charge refrigeration systems showed that the low-charge multiplex system had an immediate payback, since no installed cost difference exists between the low-charge and baseline multiplex systems. The distributed system showed paybacks ranging from 3.4 to 7.0 years, while the secondary loop system showed paybacks of 8.3 to 16.8 years. These payback values are extremely sensitive to the installed cost premium for these systems, which is highly variable depending upon arrangements between the supermarkets and their equipment suppliers and installers. These cost differences are likely to be reduced for either the distributed or the secondary loop systems as more such systems are implemented.

Supermarket HVAC systems were also addressed in this report where conventional rooftop HVAC, refrigeration heat reclaim, and water-source heat pumps were considered. The lowest operating cost for HVAC was shown for water-source heat pumps, which produced cost savings of 8.3 to 30.7 percent for the four locations examined.

For combined operation of refrigeration and HVAC, the system consisting of distributed refrigeration and water-source heat pumps showed the lowest operating cost for all locations considered. Operating cost savings were estimated to be 11.1 to 19.2 percent when compared to multiplex refrigeration with conventional HVAC. The payback on cost premium for distributed refrigeration and water-source heat pumps was found to be about 4.2 years for Worcester, MA and Washington, DC, and 4.7 years for Los Angeles, CA. The simple payback operation in Memphis, TN was 10.8 years. The lowest paybacks were seen for sites with large space heating loads. For these locations the operation of the water-source heat pumps helped to reduce the combined payback of the refrigeration and HVAC systems.

The results seen in this investigation show that low-charge refrigeration systems can reduce energy and operating costs if properly designed and operated. Demonstration of these technologies by field testing, possibly in conjunction with water-source heat pumps for HVAC, will help develop best practices for these systems and also better quantify energy savings. This information will help to accelerate the use of low-charge refrigeration systems by the supermarket industry.

Supermarket Refrigeration Systems:
Excel Spreadsheet Users' Manual

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March 2003

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Introduction

Excel spreadsheets are being developed as tools to assist supermarket planners and engineers to compare the relative energy and refrigerant requirements for alternative designs of their refrigeration systems. The spreadsheets have been completed for multiplexed direct expansion and secondary loop refrigeration systems; an additional spreadsheet may be developed for distributed refrigeration systems.

Historically, supermarket refrigeration systems have consumed both large quantities of electricity to drive their compressors, fans, and lights and large quantities of refrigerants from the thousands of feet of piping connecting the display cases, walk-in coolers and prep rooms with the machine room and condensing unit. Alternative system designs are now being considered to reduce refrigerant emissions and possibly reduce energy use.

High refrigerant leakage rates were tolerated in the past due to the low cost of CFC and HCFC refrigerants, the high cost of labor, and ignorance of their impact on the environment. The cost of the refrigerants that replaced CFCs and HCFCs and concerns about ozone depletion and global warming led to efforts by equipment suppliers and end-users to reduce emission rates; these efforts included:

- rigorous preventive maintenance programs to locate and seal leaks in the traditional multiplexed systems,
- design and installation of secondary heat transfer loops to reduce the overall refrigerant charge and to isolate leaks in the machine rooms where they are easier to locate and repair, and
- design and installation of distributed refrigeration systems to reduce charge and leakage with or without heat reclaim systems to improve building energy efficiency.

As with every business decision, there are trade-offs between a desired objective and costs, in this case between reduced refrigerant emissions, design and installation costs, and operating costs. Poor choices can result in systems that reduce refrigerant emissions at the expense of significantly higher installation costs and energy use (over 35%); careful choices can reduce emissions with only minor changes in energy use and installation costs.

Excel spreadsheets have been developed with a graphical user interface to calculate annual power consumption for multiplexed direct expansion and secondary loop refrigeration systems. In either case average loads are used for different types of display cases and walk-ins and binned weather data are used to determine floating head pressures. Calculations for multiplexed systems include estimating suction line pressure drop and suction temperature using refrigerant fluid properties and piping dimensions provided by the user. Calculations for secondary loop systems include estimating the secondary fluid pumping power from piping dimensions and correlations for properties of a user selected secondary fluid. Details about the calculations are provided in a later section of this report.

This users' manual is organized as:

- an overview using a sample store layout,
- calculations and formulas,
- worksheet data entry, and

- weather data.

Sample Store Layout

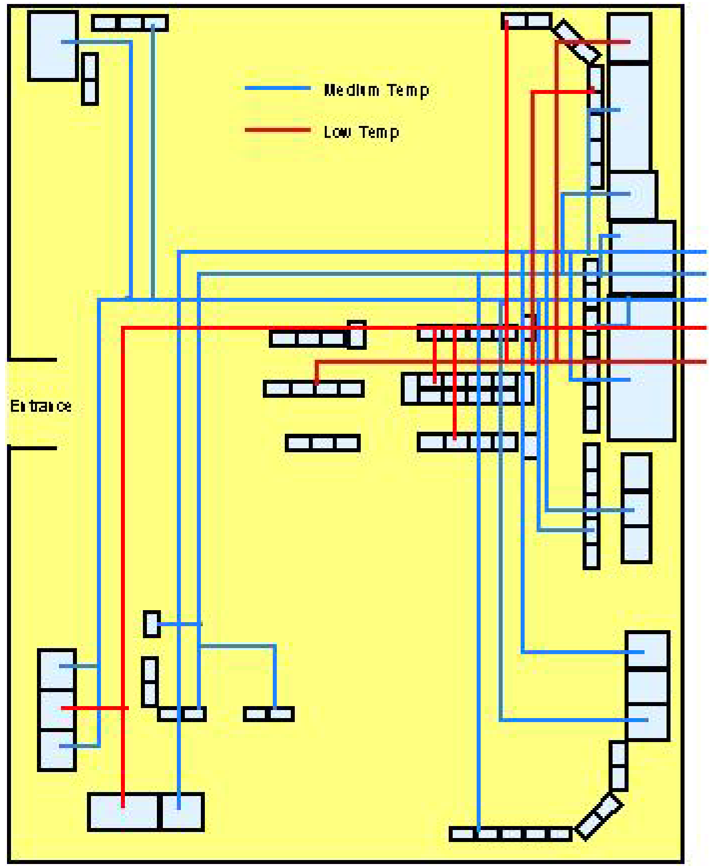


Figure 1. Secondary loop refrigeration system with central headers.

Figure 1 is a drawing illustrating refrigeration "loops" and the placement of display cases and walk-ins throughout a supermarket. In this instance, there are three medium temperature and two low temperature refrigerant or secondary coolant loops coming from the machine room at the right side of the drawing. Display cases are located along the exterior walls of the sales floor and down the middle of aisles in the center of the store. Walk-in freezers and coolers and food prep rooms are located behind the display cases illustrated on the right side of the drawing. For purposes of calculations, multiple identical display cases are treated as a single refrigeration load. The type of case (e.g. multi-deck w/ doors, coffin, dairy) is specified as well as the overall length of the line up of cases. Refrigeration loads are specified in terms of Btu/h per lineal foot. The spreadsheets are configured to accommodate up to three low temperature and up to three medium temperature refrigeration loops. Each loop can have up to ten loads, either line-ups of display cases or walk-ins (i.e. coolers, freezers, prep-rooms).

The current version of the spreadsheet assumes a relatively simple design for each secondary loops. Each loop can consist of up to ten appliances or loads (i.e. display case, walk-in, prep room) laid out in parallel along a single main and return line as illustrated in Fig. 2. The loads, of course, can be to the left or right, above or below, the main and return lines, but the branches to the loads are not allowed to branch again to serve more than a single load. The piping for each load is defined in terms of a section of the main and return lines and a branch to the appliance. The user is required to specify the pipes for the main and branch lines (i.e. length in feet, nominal size in inches, and pipe material); segments of the return line are assumed to be identical to the corresponding segments of the supply line. Lengths of the main line are defined as the distance from the

"T" for the preceding load. For instance, referring to Fig. 2, the user describes the first segment of the secondary loop by providing the materials, the lengths and the nominal sizes for piping for segment **AB** in the main line and segment **BCDE** in the branch. Segment **EF** is assumed to be identical to **AB**. The segment corresponding to the second load on the secondary loop is described in terms of segments **BG** and **GHIJ**; the third by **GK** and **KLMN**; etc.

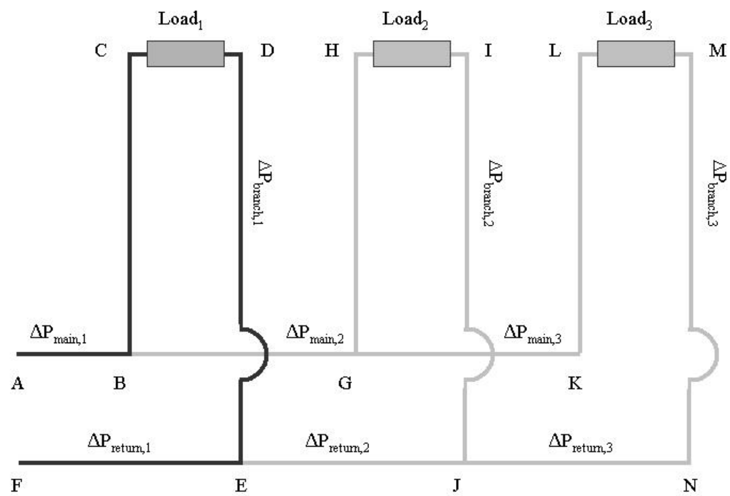


Figure 2. Schematic for secondary refrigeration loop.

Display Cases and Walk-Ins

The user provides a "description" of the refrigeration system by selecting types of loads (e.g. display cases or walk-ins) from a drop down list and providing data about piping (i.e. length and ID). Typical or representative refrigeration loads are computed for each display case or walk-in based on the type of equipment (e.g. multi-rack vertical, walk-in) and its physical dimensions (length or square footage). Two sets of data were located for supermarket refrigeration loads. The first (LBL) provides only a single representative value for several different types of display cases. The second (ASHRAE 1999) provides measured values for all the display cases and walk-ins in two different supermarkets. This information was tabulated in terms of Btu/h per foot of display case or per square foot of walk-in area. The average values from these sources are listed below with the lowest and highest values given in parentheses where a range of values exists.

Frozen Food Display Cases

multi-deck, vertical cases	1,425 Btu/h-ft
multi-deck, vertical with cases glass doors	609 (352 to 832) Btu/h-ft
tub or coffin cases	478 (275 to 689) Btu/h-ft
other frozen food cases	775 (550 to 1000) Btu/h-ft
walk-ins	85 (69 to 111) Btu/h-ft^2

Fresh Food Display Cases

multi-deck meat or dairy, vertical	1,575 (1,010 to 1,994) Btu/h-ft
multi-deck, vertical with doors	597 (550 to 660) Btu/h-ft
single-deck produce	694 (372 to 975) Btu/h-ft
tub or coffin	285 (260 to 335) Btu/h-ft

single-deck meat and seafood	296 (100 to 628) Btu/h-ft
deli, cheese, and pizza	453 (357 to 633) Btu/h-ft
other fresh food cases	934 (440 to 1,650) Btu/h-ft
walk-ins	63 (26 to 100) Btu/h-ft^2

The user is required to select the type of display case from a Windows pull-down list, specify the length or square footage, and whether to use the average, low, or high cooling load value for that appliance.

The user also needs to specify a design heat exchanger temperature (approximately 10°F below the exit air temperature) and secondary fluid pressure drop for each display case in a secondary loop system (recommended values are supplied).

Refrigerant and Secondary Loop Piping

Ultimately the decision on whether or not to use a secondary loop refrigeration system comes down to questions about installation costs and operating costs; pipes and energy. The spreadsheet provides good estimates about both materials and energy, but it requires substantial input from the user in terms of lengths of piping, numbers of elbows, pipe diameters, and secondary fluids; the secondary fluids are discussed in the next section. Pipe lengths do not need to be exact and can be estimated from a schematic or blueprint of the store; as can the number of 90° elbows.

The pressure drop in each section of piping, and the secondary loop pumping power, is strongly dependent on the pipe diameters. The smaller the pipe, the larger the ΔP and the pumping power. The desire to select small pipes to reduce materials and installation costs has to be balanced with the need to achieve low ΔP's and maintain low secondary fluid velocities in secondary loop systems (for noise control) and high vapor velocities in multiplexed DX systems (for oil return). The user can select the nominal pipe size for each section of the secondary loop from a drop down list of available sizes; the system pressure drops and pumping power are updated and displayed after each change (refrigerant velocities also be displayed for multiplexed DX systems). Experimentation with the pipe diameters usually identifies a "threshold" value; the next smaller pipe ID results in a dramatic increase in ΔP and pumping power.

Secondary Fluids

The user is required to select an operating fluid and design temperature (temperature of the fluid entering the heat exchanger of each display case) for each secondary loop; a drop-down list of available fluids is built into the spreadsheet. Currently the spreadsheet contains correlations for computing the thermophysical properties (density, dynamic viscosity, thermal conductivity, and specific heat) of thirteen heat transfer fluids:

- ethylene glycol,
- propylene glycol,
- cyclohexene,
- hydrofluoroether,
- isoparaffinic hydrocarbons,

- inhibited alkali ethanate “A”,
- inhibited alkali ethanate “B”,
- potassium formate “A”,
- potassium formate “B”,
- potassium formate “C”,
- silicon oil “A”,
- silicon oil “B”, and
- silicon oil “C”.

Data or correlations for thermophysical properties of these compounds are from the ASHRAE Handbook, the University of Illinois (Hrnjak 1996), and the U.S. EPA (Kazachi 1997). As with the pipe sizes, pumping power is also strongly dependent on the properties of the operating fluid; the user can experiment with the spreadsheet to determine the energy use associated with each fluid.

While the spreadsheet enables users to do quick comparisons of the pumping power required for different secondary fluids, it does not provide any insights or caveats into the other properties of these materials. Simple experimentation with the spreadsheet shows the attractive thermodynamic properties of a couple of secondary fluids particularly for low temperature secondary loops, with pumping powers significantly lower than those achievable with many other fluids. There is not any mechanism in the spreadsheet, though, to alert the user to problems with corrosion and environmental cleanup regulations relative to these fluids. These are important issues that need careful consideration before accepting one secondary fluid over another.

Secondary Loop Pumping Power

The secondary loop pumping power is computed using the ΔP of each loop and fixed pump and motor efficiencies (50% and 87½%, respectively). The pumps are assumed to operate 24 hours per day, 365 days per year since the refrigeration they are providing is based on specified hourly loads (this method is used instead of specifying equipment capacity and estimated run time)

Compressors and Compressor Power

Data are built into the spreadsheet for the performance of two compressors; one each for low temperature and medium temperature refrigeration. Figures 3 and 4 show the efficiencies of both compressors. The main spreadsheet contains all the information necessary to replace these compressor curves with maps for different compressors.

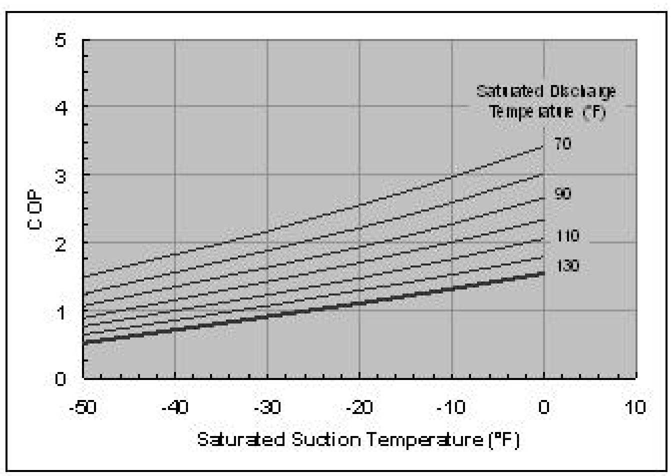


Figure 3. Low temperature compressor COP.

Heat Rejection: Condensers and Fluid Cooler Fan Power

The condenser fan power for low and medium temperature refrigeration systems are computed using specified values for the energy required to reject 1,000 Btu/h of heat. Recommended values are provided.

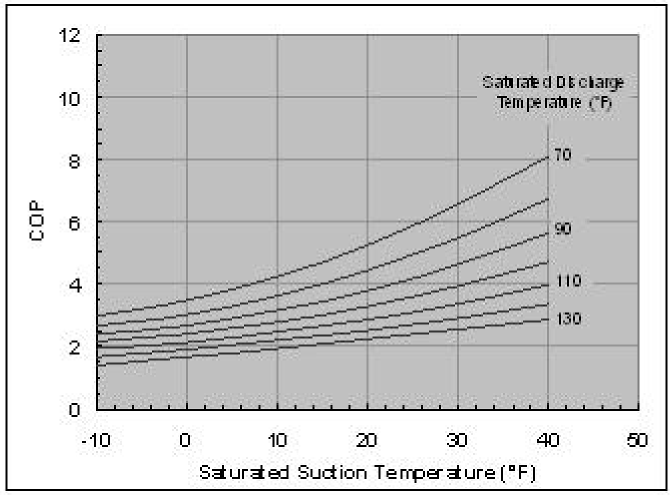


Figure 4. Medium temperature compressor COP.

City Database

Weather conditions are used to compute compressor efficiency and condenser fan power. The user is required to select a city in the U.S. from a built in list; the spreadsheet uses the dry bulb temperature distribution and mean coincident wet bulb temperatures from internal tables to compute compressor and fan power. There is at least one city to choose from in each state; most states have several cities to choose from. A complete list of cities is included in the appendix. This database uses typical meteorological year (TMY) weather data from the National Renewables Energy Laboratory (Marion 1995) and employs 5°F temperature bins to reduce the quantity of hourly calculations.

Calculations and Formulas

A brief description of the calculations used in the spreadsheet follows.

Compressor Calculations

Compressor and condenser power consumption are calculated for each of the temperature bins in the analysis using the corresponding dry bulb and mean coincident wet bulb temperatures. Compressor efficiency is calculated using curve fits for the efficiency ratio (EER) from the saturated suction temperature (SST) and the saturated discharge temperature (SDT) as shown in Eq. 1:

$$\begin{aligned} EER = a_0 + a_1 \cdot T_{SDT} + a_2 \cdot T_{SST} + a_3 \cdot T_{SDT}^2 + a_4 \cdot T_{SDT} \cdot T_{SST} + a_5 \cdot T_{SST}^2 + \\ a_6 \cdot T_{SDT}^3 + a_7 \cdot T_{SDT}^2 \cdot T_{SST} + a_8 \cdot T_{SDT} \cdot T_{SST}^2 + a_9 \cdot T_{SDT}^3 \end{aligned} \qquad (1)$$

The spreadsheet computes the saturated suction and discharge temperatures from parameters selected or specified by the user. These are summarized in Fig. 5. The saturated suction temperature for a multiplexed DX system is calculated as the design temperature at the display case minus the temperature gain in the fluid supply line:

$$T_{SST} = T_{\text{display case}} - \Delta T_{\text{supply line}} \qquad (2)$$

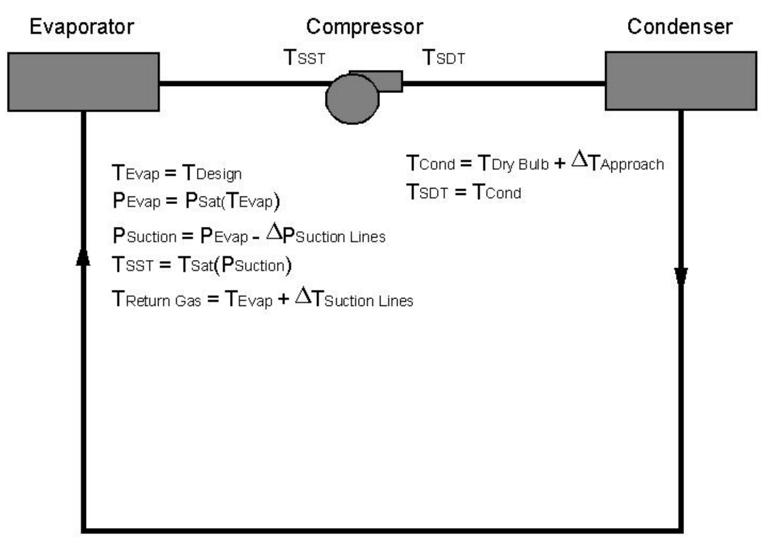


Figure 5. Compressor discharge temperature definitions.

The spreadsheets approximates performance of compressor racks with floating head pressure by using the outdoor ambient weather conditions and the type of condenser to compute the saturated discharge temperature. For a dry, air-cooled condenser, the saturated discharge temperature is computed as the outdoor dry bulb temperature plus the condenser approach temperature:

$$T_{SDT} = T_{\text{dry bulb}} + \Delta T_{\text{approach}} \qquad (3)$$

For an evaporative condenser, the saturated discharge temperature is computed as the outdoor wet bulb temperature plus the condenser approach temperature:

$$T_{SDT} = T_{wet\ bulb} + \Delta T_{approach} \tag{4}$$

The user has to specify approach temperatures for the medium and low temperature refrigeration systems (recommended values are supplied); medium temp systems are generally designed for a higher approach temperature.

Annual compressor power consumption is calculated by multiplying the ratio of the total refrigeration load and compressor EER at each outdoor temperature by the number of hours each year that the temperature occurs for the selected city:

$$P_{compressor} = \sum_{\text{dry bulb temp}} P_{\text{compressor, dry bulb temperature}} = \sum_{\text{dry bulb temp}} \frac{\sum_{cases} \dot{Q}_{loads}}{COP_{\text{dry bulb temp}}} \cdot h_{\text{dry bulb}} \tag{5}$$

The spreadsheet uses 5°F temperature bins for the outdoor dry bulb temperature.

Fan Power Calculations

Condenser fan power is calculated at each outdoor temperature by multiplying the total heat rejection (refrigeration load and compressor power) by the number of hours per year that temperature occurs and a fan requirement, W. The user is required to provide the fan requirement (W per 1000 Btu/h heat rejected, recommended values are provided) and a minimum condensing temperature for the low and medium temperature refrigeration. The condenser fans are assumed to operating continuously above the minimum condensing temperature, to cycle at air temperatures between the minimum condensing temperature and 30°F, and to operate 25% of the time below 30°F. Fan run time is assumed to vary linearly from 25% to 100% between 30° F and the minimum condensing temperature ($0 \le \delta_i \le 1$).

$$\begin{aligned} P_{fans} &= \sum_{\text{dry bulb temps}} P_{\text{fans, dry bulb temp}} \cdot \left(\delta_{\text{dry bulb temp}} \cdot h_{\text{dry bulb temp}}\right) \\ &= \sum_{\text{dry bulb temps}} \left(\sum_{cases} \dot{Q}_{loads} + 3.413 \cdot P_{\text{compressor, dry bulb temp}} \right) \cdot \frac{W_{\text{heat rejection}}}{1000} \cdot \left(\delta_{\text{dry bulb temp}} \cdot h_{\text{dry bulb temp}}\right) \\ &= \sum_{\text{dry bulb temps}} \left(\sum_{cases} \dot{Q}_{loads} + \frac{3.413 \cdot \sum_{cases} \dot{Q}_{loads}}{EER_{\text{dry bulb temp}}} \right) \cdot \frac{W_{\text{heat rejection}}}{1000} \cdot \left(\delta_{\text{dry bulb temp}} \cdot h_{\text{dry bulb temp}}\right) \end{aligned} \tag{6}$$

Secondary Loop Pump Power

Coolant mass flow rates (either refrigerant in multiplexed systems or the heat transfer fluid in secondary loop systems) are calculated from the load and the latent heat of vaporization or the specified heat exchanger ΔT. For secondary loop systems, the ΔT for each display case or walk-in is used to calculate the flow rate of the secondary fluid in the corresponding branch by Eq. 7 using the specific heat for the fluid at the design temperature:

$$\dot{m} = \frac{\dot{Q}_{load}}{C_p \Delta T} \quad (7)$$

The maximum mass flow rate in the secondary loop is thus the sum of the individual flow rates in Eq. 8:

$$\dot{m}_{max} = \sum \dot{m}_i = \sum \frac{\dot{Q}_{load}}{C_p \Delta T_i} \quad (8)$$

The pressure drop for the secondary loop is given by Eq. 9 where the maximum is evaluated for segments i from 1 to n:

$$\Delta P_{loop} = \max\left[\sum_{j=1}^{i}\left(\Delta P_{main,i} + \Delta P_{return,i}\right) + (\Delta P_{branch,i} + \Delta P_{load,i}\right] \quad (9)$$

Pumping power is computed using the lost head pressure that the pump is working against with fixed pump and motor efficiencies; eventually pump and motor characteristics should be added to the spreadsheet. The power output required by the ΔP is given by Eq. 10:

$$P_{out} = \frac{Q_v\,(\text{gallons/min}) \cdot \Delta P\,(\text{psi}) \cdot 2.31\ (\text{ft } H_2O/\text{psi})}{\left[\frac{7.48\text{ gallons}}{\text{ft}^3}\right] \cdot \left[\frac{60\text{ sec}}{\text{min}}\right] \cdot \left[\frac{1\text{ ft } H_2O}{62.32\text{ lb}_f\ /\text{ft}^2}\right] \cdot \left[550\text{ ft} - \text{lb}_f\ /\text{HP}\right]} \quad (10)$$

The shaft power, then simplifies to Eq. 11:

$$P_{shaft} = \frac{Q_v\ (\text{gpm}) \cdot \Delta P\ (\text{psi}) \cdot 2.31}{3960 \cdot \eta_{pump}} \quad (\text{HP}) \quad (11)$$

The electrical input power is given by Eq. 12:

$$P_{input} = \frac{P_{shaft} \cdot 0.746\ (\text{kW/HP})}{\eta_{motor}} \quad (\text{kW}) \quad (12)$$

Coolant Piping ΔP

Pressure drop calculations are performed for both the multiplexed direct expansion systems and secondary loop systems. Pressure drops in the multiplexed systems are used to determine compressor suction temperatures while those for secondary loop systems are used to compute the parasitic pumping power. In each instance the relatively simple piping connections illustrated in Fig. 6 are used. These consist of straight sections of pipe, T-couplings, 90° elbows, constrictions and expansions, and optional valves. Constant fluid properties are assumed for each refrigeration loop (e.g.

density, viscosity) while mass flow rates and velocities are determined from pipe diameters and refrigeration loads.

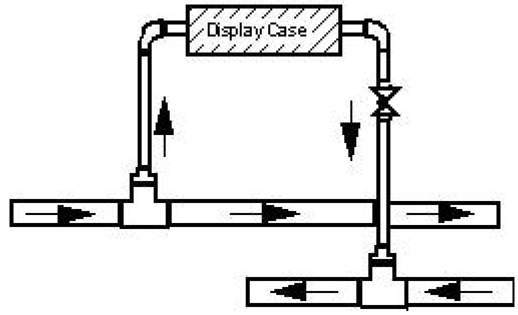


Figure 6. Assumed piping configurations.

Straight Piping

Pressure drops are computed in using the conventional correlations; ΔP is given by Eq. 13:

$$\Delta P = f \cdot \frac{L}{D} \cdot \frac{V^2}{2} \cdot \left[\frac{\rho}{144 \frac{in.^2}{ft^2} \cdot 32.174 \frac{lbm \cdot ft}{lbf \cdot sec^2}} \right] \quad (13)$$

The friction factor f is determined from the Reynolds number and the relative roughness of the pipe, e/D. The roughness, e, is a property of the pipe material and is in the range of 0.0000005 inches for drawn tubing and 0.00015 for commercial steel pipe. The pipe diameter, D, is also specified in inches. The Reynolds number is computed by Eq. 14:

$$Re = \frac{\rho \cdot \overline{V} \cdot D}{\mu} \quad (14)$$

where the density (ρ) and dynamic viscosity (μ) are computed using curve fits as described on page 11.

The spreadsheet uses tabulated data for friction factors for the appropriate flow regime (i.e. laminar, transitional, and turbulent). The table covers the range in Reynolds numbers from 100 to 75,000 and relative roughness from 10^{-5} to 10^{-2}. Linear interpolation is used to compute intermediate values.

Elbows

Elbow pressure drops are computed using fits to published data for the equivalent length of straight piping as functions of fluid velocity and pipe diameter (ASHRAE 2001a). These data were fit with curves of the form shown in Eq. 15 so the equivalent length could be computed from the velocity. The pressure drop for each elbow is then computed using Eq. 13.

$$L_E = a_0 + a_1 \cdot \ln(V) \quad (15)$$

Contractions and Expansions

Pressure drops due to expansions and contractions of pipes are computed using a table of coefficients for head loss as functions of the ratio of upstream and downstream pipe diameters (Roberson 1975). The lost head is given by Eq. 16:

$$h_L = K_C \cdot \frac{V_2^2}{2 \cdot g} \text{ for contractions and } h_L = K_E \frac{V_1^2}{2 \cdot g} \text{ for expansins} \qquad (16)$$

where V_1 and V_2 are the upstream and downstream fluid velocities and the coefficients K_C and K_E are taken from Table 1. The spreadsheet computes the ratio of upstream and downstream pipe diameters and uses a table lookup to find the appropriate loss coefficient (Excel does not interpolate between table entries, this lookup is performed returning a slightly higher coefficient and overestimating the ΔP).

Table 1. Loss Coefficients for Expansions and Contractions

D_2/D_1	K_C	D_1/D_2	K_E
0.0	0.50	0.0	1.00
0.1	0.49	0.1	0.98
0.2	0.48	0.2	0.94
0.4	0.44	0.4	0.71
0.6	0.32	0.6	0.41
0.7	0.23	0.7	0.22
0.8	0.15	0.8	0.13
0.9	0.06	0.9	0.04

The pressure drop for contractions or expansions is thus computed with Eq.17:

$$\Delta P_{psi} = K_L \cdot \frac{V_{max}^2}{2} \cdot \left[\frac{\rho}{144 \cdot 32.174} \right] \qquad (17)$$

T Fittings

T-fittings can occur in two different configurations: (1) when a single incoming stream splits into two outgoing streams and (2) when two incoming streams merge to form a single exit. Somewhat oddly, the pressure drop for both configurations have been correlated to an equivalent number of 90° elbows based on the percentage of the flow in the straight through branch (in the exit of splitting flows or the inlet of merging flows). Graphical data (ASHRAE 2001b) were fit to generate correlations for use in the spreadsheet. The number of elbow equivalents for a T fitting with one inlet and two outlets is computed with Eq. 18 where "x" is the percentage of the total fluid leaving from the straight-through branch.

$$N_{Elbows} = 0.0538 \cdot x^{-3.4886} \qquad (18)$$

Calculations for a T with two inlets and a single outlet are given by Eq. 19 where "x" is the percentage of the flow that enters in the straight-through branch of the T.

$$N_{Elbows} = 0.7468 \cdot x^{-1.8503} \qquad (19)$$

The pressure drop from the T-fitting is then given by Eqs. 20 and 21:

$$\Delta P_{elbows} = F \cdot \frac{L_E}{D} \cdot \frac{V^2}{2} \cdot \left[\frac{\rho}{144 \frac{in.^2}{ft^2} \cdot 32.174 \frac{lbm \cdot ft}{lbf \cdot sec^2}} \right] \tag{20}$$

$$\Delta P_{psi} = N_{Elbows} \cdot K_L \cdot \frac{V_{max}^2}{2} \cdot \left[\frac{\rho}{144 \cdot 32.174} \right] \tag{21}$$

Valves

The pressure drop for open gate and globe valves is also expressed in terms of 90° elbow equivalents. Table 2 contains the number of elbow equivalents for iron and copper gate and globe valves in the open position (ASHRAE 2001c). The spreadsheet incorporates the elbow equivalents into tables of coefficients so the equivalent length and ΔP can be computed using Eq. 20 and 21. Iron and steel are treated the same for these purposes.

Table 2. Valve ΔP Elbow Equivalents

Valve Type	Iron	Copper
Open Gate	0.5	0.7
Open Globe	12.0	17.0

Worksheet Data Entry

There is a significant amount of data that needs to be entered into the spreadsheet to describe the refrigeration system, and needless to say it is important that numbers be entered into the proper locations and that the contents of other locations, such as formulas, not be changed inadvertently. A user interface was written in VBA (Visual Basic for Applications) to assist in data entry to ensure that numbers and selections are entered into the correct cells of the spreadsheet. The interface uses a series of “dialog boxes” with common data entry features such as “drop down lists,” “text boxes,” and “radio buttons” where the user can enter selections. As a matter of nomenclature, individual pages of information organized into rows and columns are referred to as “worksheets.” Spreadsheets can contain many different worksheets, with each worksheet identified by a “tab” at the bottom of the window for the application. The Supermarket Refrigeration System Model consists of a main spreadsheet, called the Central Manager, and templates for multiplexed direct expansion spreadsheets and secondary loop spreadsheets. The Central Manager can be used to read the template spreadsheets, or previously edited versions of the templates, and display saved results. The templates contain the appropriate user interface to enter data, edit information, and save the resulting spreadsheets for future use.

Opening Screen

When the Central Manager spreadsheet is opened in Excel the initial screen display is shown in Fig. 7. This worksheet only contains some descriptive information and a “button” to initiate the user interface. The tabs along the bottom of the window identify additional worksheets containing charts and tables of saved results, coefficients for curve fits, binned weather data, and other information that is common to the multiplexed and secondary loop calculations. The user does not normally need to examine these worksheets, except possibly to recompute a set of coefficients for fits to the compressor maps. The user interface is initiated by moving the computer cursor over the “Start” button and clicking the left mouse button.

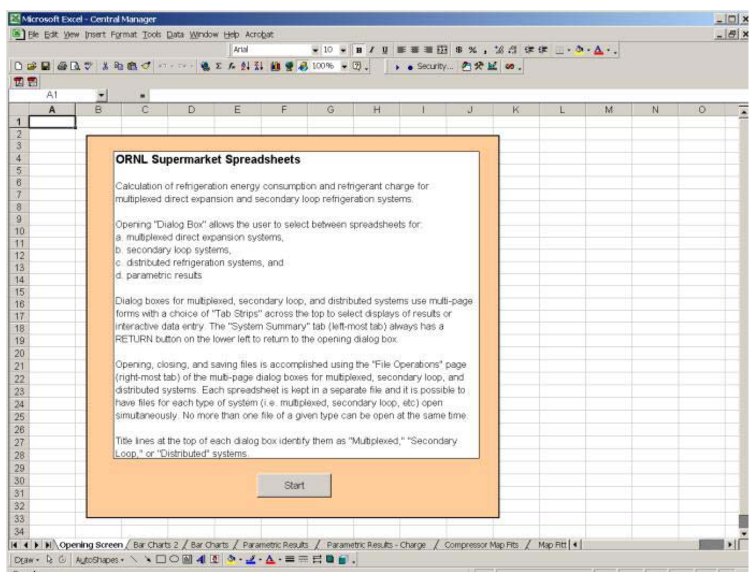


Figure 7. Opening spreadsheet screen.

Main Dialog Box

The Main Dialog Box illustrated in Fig. 8 is used to open spreadsheets for either multiplexed direct expansion or secondary loop systems (the option for distributed systems has not been implemented). It is possible to work with one multiplexed spreadsheet and one secondary loop spreadsheet simultaneously, but not to use more than one of either type system at the same time. The radio buttons on the right of the form allow the user to shrink (or enlarge) the dialog box to either see the information behind it or to make the dialog box itself easier to read. Clicking on either the "Multiplexed DX Spreadsheet" or "Secondary Loop Spreadsheet" file operation buttons closes this dialog box and opens a multipage form containing input data and computed results. All of the entries on this second form will be empty until the user opens a spreadsheet of the appropriate type (i.e. either multiplexed or secondary loop) using the last page of the form. This page is shown in Fig. 9.

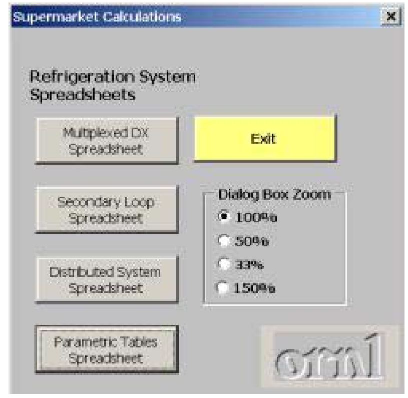


Figure 8. Main dialog box.

File Operations Page

Clicking on either the multiplexed DX or secondary loop buttons on the form shown in Fig. 9 opens a multipage form similar to the one shown in Fig. 10 (the forms for multiplexed DX and secondary loop systems are slightly different, the multiplexed DX form is used here for illustration purposes). The main data form generally opens with the System Summary page displayed and the user will need to click on the File Operations tab at the top of the form to see the page in Fig. 7. The series of buttons along the bottom left of the page are used to open the appropriate template file or previously saved spreadsheets, save the spreadsheet under the same or different names, and to close files. Information summarizing the data in the spreadsheet are displayed toward the top of this page if a file

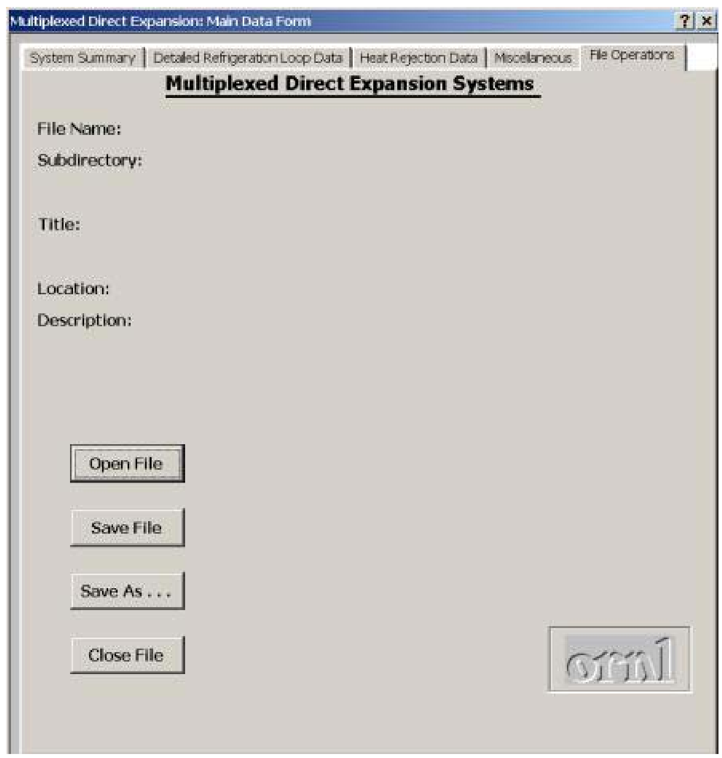


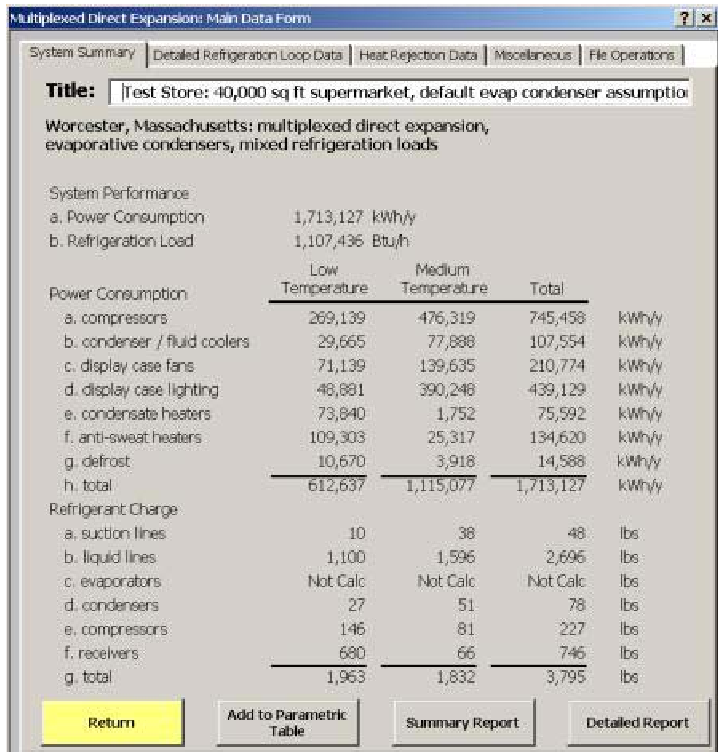
Figure 9. File operations page of the main data form.

is already open. The file operations are performed using standard dialog boxes that are part of the Windows operating system and probably do not require any explanation. The user selects folders and filenames using the computer mouse as is done in many application programs.

System Summary Page

When a spreadsheet is opened using the "Open File" button information from the cells on the spreadsheet are read and displayed on the multi-page main data form. After the form is fully populated with data from the spreadsheet the System Summary page is brought to the foreground. Figure 10 shows the System Summary page that appears when the file "Multiplexed Direct Expansion Spreadsheet.XLS" is opened.

At the very top of this page there is a text box where the user can enter a descriptive title for the dataset; this can be lengthy if desired and extend beyond the right side of the form. Data from the spreadsheet is displayed below the title box. These include a computer generated summary including the name of the city used, whether it is a multiplexed or secondary loop system, what type of condensers are used, and whether the refrigeration loads are set at the "high," "low," or "average" settings for each type of display case or if a mix of the three settings is used. This is followed by tables of power consumption and refrigerant charge for the medium and low temperature systems (totals for all loops used). Finally, at the bottom of the page there are buttons to:



Multiplexed Direct Expansion: Main Data Form

System Summary | Detailed Refrigeration Loop Data | Heat Rejection Data | Miscellaneous | File Operations

Title: Test Store: 40,000 sq ft supermarket, default evap condenser assumptio

Worcester, Massachusetts: multiplexed direct expansion, evaporative condensers, mixed refrigeration loads

System Performance
a. Power Consumption 1,713,127 kWh/y
b. Refrigeration Load 1,107,436 Btu/h

Power Consumption	Low Temperature	Medium Temperature	Total	
a. compressors	269,139	476,319	745,458	kWh/y
b. condenser / fluid coolers	29,665	77,888	107,554	kWh/y
c. display case fans	71,139	139,635	210,774	kWh/y
d. display case lighting	48,881	390,248	439,129	kWh/y
e. condensate heaters	73,840	1,752	75,592	kWh/y
f. anti-sweat heaters	109,303	25,317	134,620	kWh/y
g. defrost	10,670	3,918	14,588	kWh/y
h. total	612,637	1,115,077	1,713,127	kWh/y
Refrigerant Charge				
a. suction lines	10	38	48	lbs
b. liquid lines	1,100	1,596	2,696	lbs
c. evaporators	Not Calc	Not Calc	Not Calc	lbs
d. condensers	27	51	78	lbs
e. compressors	146	81	227	lbs
f. receivers	680	66	746	lbs
g. total	1,963	1,832	3,795	lbs

Return | Add to Parametric Table | Summary Report | Detailed Report

Figure 10. System summary page of the main data form.

- return to the Central Manager Main Data Form,
- add these results to the parametric tables saved on a worksheet in the Central Manager,
- print a summary report of information in the spreadsheet, and
- print a detailed report of information in the spreadsheet.

Detailed Refrigeration Loop Page

The next tab along the top of the main data form is used to display detailed information about the refrigeration loops. Clicking on this tab displays a page similar to the one shown in Fig. 11. The radio buttons in the upper right corner of this form are used to change the display between the three low temperature loops and the three medium temperature loops. Data for one loop is displayed at a time; the loop ID is shown in the upper left of the page. A text box is used to enter the desired evaporating temperature for each display case or walk-in of a multiplexed system or the average coil temperature for a secondary loop system. The total refrigeration load for the loop, refrigerant or coolant pressure drop, and the necessary suction temperature for a multiplexed system are displayed below the design temperature. This is followed by an identification of the different refrigeration loads on the loop. Five "buttons" appear, three to set all of the loads on the loop to their high, average, or low values and two to initiate data entry forms for the refrigeration loads and the major refrigerant vessels.

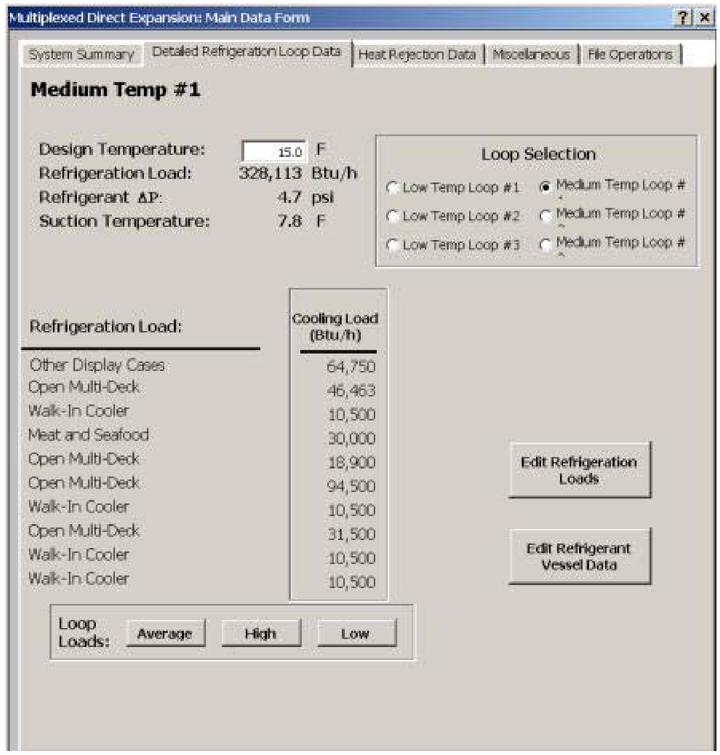


Figure 11. Detailed refrigeration loop data.

Clicking on the button for "Edit Refrigeration Loads" closes the multi-page form for the multiplexed or secondary loop systems and opens a form for detailed input of data for a single loop. An example of this form for multiplexed systems is shown in Fig. 12.

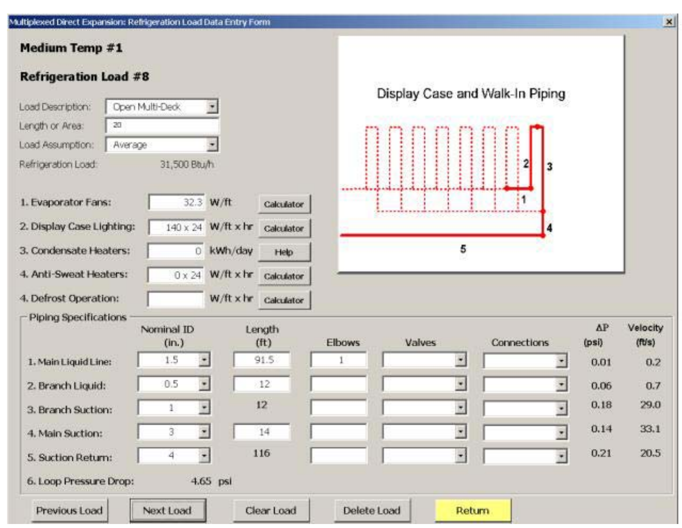


Figure 12. Refrigeration load data form.

The diagram in the upper right of the form identifies the different pipe segments discussed earlier with regard to Fig. 2. The buttons along the bottom of the form are used to move from one load on the loop to the next, up to a maximum of ten. Buttons are also provided to clear the data for a load, delete the load from the loop altogether, and to return to the multiplexed or secondary loop data form.

In the case of a secondary loop system, the user is required to provide both a ΔT and ΔP for the heat exchanger for each load on secondary loops. The refrigerant load form requires the user to select a type of display case (or walk-in) from a drop down list, specify the length of a lineup of cases or area of a walk-in, and the load assumption (i.e. low, average, or high). Power use at the case can be specified in a series of text boxes or computed using "calculators" initiated by the buttons in the middle of the form. An example of one of these is illustrated in Fig. 13 where a data entry "subform" is overlayed on the form in Fig. 12 by clicking the "Calculator" button beside item "2. Display Case Lighting." The text boxes and radio buttons on this subform can be filled in by the user to specify the display case lighting schedule, the length of flourescent bulbs used, the type of ballasts, and the number of regular and/or high efficiency bulbs. The calculator then computes the overall power requirement for lighting and prepares the text string specifying the lighting requirement for the lineup of display cases or walk-in (e.g. "140 x 24" to designate 140 W/ft and 24 h/d). Similar subforms can be used to specify power requirements for evaporator and air curtain fans, anti-sweat heaters, and case defrost energy. A simple "Help" box can be displayed that shows ranges of typical values for condensate heaters. Each of these subforms is closed by clicking on the "Done" button.

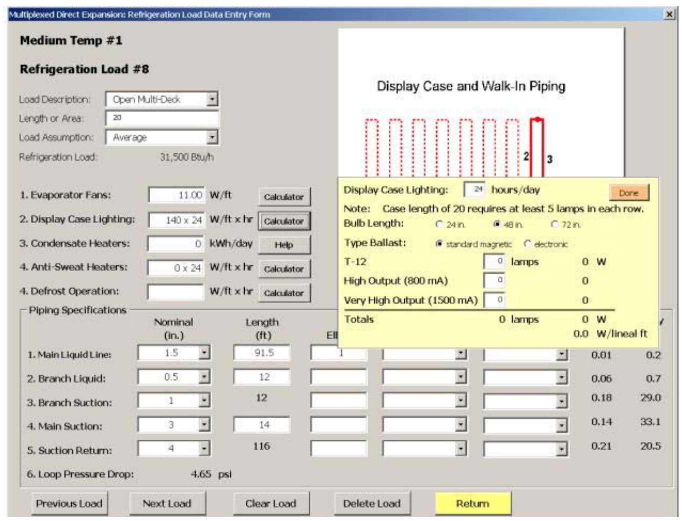


Figure 13. On screen "calculator" for display case lighting.

The bottom section of the form is used to enter or edit data for refrigerant or secondary loop piping. Pipe ID's can be selected for each section of pipe from a drop-down list; pipe lengths and the number of 90° elbows can be typed in, and types of valves and connections can be selected from drop down lists. As illustrated, the "main liquid line" for each load is considered as the section of piping delivering refrigerant or coolant from the preceding upstream load. The "branch liquid" and "branch suction" segments of multiplexed systems are assumed to be of equal length and extend from the shared refrigerant or coolant supply line to the load and back to the shared return lines. The "main suction" segment is the portion of the suction line between the load and the suction line of the next downstream load, and the "suction return" connects the final load main suction line to the machine room. Pressure drops are computed and displayed for each pipe segment as is the refrigerant velocity for multiplexed systems.

Heat Rejection Data

Energy consumption by condenser fans is determined using binned weather data for incidence of dry bulb temperature and corresponding mean wet bulb temperature (assumes floating head pressure with a specified minimum condensing temperature). Weather data is selected using the two drop down lists on the "Heat Rejection Data" page of the main data form (see Fig. 14). An example of the binned temperature data is shown in Fig. 15.

The type of condensers, either air-cooled or evaporative, is selected using the radio buttons on this page (it is assumed that there is a separate condenser for the low and medium temperature compressors and that both are of the same type). Other data inputs on this page include:

- minimum condensing temperature,
- condenser approach temperature,
- fan power requirement (watts-hours per 1000 Btu rejected), and
- heat loss to the ambient in the piping from the machine room to the condensers.

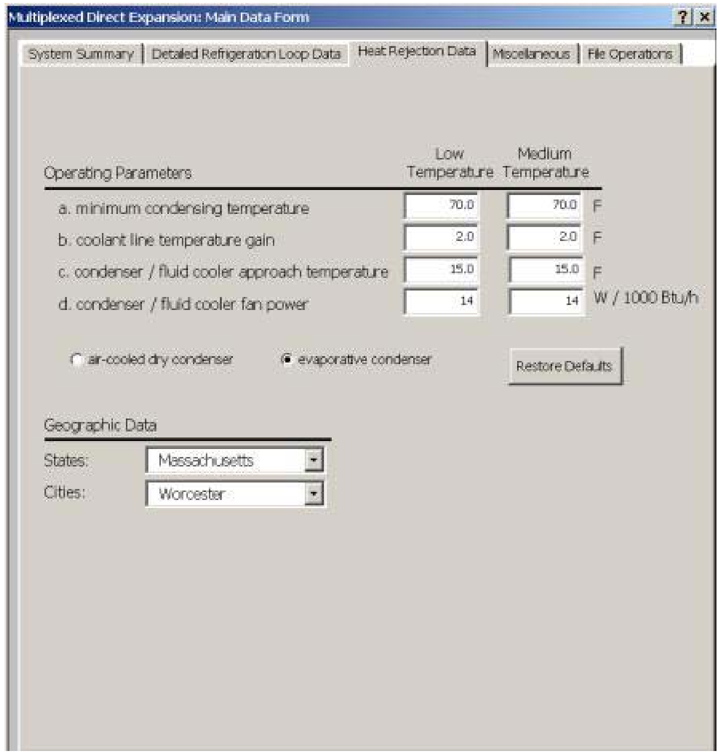


Figure 14. Heat rejection data page of the main data form.

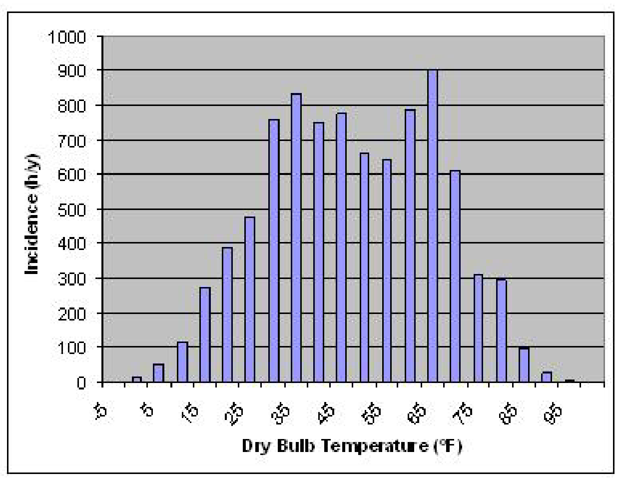


Figure 15. Temperature incidence for Worcester, Massachusetts.

Miscellaneous

Miscellaneous operations are performed through the final page of the main data form. An example is shown in Fig. 16. This page includes a collection of buttons on the left side that allow the user to select which worksheet is displayed behind the user interface dialog boxes. The text box and scroll buttons at the lower left of the form allow the user to zoom in or out to display more or less of the worksheet. Scroll buttons and "sliders" are displayed in the lower right corner of the form to shift the displayed worksheet left or right and up or down. There are also radio buttons to make the dialog box larger or smaller to either make it easier to read or to make more of the worksheet visible.

Parametric Results

As mentioned previously, results can be stored as part of the Central Manager spreadsheet so that alternative configurations or scenarios can be compared. They can be viewed, either in tabular form or as bar charts, by using the "Return" button on the multiplexed or secondary loop data form. This returns control to the Main Dialog Box illustrated in Fig. 9; clicking the "Parametric Results Spreadsheet" button opens a data form that displays results from up to ten different scenarios or cases. Examples of the four pages in the Parametric Results data form are shown in Figs. 17-20.

This data form also has multiple pages to display tables of

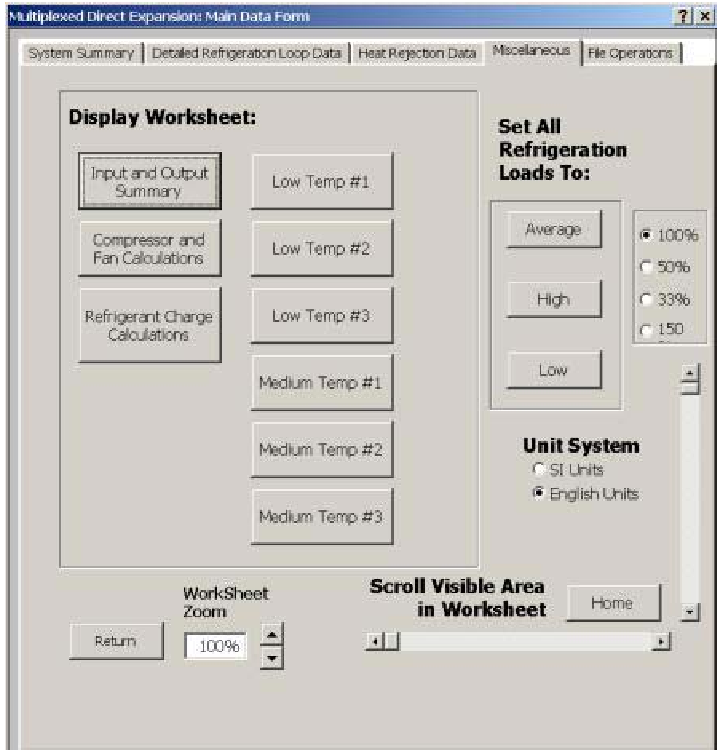
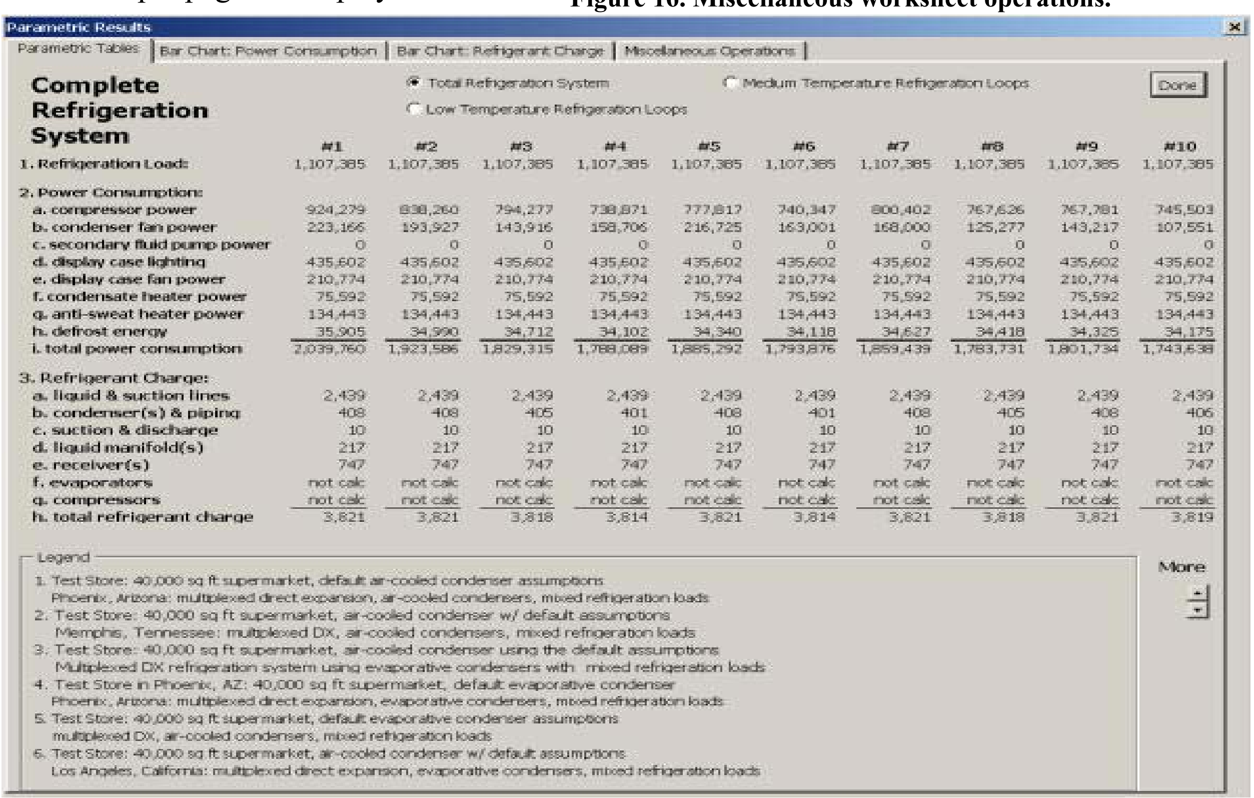


Figure 16. Miscellaneous worksheet operations.



Complete Refrigeration System	#1	#2	#3	#4	#5	#6	#7	#8	#9	#10
1. Refrigeration Load:	1,107,385	1,107,385	1,107,385	1,107,385	1,107,385	1,107,385	1,107,385	1,107,385	1,107,385	1,107,385
2. Power Consumption:										
a. compressor power	924,279	838,260	794,277	738,871	777,817	740,347	800,402	767,626	767,781	745,503
b. condenser fan power	223,166	193,927	143,916	158,706	216,725	163,001	168,000	125,277	143,217	107,551
c. secondary fluid pump power	0	0	0	0	0	0	0	0	0	0
d. display case lighting	435,602	435,602	435,602	435,602	435,602	435,602	435,602	435,602	435,602	435,602
e. display case fan power	210,774	210,774	210,774	210,774	210,774	210,774	210,774	210,774	210,774	210,774
f. condensate heater power	75,592	75,592	75,592	75,592	75,592	75,592	75,592	75,592	75,592	75,592
g. anti-sweat heater power	134,443	134,443	134,443	134,443	134,443	134,443	134,443	134,443	134,443	134,443
h. defrost energy	35,905	34,990	34,712	34,102	34,340	34,118	34,627	34,418	34,325	34,175
i. total power consumption	2,039,760	1,923,586	1,829,315	1,788,089	1,885,292	1,793,876	1,859,439	1,783,731	1,801,734	1,743,638
3. Refrigerant Charge:										
a. liquid & suction lines	2,439	2,439	2,439	2,439	2,439	2,439	2,439	2,439	2,439	2,439
b. condenser(s) & piping	408	408	405	401	408	401	408	405	408	406
c. suction & discharge	10	10	10	10	10	10	10	10	10	10
d. liquid manifold(s)	217	217	217	217	217	217	217	217	217	217
e. receiver(s)	747	747	747	747	747	747	747	747	747	747
f. evaporators	not calc	not calc	not calc	not calc	not calc	not calc	not calc	not calc	not calc	not calc
g. compressors	not calc	not calc	not calc	not calc	not calc	not calc	not calc	not calc	not calc	not calc
h. total refrigerant charge	3,821	3,821	3,818	3,814	3,821	3,814	3,821	3,818	3,821	3,819

Legend

1. Test Store: 40,000 sq ft supermarket, default air-cooled condenser assumptions
Phoenix, Arizona: multiplexed direct expansion, air-cooled condensers, mixed refrigeration loads
2. Test Store: 40,000 sq ft supermarket, air-cooled condenser w/ default assumptions
Memphis, Tennessee: multiplexed DX, air-cooled condensers, mixed refrigeration loads
3. Test Store: 40,000 sq ft supermarket, air-cooled condenser using the default assumptions
Multiplexed DX refrigeration system using evaporative condensers with mixed refrigeration loads
4. Test Store in Phoenix, AZ: 40,000 sq ft supermarket, default evaporative condenser
Phoenix, Arizona: multiplexed direct expansion, evaporative condensers, mixed refrigeration loads
5. Test Store: 40,000 sq ft supermarket, default evaporative condenser assumptions
multiplexed DX, air-cooled condensers, mixed refrigeration loads
6. Test Store: 40,000 sq ft supermarket, air-cooled condenser w/ default assumptions
Los Angeles, California: multiplexed direct expansion, evaporative condensers, mixed refrigeration loads

Figure 17. Parametric table of results.

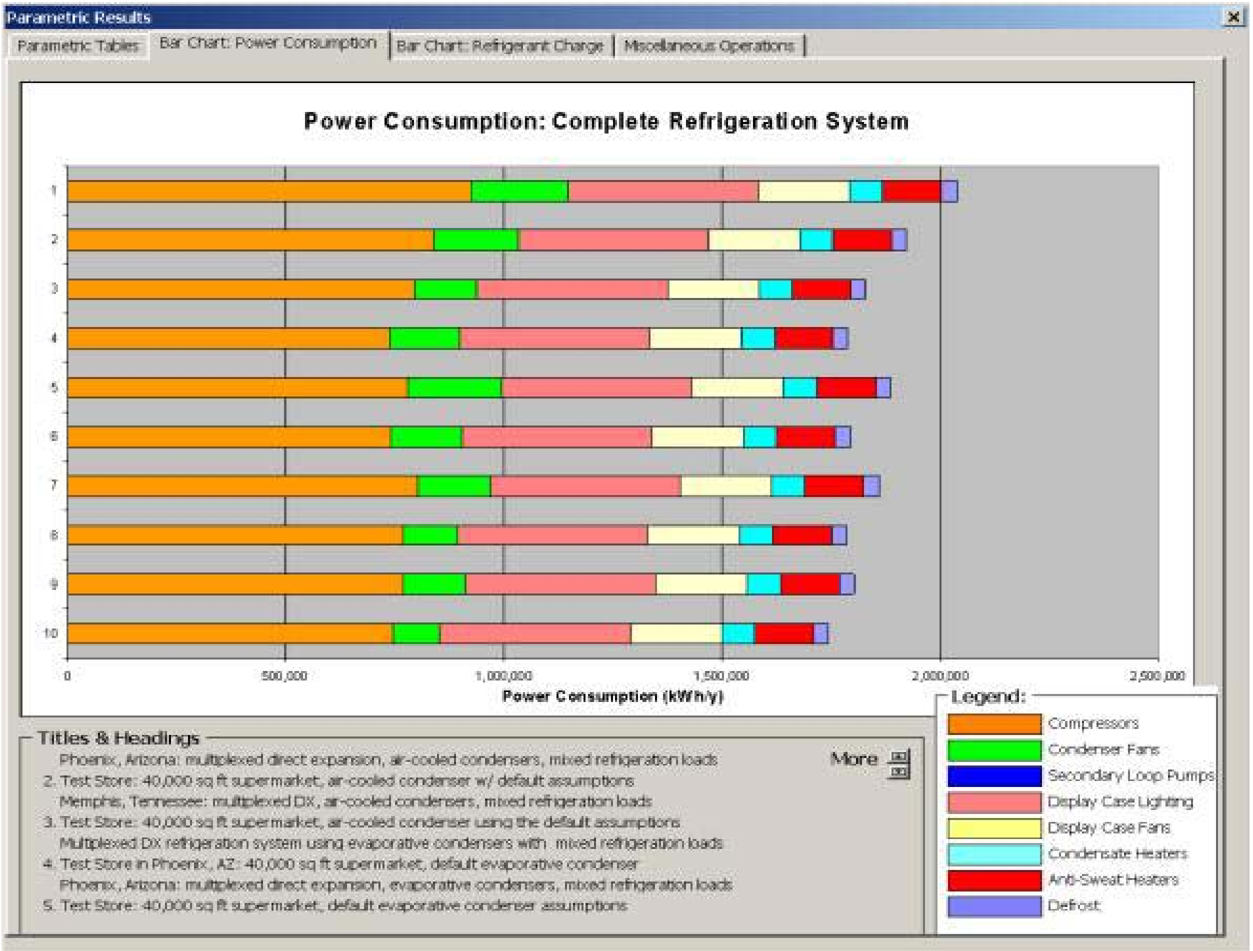


Figure 18. Bar chart of parametric results; power consumption.

power consumption and refrigerant inventories, bar charts for power and charge, and operations to rearrange entries in the results database, delete individual entries, or clear the database altogether. Titles and descriptions can be edited by double-clicking on the line of text. Each case in the parametric results is identified with a number that corresponds to the title and description listed below the table of results or chart. There is insufficient space on the form to display all of these entries simultaneously, so scroll buttons are provided to move the list up and down.

Radio buttons at the top of the tabular display of results allow the user to select between the results for the entire refrigeration system, the low temperature loops, or the medium temperature loops. This selection carries over to the two pages containing bar charts for power consumption and refrigerant charge.

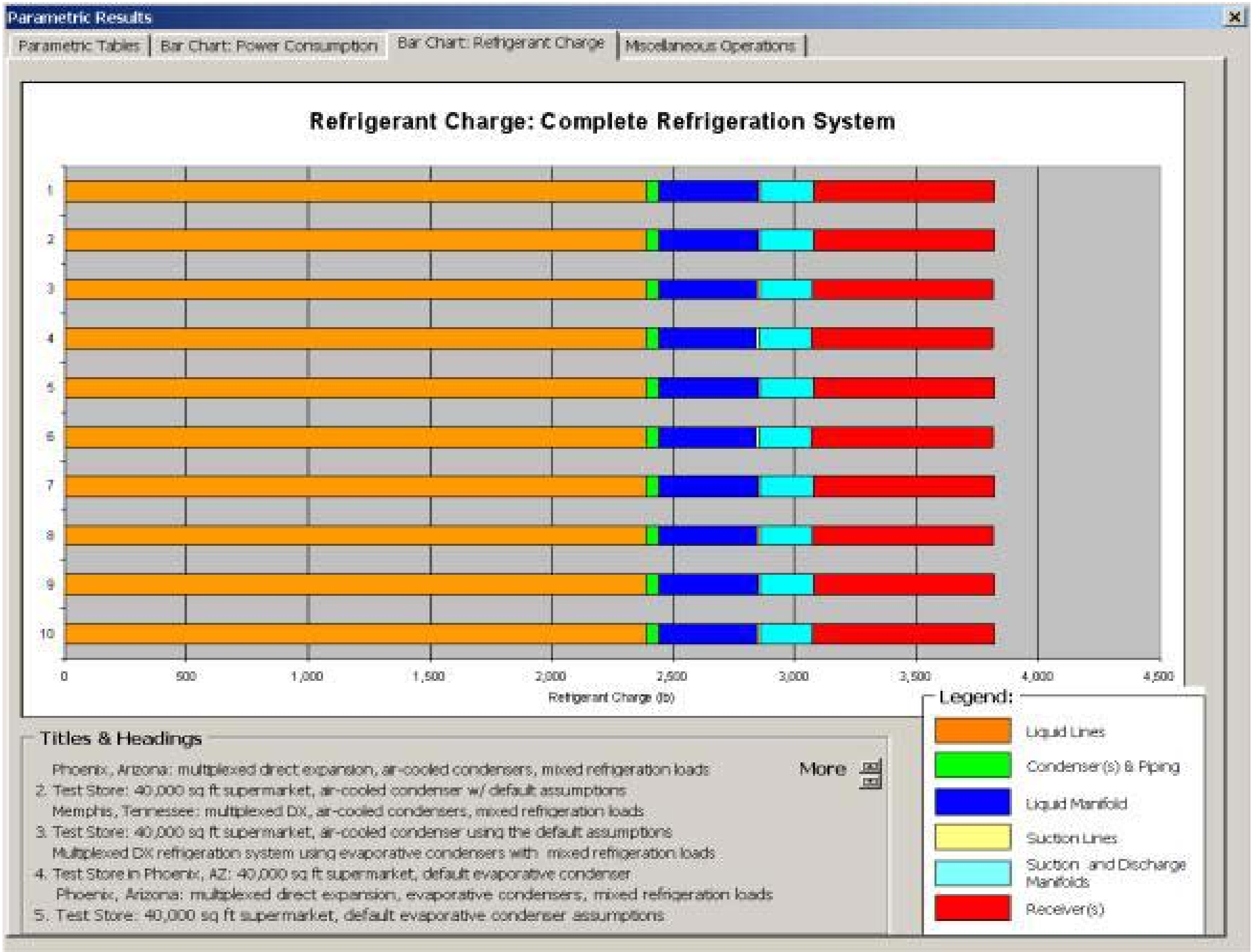


Figure 19. Bar chart of parametric results; refrigerant charge.

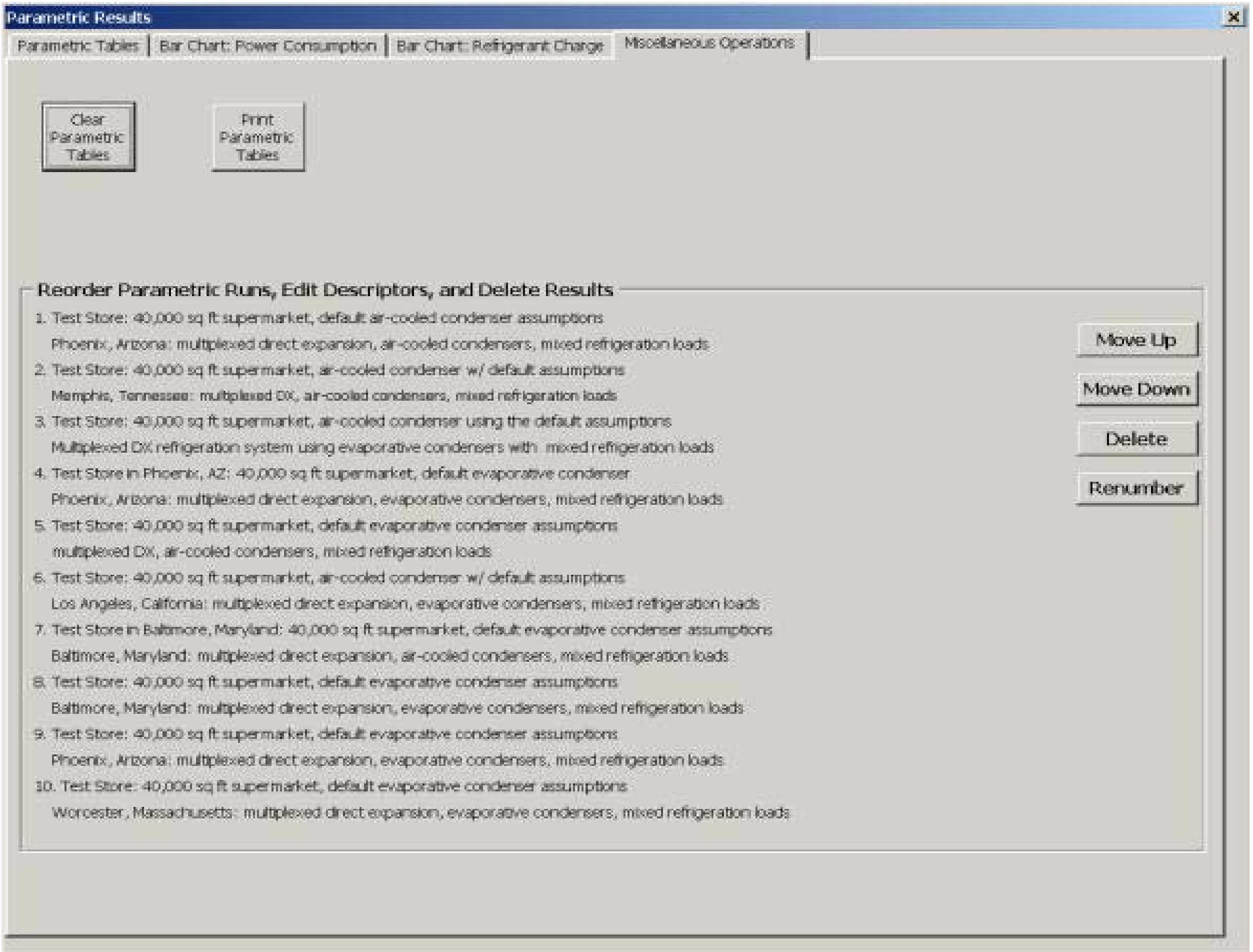


Figure 20. Miscellaneous operations for the parametric results.

Validation

Refrigeration Loads

Field data were obtained for a supermarket in Worcestor, Massachusetts. This store uses a conventional multiplexed, direct-expansion refrigeration system. There are essentially six compressor racks; two low temperature (main and satellite) and four medium temperature (two main and two satellite). The racks have been arbitrarily designated as A, B, and C with main and satellite compressors for each.

The field data included details on the types of display cases, their exit air temperatures, design loads, and average loads. Lengths of individual cases and case line-ups and floor area for walk-ins and prep rooms were not available. The first step in model validation was to match up display case types with manufacturer's literature with estimates of individual case lengths to agree with the known design loads. This process was done for each of the six compressor racks.

The design loads were found to be significantly different from the monthly or annual operating loads. Adjustments were made by assigning a single, average compressor run time for each rack so that the run time multiplied by the sum of the design case loads for each case on the rack approximated the measured average load on the rack.

The refrigeration loads were also found to have a reasonably strong dependence on outside temperature, with a broad scatter as illustrated in Fig. 21 for the two low temperature racks (main and satellite). The sheer volume of data was difficult to work with, so the refrigeration loads (as well as the measured compressor COPs and condenser fan powers) average values were computed for 5°F temperature bins. Figure 22 shows the results of "binning" the loads for the main low temperature compressor rack. The field data are illustrated with the vertical bars; the correlation for loads used in validating the model is show with the solid line. Initial efforts to validate the model did not include the temperature dependence of the loads (based on annual average load); these efforts did not provide satisfactory results due to under-estimating compressor power at the

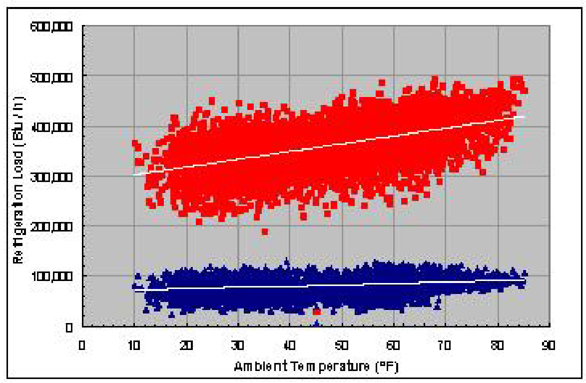


Figure 21. Field data for low temperature loads on main and satellite compressor racks.

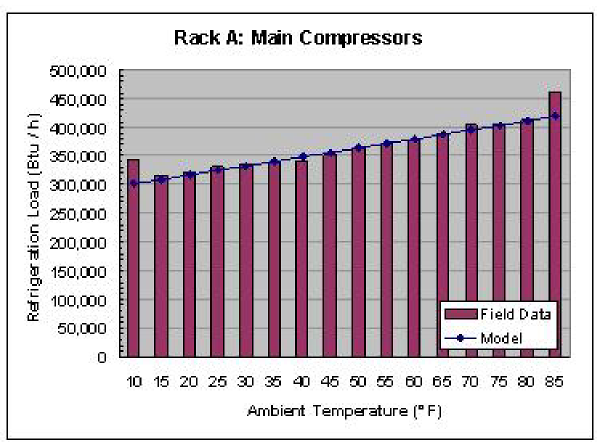


Figure 22. Refrigeration loads for main low temperature compressor rack; field data (bars)

higher ambient temperatures. Figure 23 shows the loads data and correlations for all six compressor racks.

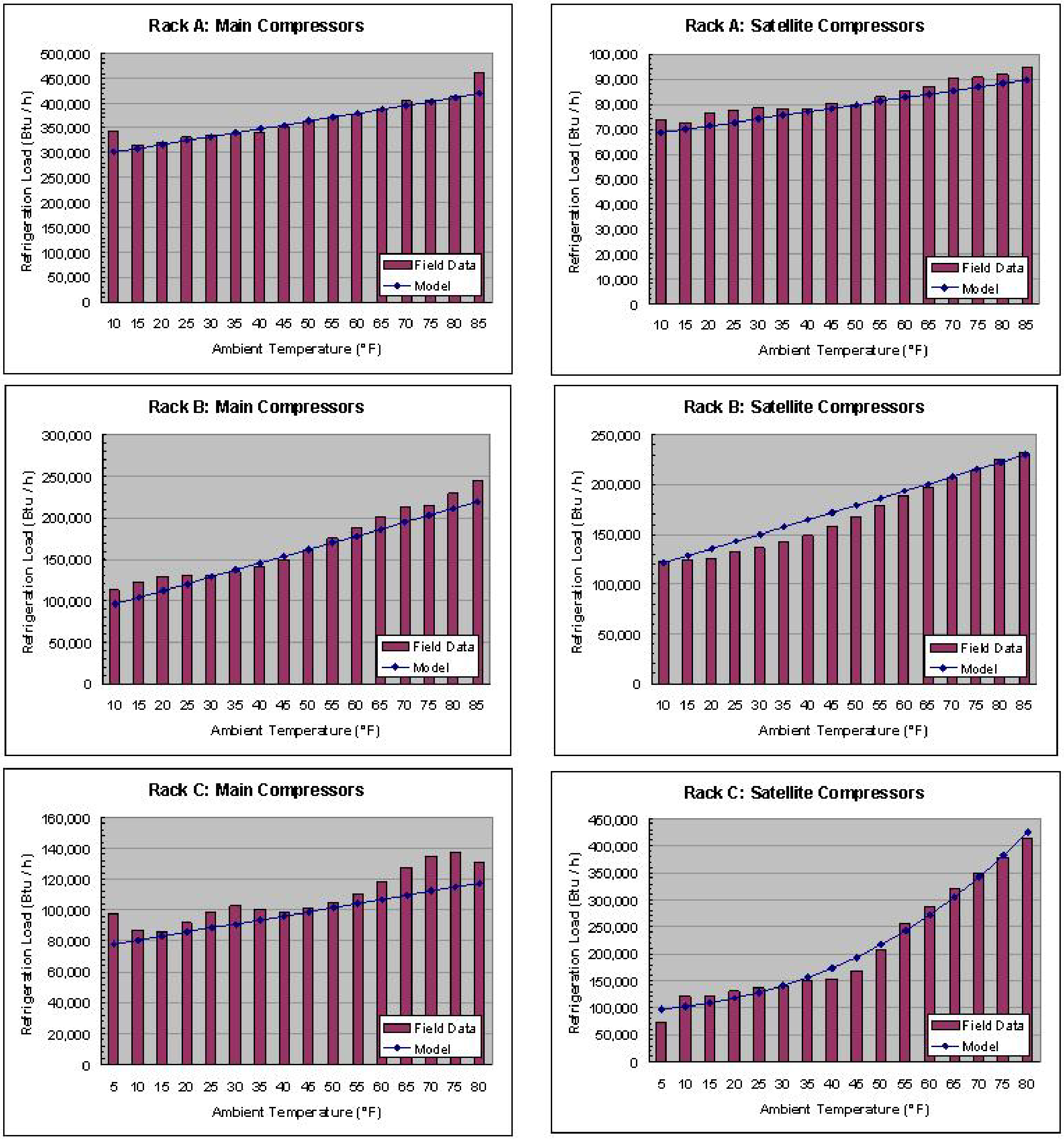


Figure 23. Binned refrigeration loads for main and satellite compressor racks.

There is a definite temperature dependence in the loads for each compressor rack, and the model will need to be adapted to incorporate this. A linear correlation is sufficient for five of the six racks, although there is strong non-linear behavior for the medium temperature satellite compressors, Rack C. These compressors provide both the mechanical subcooling for the low and medium temperature suction groups as well as the refrigeration for walk-ins and prep rooms located in service areas of the building with less temperature control than the sales areas of the building.

Compressor COPs

The field data includes hourly measurements of compressor power, refrigeration load, and compressor COP for all six compressor racks as well as maps for each of the compressors on each rack by way of ten coefficients for compressor power, capacity, and mass flow rate. The coefficients for power and capacity were used to generate data for COP as functions of saturated suction and discharge temperatures. These data were subsequently fit to generate ten coefficient correlations for COP as a function of saturation temperatures for each compressor; a single set of coefficients for each rack was formed by taking the averages of the corresponding coefficients for each compressor on the rack.

Figures 24 and 25 show the results of applying these derived correlations for compressor rack COP to the ambient temperatures and suction temperatures in the field data (saturated discharge temperature was determined from the ambient temperature as described below). In these figures, the measured hourly COPs have been grouped together into temperature bins and averaged in the same manner as the data for the refrigeration loads. The solid lines in these figures are the modeled COPs; there is a single discharge temperature associated with each ambient temperature for each rack and the suction temperatures were held constant at their design values.

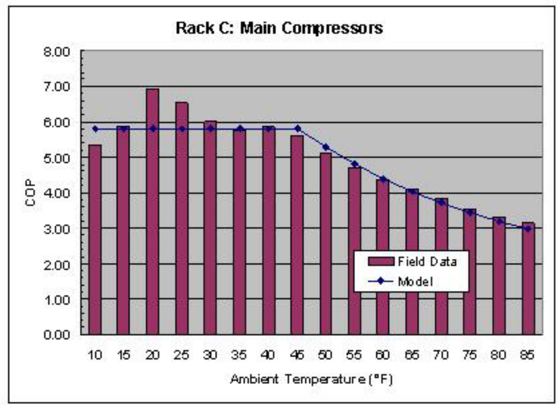


Figure 24. Measured and modelled compressor COPs for Rack C.

Both the bars and the solid lines for Racks A and B in Figs. 24 and 25 show the effects of floating the head pressure to a point corresponding to an ambient temperature of 50°F and then not allowing it to go any lower. The modeled results for the main and satellite rack C's imposed a minimum condensing temperature (as evidenced by the solid lines in Fig. 25), but the field data do not support this assumption nearly to the degree that they do for racks A and B.

Figure 25. Measured and modelled COPs for low and medium temperature compressor racks.

Condenser Fan Power

Condenser fan power calculations are performed using a simplified model; the fans are assumed to operate continuously above a specified temperature, to operate at a specified minimum run time (percent full load) below another specified temperature, and to cycle on and off between these two temperatures in a linear fashion. The fan power, thus, is assumed to be a constant (watts per 1000 Btu of heat rejected), times the refrigeration load, times the percent run time (and the number of hours for each temperature bin). The field data for the condensing unit serving the main and satellite low temperature compressor racks (A) can be modeled fairly well with this kind of algorithm as shown in Fig. 26. The condensing units for the B and C racks are not modeled as well with this approach. In fact, a fixed power requirement was set for the medium temperature condensing unit C instead of using the on/off temperature controls.

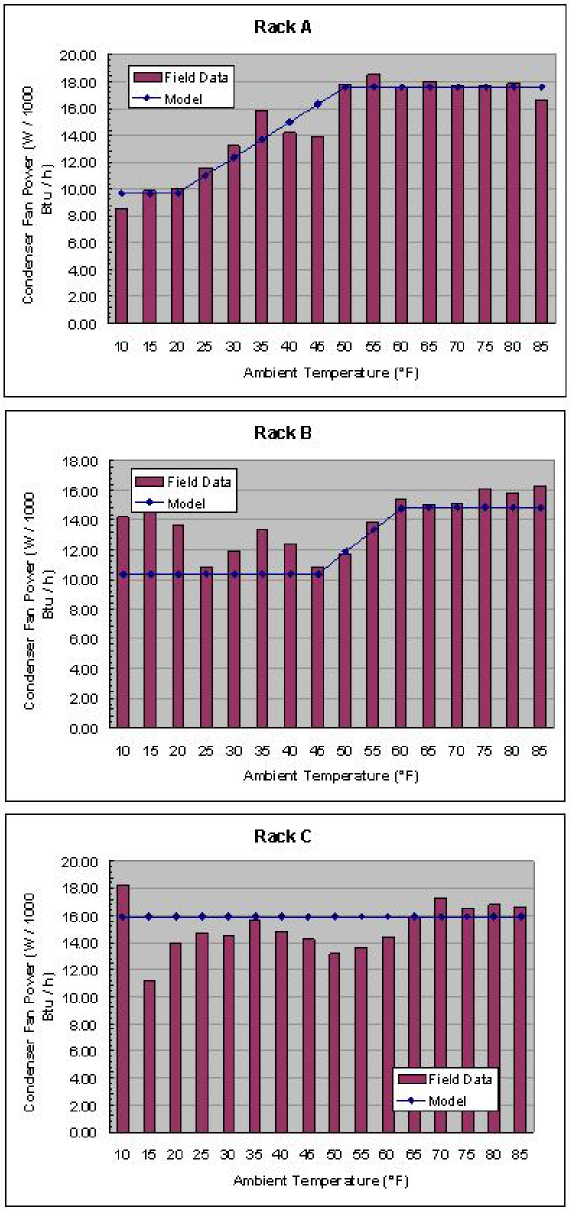


Figure 26. Binned condensing unit fan powers and model correlations for low and medium temperature compressor racks.

Temperature Profile

While the field data were available on an hourly basis, the spreadsheet model works using binned weather data. Ordinarily the number of hours of temperature occurrence for each temperature bin is determined when the user selects a city from the TMY2 weather data for one of 234 cities in the U.S. For validation purposes, the ambient temperatures recorded in the field data were used to compile bin data for the temperature occurrence corresponding to the weather in Worchester during the test period (April 28, 2001 to March 1, 2002). The temperature profile is shown in Fig. 27.

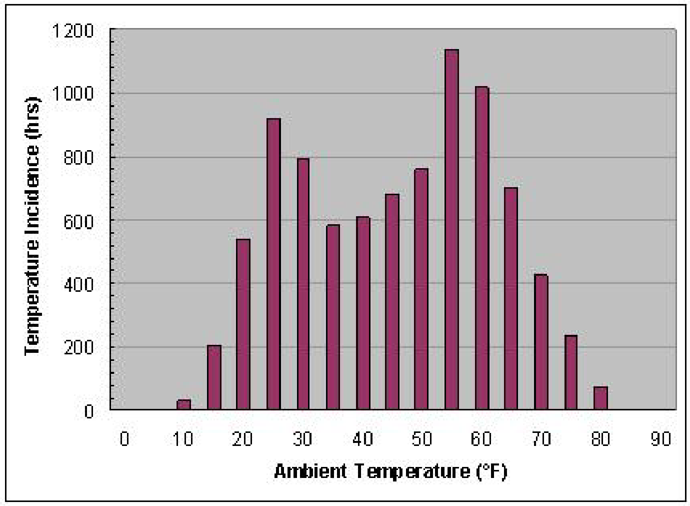


Figure 27. Ambient temperature profile for test site: April 28, 2001 to March 1, 2002.

Comparison of Measurements and Calculations

The field data measurements are summarized in Table 3. This information includes the number of hourly measurements for each of the three suction groups represented by compressor racks A, B, and C (main and satellite compressors in each case). The refrigeration load for each rack is presented (sum of the hourly loads for the entire period) and the compressor and condensing unit power consumption for the period.

Table 3. Measured Loads and Power Consumption

Compressor Rack	Rack A	Rack B	Rack C
Hours of Data	8,705	7,696	6,268
Refrigeration Load (10^6 Btu)			
a. main compressors	3,148	1,242	718
b. satellite compressors	715	1,265	1,358
c. total	3,863	2,507	2,076
Compressor Power (kWh)			
a. main compressors	335,455	69,799	43,956
b. satellite compressors	94,339	79,746	74,337
c. total compressor power	429,794	149,545	118,293
Condenser Fan Power (kWh)	84,872	40,390	34,812

Tables 4, 5, and 6 contain information for model validation. The process was simplified by scaling the field measured data for racks B and C so they represented the same number of hours as rack A; a simple ratio was used in each case to adjust the refrigeration loads and power consumption data. The results of these adjustments are shown in Table 2. Table 3 shows the calculated loads and power consumption for the period and Table 4 shows the percentage difference between the measured data and the calculations (negative values indicate that the calculated values are below the measurements, positive values indicate that the calculated values are too high).

Table 4. Scaled Field Data Measurements

Compressor Rack	Rack A	Rack B	Rack C
Hours of Data	8,705	8705	8705
Refrigeration Load (10^6 Btu)			
d. main compressors	3,148	1,405	997
e. satellite compressors	715	1,431	1,886
f. total	3,863	2,836	2,883
Compressor Power (kWh)			
d. main compressors	335,455	78,950	61,046
e. satellite compressors	94,339	90,201	103,239
f. total compressor power	429,794	169,151	164,285
Condenser Fan Power (kWh)	84,872	45,685	48,347

Table 5. Calculated Refrigeration Loads and Power Consumption

Compressor Rack	Rack A	Rack B	Rack C
Hours of Data	8,705	8705	8,705
Refrigeration Load (10^6 Btu)			
g. main compressors	3,108	1,344	864
h. satellite compressors	683	1,502	1,657
i. total	3,791	2,847	2,521
Compressor Power (kWh)			
g. main compressors	319,563	82,003	51,373
h. satellite compressors	92,980	104,639	102,862
i. total compressor power	412,543	186,642	154,235
Condenser Fan Power (kWh)	81,442	40,401	47,740

Table 6 Comparison of Measured and Modeled Load and Power

Compressor Rack	Rack A	Rack B	Rack C
Hours of Data	8,705	8,705	8,705
Refrigeration Load (10^6 Btu)			
j. main compressors	-1.3%	-4.3%	-13.3%
k. satellite compressors	-4.4%	5.0%	-12.1%
l. total	-1.9%	0.4%	-12.5%
Compressor Power (kWh)			
j. main compressors	-4.7%	3.9%	-15.8%
k. satellite compressors	-1.4%	16.0%	-0.4%
l. total compressor power	-4.0%	10.3%	-6.1%
Condenser Fan Power (kWh)	-4.0%	-11.6%	-1.3%

Table 6 shows reasonable agreement between the measurements and calculations for the low temperature refrigeration A main and satellite racks, poor agreement for the medium temperature satellite rack B, and main rack C. The overestimate of compressor power for Satellite rack B is the consequence of overestimating the loads (illustrated in Fig. 23) and underestimating the COP (see Fig. 25). There is an offsetting overestimate of the refrigeration load for the main compressors of Rack B and an underestimate of the COP, so that overall the calculated power consumption is reasonably close to the measured values.

The underestimates of load and COP for medium temperature Rack C main compressors result in a poor comparison of compressor power consumption. The poor fit between the data and condenser fan model for Rack B results in underestimating power consumption by nearly 12%.

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Appendix: Weather Data

The supermarket spreadsheets developed by ORNL use typical meteorological weather data from the National Renewables Energy Laboratory in Golden, Colorado. Data sets are available for 237 cities in the U.S. These data include hourly readings for wet and dry bulb temperatures as well as cloud cover, solar insolation, precipitation. The hourly weather data were used to compile the binned temperature data in worksheets "Dry Bulb Temperatures" and "Wet Bulb Temperatures." The binned data are used in the compressor power calculations in conjunction with the type of condenser (i.e. air-cooled or evaporative) and the approach temperature. Temperature distributions are included for the following cities:

Alabama
Birmingham
Huntsville
Mobile
Montgomery
Alaska Anchorage
Annette
Barrow
Bethel
Bettles
Big Delta
Cold Bay
Fairbanks
Gulkana
King Salmon
Kodiak
Kotzebue
McGrath
Nome
St Paul Island
Talkeetna
Yakutat
Arizona Flagstaff
Phoenix
Prescott
Tucson
Arkansas Fort Smith
Little Rock
California Arcata
Bakersfield
Dagget
Fresno
Long Beach
Los Angeles
Sacramento
San Diego
San Francisco
Santa Maria
Colorado Alamoso
Colorado Springs
Boulder/Denver
Eagle
Grand Junction
Pueblo
Connecticut Bridgeport
Hartford
Delaware
Wilmington
Florida Daytona Beach
Jacksonville
Key West
Miami
Tallahassee
Tampa
West Palm Beach
Georgia Athens
Atlanta
Augusta
Columbus
Macon
Savannah
Hawaii Hilo
Honolulu
Kahului
Lihue
Idaho Boise
Pocatello
Illinois Chicago
Moline
Peoria
Rockford
Springfield
Indiana Evansville
Fort Wayne
Indianapolis
South Bend
Iowa Des Moines
Mason City
Sioux City
Waterloo
Kansas Dodge City
Goodland
Topeka
Wichita
Kentucky Covington
Lexington
Louisville
Louisiana Baton
Rouge
Lake Charles
New Orleans
Shreveport
Maine Caribou
Portland
Maryland Baltimore
Massachusetts Boston
Worcester
Michigan Alpena
Detroit
Flint
Grand Rapids
Houghton
Lansing
Muskegon
Sault Ste Marie
Traverse City
Minnesota Duluth
International Falls
Minneapolis
Rochester
St Cloud
Mississippi Jackson
Meridian
MissouriColumbia
Kansas City
Springfield
St Louis
MontanaBillings
Cut Bank
Glasgow

Great Falls
Helena
Kalispell
Lewiston
Miles City
Missoula
Nebraska Grand Island
Norfolk
North Platte
Omaha
Scottsbluff
Nevada Elko
Ely
Las Vegas
Reno
Tonopah
Winnemucca
New Hampshire Concord
New Jersey Atlantic City
Newark
New Mexico
Albuquerque
Tucucari
New York Albany
Binghamton
Buffalo
Massena
New York City
Rochester
Syracuse
North Carolina Asheville
Cape Hatteras
Charlotte
Greensboro
Raleigh/Durham
Wilmington
North Dakota Bismarck
Fargo
Minot
Ohio Akron/Canton
Cleveland
Columbus
Dayton
Mansfield
Toledo
Youngstown
Oklahoma Oklahoma City
Tulsa
Oregon Astoria
Burns
Eugene
Medford
North Bend
Pendleton
Portland
Redmond
Salem
Pennsylvania Allentown
Bradford
Erie
Harrisburg
Philadelphia
Pittsburgh
Wilkes_Barre
Williamsport
Rhode Island Providence
South Carolina Charleston
Columbia
Greenville
South Dakota Huron
Pierre
Rapid City
Sioux Falls
Tennessee Bristol
Chattanooga
Knoxville
Memphis
Nashville
Texas Abilene
Amarillo
Austin
Brownsville
Corpus Christi
El Paso
Fort Worth
Houston
Lubbock
Lufkin
Midland
Port Arthur
San Angelo
San Antonio
Victoria
Waco
Wichita Falls
Utah Cedar City
Salt Lake City
VermontBurlington
Virginia Lynchburg
Norfolk
Richmond
Roanoke
Sterling
Washington Olympia
Quillayute
Seattle
Spokane
Yakima
West Virginia Charleston
Elkins
Huntington
Wisconsin Eau Claire
Green Bay
La Crosse
Madison
Milwaukee
Wyoming Casper
Cheyenne
Lander
Rock Springs
Sheridan

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US Report – Part C
Field Testing of an Advanced Low-Charge Supermarket Refrigeration System

Introduction

A field test was conducted to compare the performance of multiplex and distributed supermarket refrigeration systems. Two supermarkets in the vicinity of Worcester, Massachusetts were the sites utilized for the field test. One store was equipped with a multiplex refrigeration system that had 3 compressor racks and air-cooled condensers. At the second store, a low-refrigerant-charge distributed refrigeration system was installed that consisted of 10 compressor cabinets. Heat rejection for the compressor cabinets was accomplished through water-cooled condensers piped to a fluid loop that used dry fluid coolers. The second store also had water-source heat pumps for space heating and cooling that were piped into the fluid loops. Both sites were instrumented to determine energy consumption, refrigeration and heating, ventilating and air conditioning (HVAC) loads, and numerous system state points to characterize operation. Data were collected for approximately 18 months. A comparison of the performance of the two refrigeration systems was made based on the information gathered. The results showed that the multiplex refrigeration system used less energy and showed better EER than the distributed system. A TEWI comparison between the two refrigeration systems showed significant reductions in atmospheric CO_2 generation by the distributed system despite added energy use. Test results also showed that the water-source heat pumps produced energy cost savings for store space heating by recovery of the refrigeration reject heat.

1. Background

Present supermarket systems employ direct expansion of refrigerant in the display cases and walk-in coolers with remotely operated multiplexed compressor racks and condensers. Because of the large floor area of modern supermarkets, the amount of refrigerant needed for operation is very large, on the order of 3,000 – 5,000 lb. With growing concern over the environmental impact of refrigerant leakage from these systems, several new system configurations have been designed and constructed that require less refrigerant for their operation. Examples of such systems include:

- Distributed – Display cases and walk-in's are piped to compressors located in cabinets on the sales floor or around the perimeter of the store. Heat rejection may be done through the use of water-cooled condensers and a fluid loop or with air-cooled condensers on the store roof above each compressor cabinet.
- Secondary loop – Refrigeration to the display cases and walk-in's is provided by a secondary fluid loop that is refrigerated by a central chiller system. Heat rejection is done through air-cooled or evaporative condensers, or a fluid loop can be employed to minimize the refrigerant charge.
- Advanced self-contained – each display case or group of several cases has a condensing unit installed in the case. Heat rejection is done through the use of a fluid loop.

While each of these systems uses significantly less refrigerant than standard multiplex compressor systems, the energy consumption of these systems is also very important in determining their impact on the environment. A complete assessment of the environmental impact of a refrigeration system can be made through a TEWI (Total Equivalent Warming Impact) evaluation that takes into account both primary and secondary global warming caused by direct refrigerant leakage and electric generation, respectively [1].

The energy consumption of low-charge refrigeration systems can be greater than that of multiplex systems because of the methods used to limit their refrigerant charges. For distributed refrigeration, low noise-level scroll compressors must be employed in order to locate the compressor cabinets in the sales area. Scroll compressors generally have lower EER values than reciprocating compressors today. If a fluid loop is used for heat rejection, this will add an extra temperature difference that will increase the operating condensing temperature, and it will also result in adding pump energy consumption for fluid circulation in the loop. For the secondary loop system, the pumping energy associated with the secondary loops adds significantly to the total refrigeration energy.

System design methods and enhancements exist that help offset the added energy associated with low-charge operation. Both distributed and secondary loop systems can be close-coupled to their evaporator loads which helps maintain higher suction pressure and tends to lower the return gas temperature. Scroll compressors can be operated at very low head pressures, because they have no suction or discharge valves. Scroll compressors through mid-scroll vapor injection can employ refrigerant subcooling. For the secondary loop system, refrigerant subcooling can be obtained through the use of hot brine defrost where the brine used for defrost is heated by subcooling the liquid refrigerant. The use of evaporative heat rejection helps to lower the condenser temperature penalty seen with fluid loops.

An analytical investigation of low-charge refrigeration systems showed that with proper design and operation both distributed and secondary loop refrigeration systems could use less energy than a baseline air-cooled multiplex system. [2,3]. The estimated energy savings ranged from 6.3 - 12.4 % for distributed and 3.7 - 10.2 % for secondary loop refrigeration, respectively. A TEWI analysis of the distributed and secondary loop systems showed significant reductions (up to 60%) in CO_2 generation versus the baseline air-cooled multiplex refrigeration system.

Another energy saving method investigated was the use of water-source heat pumps in conjunction with fluid loops for heat rejection. In this approach, the heat of rejection from the refrigeration system can be reclaimed by the water-source heat pumps to provide space heating for the store. The advantage of this method over standard refrigeration heat reclaim is that the condensing temperature of the refrigeration system can be floated to minimum levels allowable without limiting heat recovery by the heat pump. A larger fraction of the reject heat can be utilized with the water-source heat pumps than can be recovered with conventional heat reclaim.

In order to verify these analysis results a field test of a baseline system and a low-charge refrigeration system coupled with water-source heat pumps for HVAC was undertaken in two supermarkets with the cooperation and assistance of Price Chopper Supermarkets (Golub Corporation, Schenectady, NY) and Massachusetts Electric. One test supermarket located in Marlborough, MA. was equipped with multiplex compressor racks with air-cooled condensers (baseline system). The store consisted of 52,000 ft^2 of floor space and was a new construction site that opened in 1997. The second test supermarket was located in Webster, MA and also had approximately 52,000 ft^2 of floor space. A distributed refrigeration system was installed along with 3 water-source heat pumps for store HVAC. Both stores were thoroughly instrumented and were monitored for approximately 2 years. During that time period energy and performance data were gathered for the refrigeration and HVAC systems at both sites.

2. Description of the Test Refrigeration Systems

Table 1 provides a listing of the refrigerated fixtures that were connected to the multiplex refrigeration system. Three racks are employed with each rack having two suction groups. Rack 1 is the low temperature refrigeration system that employs R-404A as the refrigerant. Racks 2 and 3 are used for medium temperature refrigeration and both racks use R-22 as the refrigerant. The high temperature suction group of Rack 3 supplies mechanical subcooling for both Racks 1 and 2. Mechanical subcooling is normally used on low temperature refrigeration only, making the subcooling for the medium temperature Rack 2 somewhat unique. Separate air-cooled condensers were used for each compressor rack. The low temperature condenser was sized with a design temperature difference of 10^{o}F while the medium temperature condensers were sized to operate with a 15^{o}F temperature difference. During low ambient temperature operation, fan cycling was used to maintain condensing temperature at approximately 70^{o}F. The design refrigeration loads were 422,825 and 899,953 Btu/h for the low and medium temperature refrigeration, respectively. The design subcooling load for Racks 1 and 2 was 361,670 Btu/h.

The details of the distributed refrigeration system are given in Table 2. The refrigeration load is divided among 10 compressor cabinets (A – J). Cabinets A, B, C, and D are dedicated to low temperature refrigeration, while Cabinets E, G, and I are medium temperature refrigeration. Cabinets F, H, and J each have two suction groups, one for low temperature refrigeration and the other for medium temperature. The low temperature cabinets A, B, C, and, D were equipped with subcooling by mid-scroll vapor injection. The design refrigeration loads for the distributed system were 404,845 and 1,010,936 Btu/h for low and medium temperature refrigeration, respectively.

Fluid-cooled condensers located at each compressor cabinet accomplished heat rejection for the distributed refrigeration system. Two fluid loops using a propylene glycol/water solution were connected to these condensers and heat was rejected by fluid coolers located on the roof of the supermarket. The fluid coolers were dry units, rather than evaporative. Price Chopper made the choice of dry coolers, because the store is located a considerable distance from their headquarters and the added maintenance of evaporative units could not be addressed by their in-house maintenance people. Price Chopper felt that improperly maintained evaporative fluid coolers could result in refrigeration system reliability issues for the store.

The use of dry fluid coolers was detrimental to the performance of the distributed system for several reasons. The added temperature difference associated with the operation of the fluid loops for heat rejection could not be offset by the dry coolers, since the coolers reject heat to the ambient dry-bulb temperature. This factor was particularly evident during summer operation when the condensing temperatures of the distributed cabinets were considerably higher than those of the multiplex racks. The fan power needed for the dry fluid coolers was also considerably larger than that associated with the multiplex air-cooled condensers, primarily because of the inclusion of the water-source heat pumps in the fluid loops. The fluid coolers had to be sized to provide heat rejection for the refrigeration and the water-source heat pumps when operating in space cooling mode.

Water-source heat pump operation also impacted the sizing of the fluid loop pumps since added fluid flow was needed for space cooling heat rejection. Each fluid loop had a circulating pump. The power of each pump was measured at 6.5 kW. Pumping power for these loops was higher than expected because the loops were filled with a 50/50 mixture of propylene glycol and water for freeze protection. The concentration of glycol was higher than needed for this particular location, but the concentration was chosen to insure reliable operation of the fluid loops during winter.

Three water-source heat pumps were installed at the distributed system test store. The heat pumps were integrated into the fluid loops so that the refrigeration heat of rejection could be used by the heat pumps for store space heating and the fluid loops could be used for heat rejection from the heat pumps during store space cooling. Table 3 provides descriptions of the water-source heat pump units that were installed. The 2 large heat pumps were constructed to utilize a dual-path approach for space cooling operation. The heat pump was equipped with 2 sets of compressors and evaporator coils; one set of compressors and a coil were used for cooling return air from the store, while the other set of compressors and coil provided cooling and dehumidification to outside ventilation air. The cooling capacity of the heat pump was divided such that the outside air system provided approximately 25 tons of cooling while the return air system provided approximately 20 tons. In space heating mode, only the return air system was used. The third heat pump was considerably smaller than the 2 dual-path units and was operated as a single-path system where outside air was mixed with return air prior to passing through the heat pump coil.

Operating data gathered from the heat pumps showed that they were very effective in recovering the rejected heat from the refrigeration system for store space heating. The dual-path approach for space cooling was found not to be appropriate for this particular site for space cooling. The dehumidification requirements for the site were much less than the dual-path system was designed for, and it was found that both the return and outside air systems had to be run in order to meet the total space cooling load of the supermarket. The outside air system required considerably more energy for cooling because of the higher temperature and humidity of the outside air. Energy consumption for space cooling was found to be very high for this particular site. A conventional single-path arrangement for all 3 heat pumps would have been more appropriate.

Space heating operation of the water-source heat pumps also affected the minimum operating temperature of the fluid loops, which, in turn, impacted the minimum condensing temperature that could be achieved by the distributed refrigeration system. Originally, it was decided that the minimum condensing temperature for the distributed system should be set at 50^{o}F, which meant that the fluid loops had to be operated at a minimum temperature of 40^{o}F. During the first winter of operation it was found that the heat pumps required a fluid temperature of 50^{o}F in order to meet the largest space heating load at lowest ambient temperature. At the largest space heating load, the heat pumps needed both heat recovery from the fluid loop and auxiliary heating in order to meet this load. For this reason, the fluid loops were maintained at a minimum temperature of 50^{o}F during winter operation, which resulted in a minimum condensing temperature for the distributed refrigeration system on the order of 60^{o}F.

Table 3 - Operating Characteristics of the Water-Source Heat Pumps

Heat Pumps WSHP-1 and WSHP-2 Space Cooling			
Entering Air Temp (^{o}F) Dry- Bulb 73 Wet-Bulb 60		Entering Water Temp(^{o}F) - 100	
	Cooling Capacity (Btu/h)	Compressor Power (kW)	Water Flow (gpm)
Outside Air	293,173	26.4	68
Return Air	252,202	19.4	66
Space Heating Entering Air Temp (^{o}F) - 70		Entering Water Temp (^{o}F) - 70	
	Heat Absorbed (Btu/h)	Compressor Power (kW)	Water Flow (gpm)
Return Air	278,672	17.2	66
Heat Pump WSHP-3 Space Cooling			
Entering Air Temp (^{o}F) Dry- Bulb 73 Wet-Bulb 60		Entering Water Temp(^{o}F) - 100	
	Cooling Capacity (Btu/h)	Compressor Power (kW)	Water Flow (gpm)
	65,634	6.5	20
Space Heating Entering Air Temp (^{o}F) - 70		Entering Water Temp (^{o}F) - 70	
	Heat Absorbed (Btu/h)	Compressor Power (kW)	Water Flow (gpm)
	87,282	6.3	20

Both sites were equipped with similar energy management systems (EMS) that were used for control of the refrigeration. The display cases and walk-in coolers at both sites employed case controllers for control of evaporator refrigerant flow and discharge air temperature. The case controllers employ 4 temperature readings in their control algorithms that are located at the refrigerant inlet and outlet of the coil and at the air discharge and return. Refrigerant flow was regulated at the evaporator by either an electronic expansion valve or by a combination of solenoid valve and electronic suction pressure regulator.

2.1 Test Instrumentation and Data Collection

Both the multiplex and distributed refrigeration systems were thoroughly instrumented in order to assess and compare their operating performances. Table 4 lists the instrumentation associated with the multiplex refrigeration system. Each rack and each condenser was equipped with a watt transducer for power measurement; the watt transducer readings were used for all energy consumption comparisons. Data were collected at 5-minute intervals and stored in the EMS for later retrieval.

The refrigeration load of each suction group was calculated based upon the suction and discharge pressures measured at the rack, the compressor on-off digital signals, and through the capacity curves for each compressor as specified by the manufacturer [reference Copeland Compressor data]. The procedure to determine the refrigeration load consisted of first finding the rated capacity of each compressor at the measured saturated suction and discharge temperatures (determined from the suction and discharge pressure readings). The rated capacities were then adjusted to take into account the actual return gas and liquid refrigerant temperatures. The refrigeration load for the rack suction group was then determined from

$$Q_{sg} = \sum RF_i Cap_i$$

where

Q_{sg} is the refrigeration load of the rack suction group (Btu/h)

RF_i is the on/off status of the compressor (on=1, off=0)

Cap_i is the refrigeration capacity of the compressor for the measured operating conditions

The load calculation was performed using the 5-minute data. The results were later averaged to determine the hourly and daily refrigeration loads.

The refrigerant liquid subcooling loads for racks 1 and 2 were found by calculating a refrigerant mass flow rate from the load calculations and the enthalpy difference between the liquid and suction manifolds of each rack. The amount of subcooling could then be calculated using the liquid temperatures at the condenser outlet and the liquid manifold. The compressor power for subcooling was included in the power measurement of Rack 3. The power used for subcooling was estimated from the compressor performance curves based upon the subcooling load and the state points of the compressors.

Table 4 – Multiplex Refrigeration Instrumentation (Each Compressor Rack)

Rack Suction Pressure (one per suction group)
Rack Discharge Pressure
Return Gas Temperature (one per suction group)
Liquid Temperature at Condenser Outlet
Liquid Manifold Temperature (after subcooler)
Rack Power (kW)
Compressor On/Off Digital (one per compressor)
Condenser Fan Power

Table 5 describes the instrumentation installed on the distributed refrigeration system for the field test. Watt transducers were installed on all compressor cabinets and on the fluid coolers. Refrigeration state point measurements consisted of suction and discharge pressures and return gas temperature. For cabinets with two suction groups separate measurements were made for each suction pressure and return gas temperature

Table 5 – Distributed Refrigeration Instrumentation (Each Compressor Cabinet)

Cabinet Suction Pressure (one per suction group) Cabinet Discharge Pressure Return Gas Temperature (one per suction group) Liquid Manifold Temperature Cabinet Power (kW)
Fluid Cooler Fan Power (one per fluid cooler)

The large number of compressors made it too costly to install digital readings on each compressor. The refrigeration load for each cabinet was estimated based upon the rated capacity of compressors at the measured operating conditions. The rated power for each compressor was also estimated and the refrigeration load was adjusted by the ratio of the measured to the rated compressor power. The refrigeration loads of the compressor cabinets estimated using the instrument reading measured at 5-minute intervals. The refrigeration loads were calculated for each 5-minute interval and later averaging of these load estimates determined the hourly and daily refrigeration load values.

Table 6 lists the instrumentation used for the evaluation of the water-source heat pumps. The operating mode of the heat pumps (heating or cooling) was determined by monitoring the heat pump power, the return air coil temperatures, and the water outlet temperature. Space heating or cooling was indicated by the coil temperature, with a temperature above 150^{o}F indicating heating and a temperature below 50^{o}F indicating cooling. Compressor cycling could also be detected by a change in coil temperature accompanied by a change in total heat pump power. Heat recovered from the fluid loop was determined by the change in fluid temperature and assuming a constant flow rate through the heat exchanger of 66 gpm.

Table 6 – Instrumentation Associated with the Water-Source Heat Pumps

Inlet Fluid Temperature to Water-Refrigerant HX Outlet Fluid Temperature leaving Heat Pump Water-Refrigerant HX Outlet Fluid Temperature leaving Water-Refrigerant HX for Outside Air Cooling Refrigerant-Air Coil Temperature - Return Air Compressor 1 Refrigerant - Air Coil Temperature - Return Air Compressor 2 Return Air Temperature Supply Air Temperature Heat Pump Power

Outside ambient temperature and relative humidity readings were installed at both sites. After several months, it became apparent that the ambient temperature reading at the distributed refrigeration site was not reading correctly. The ambient temperature probe was located on the supermarket roof and the readings were strongly influenced by the sun. Also, the relative humidity readings at both sites stopped functioning after approximately 6 months of operation. It was decided that a better approach for ambient data was to use hourly temperature and humidity readings that were obtained from NOAA for the Worcester Airport. Comparison of the airport temperature readings with readings from the multiplex refrigeration site showed good agreement and the airport relative humidity measurements were considered to be more representative. The NOAA ambient readings were used for analysis of refrigeration performance data for both sites. Both stores were equipped with inside dry-bulb temperature and relative humidity measurements that were used in conjunction with the EMS for HVAC control.

All display cases and walk-in coolers at both sites were equipped with discharge air temperature sensors that were used by the EMS for refrigeration and fixture temperature control. Discharge air temperature readings for all cases and walk-in coolers were collected regularly as part of the field test data.

All instrumentation at both sites was connected to the store EMS and readings were taken at 5-minute intervals. The EMS at both sites had adequate data storage capability so that they could be used as data acquisition systems for the field-testing.

2.2 Discussion of Field Test Results

Data collection was begun at both test stores in November 1999 and is continuing through the present. Test results for the first year of operation and comparison showed that the water-cooled condensers of the distributed system were under-sized, particularly for the low temperature compressor cabinets. The water-cooled condensers on all compressor cabinets were replaced with larger heat exchangers prior to May 2001. The results shown below are based upon data collected after the condenser replacement.

Comparison between the two systems was done on the basis of low temperature (LT) and medium temperature (MT) refrigeration. For the multiplex system, the low temperature refrigeration data consisted of measurements taken from the low temperature compressor rack. For the distributed system, data for the low temperature refrigeration consisted of measurements taken on 4 compressor cabinets (Cabinets A-D) and on the 3 low-temperature suction groups associated with Cabinets F, J, and H. The medium temperature data for the multiplex system were taken from the remaining 2 compressor racks. For the medium temperature refrigeration load of the multiplex system, the refrigeration provided for mechanical subcooling was estimated and removed, since this cooling was not used directly by the display cases or walk-in coolers. The compressor power associated with mechanical subcooling was included with the total power for the medium temperature refrigeration. The medium temperature refrigeration data for the distributed system consisted of measurements taken for the remaining compressor cabinets and the medium temperature suctions groups of the 3 split cabinets. The power and energy data for heat rejection of either system were combined for both medium and low temperature refrigeration, because each fluid loops of the distributed system service both the low and medium temperature compressor cabinets.

Table 7 and Figure 1 show the energy consumption of the two refrigeration systems for two time periods, the first consisting of May through August 2001, and second from November 2001 through February 2002. Data for the months of September and October 2001 were not included, because problems incurred with the EMS at both sites during that period did not allow adequate data to be collected for representative comparison.

Table 7 - Energy Consumption Comparison between Multiplex and Distributed Refrigeration

Energy Consumption (kWh/day)				
	Distributed	Multiplex	Difference	% Difference
May-Aug				
LT Compressor	1306.4	1290.2	16.2	1.2
MT Compressor	1594	1201.2	392.8	24.6
Heat Reject	702.7	608.4	94.3	13.4
Total	3603.2	3100.5	502.7	14.0
Nov - Feb				
LT Compressor	863.2	957.9	-94.7	-11.0
MT Compressor	951.3	635.9	315.4	33.2
Heat Reject	316.1	364.4	-48.3	-15.3
Total	2130.5	1958.2	172.3	8.1

The energy data show that the multiplex compressors consumed less energy for both low and medium temperature refrigeration during summer operation. For the winter, the distributed system showed lower energy consumption for low temperature refrigeration, and higher energy consumption for medium temperature refrigeration. For heat rejection, the multiplex system had lower energy consumption during summer and higher during winter. This finding is significant in that the energy for heat rejection for the distributed system included both fan and pump energy, and the minimum rejection temperature for the fluid loops was 50^{o}F, versus 70^{o}F for the multiplex condensers. It is likely that the reduction in heat rejection energy can be attributed to the operation of the water-source heat pumps for space heating. In general, the energy associated with heat rejection accounted for 15 - 20% of the total refrigeration energy for both the multiplex and distributed refrigeration systems.

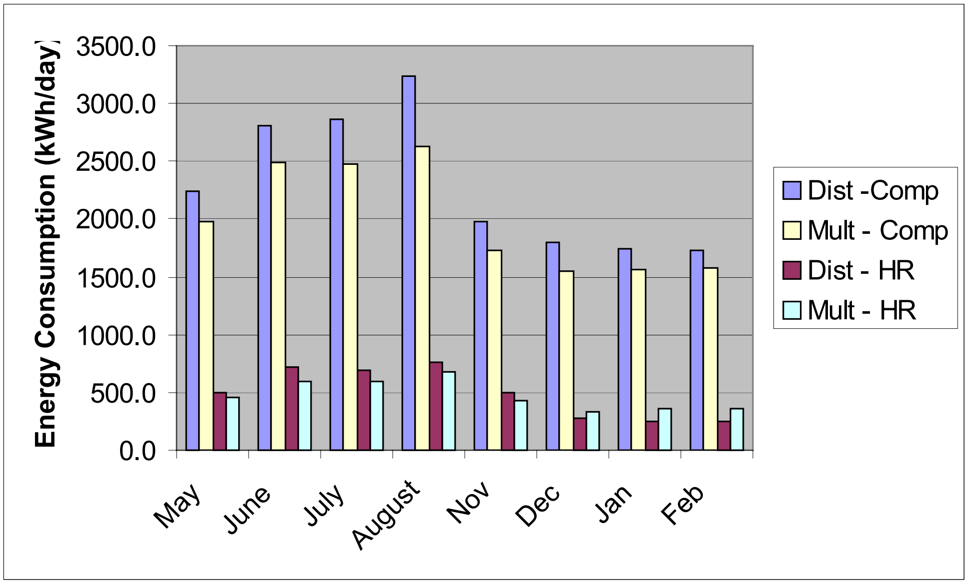

Figure 1 - Average daily energy consumption for the Multiplex and Distributed refrigeration systems

Table 8 provides a description of the operating state points, refrigeration loads, and energy efficiency ratios (EER's) measured for each refrigeration system for summer and winter operation. The operating state points (i.e. saturated suction and discharge temperatures, and return gas and liquid refrigerant temperatures) have been combined for each system in terms of low and medium temperature refrigeration. The average state point values were calculated on the basis of refrigeration load by

$$SP_{avg} = \frac{\sum SP_i Q_i}{\sum Q_i}$$

where

SP_{avg} = the average state point value
SP_i = the state point value for a particular suction group
Q_i = the refrigeration load associated with the suction group

The refrigeration load measurements indicated that the low temperature refrigeration loads are significantly different for the multiplex and distributed systems and the load for the distributed system is lower than that of the multiplex system. The medium temperature load for the distributed system is higher than that measured for the multiplex system.

Normalization of the test results was done by comparing EER values. This comparison shows that the multiplex system had significantly higher EER values for low and medium temperature refrigeration for both summer and winter operation. For summer operation the multiplex system EER values were 34.7 and 18.5% higher than the EER values of the distributed system for low and medium temperature refrigeration, respectively. For winter operation the multiplex system EER values were higher than those of the distributed system by 12.1 and 22.2% for low and medium temperature refrigeration, respectively. Figure 2 shows the EER values for both systems as a function of average daily dry-bulb temperature. The results in this plot show that the EER values for the multiplex system were consistently higher than those of the distributed system.

Table 8 - Average State Points , Refrigeration Loads, and EER's for the Multiplex and Distributed Refrigeration Systems

	May - Aug		Nov - Feb	
Low Temp	Distributed	Multiplex	Distributed	Multiplex
SST (^{o}F)	-15.8	-19.2	-17.1	-20.2
SDT (^{o}F)	90.0	82.8	61.0	71.0
Return Temp (^{o}F)	11.5	2.2	16.3	11.0
Liquid Temp (^{o}F)	79.1	57.8	56.9	50.4
Ref Load (Btu/h)	340,800	455,600	309,000	385,900
Comp Power (kW)	54.4	53.8	36.0	39.9
EER (Btu/W-h)	6.34	8.54	8.63	9.67
Med Temp	Distributed	Multiplex	Distributed	Multiplex
SST (^{o}F)	16.2	22.9	13.4	20.2
SDT (^{o}F)	92.5	83.3	72.1	67.0
Return Temp (^{o}F)	46.4	48.9	44.3	49.3
Liquid Temp (^{o}F)	85.0	68.1	67.1	58.3
Ref Load (Btu/h)	729,900	653,200	541,000	442,800
Comp Power (kW)	66.4	50.1	39.6	26.5
EER (Btu/W-h)	11.16	13.22	13.71	16.75

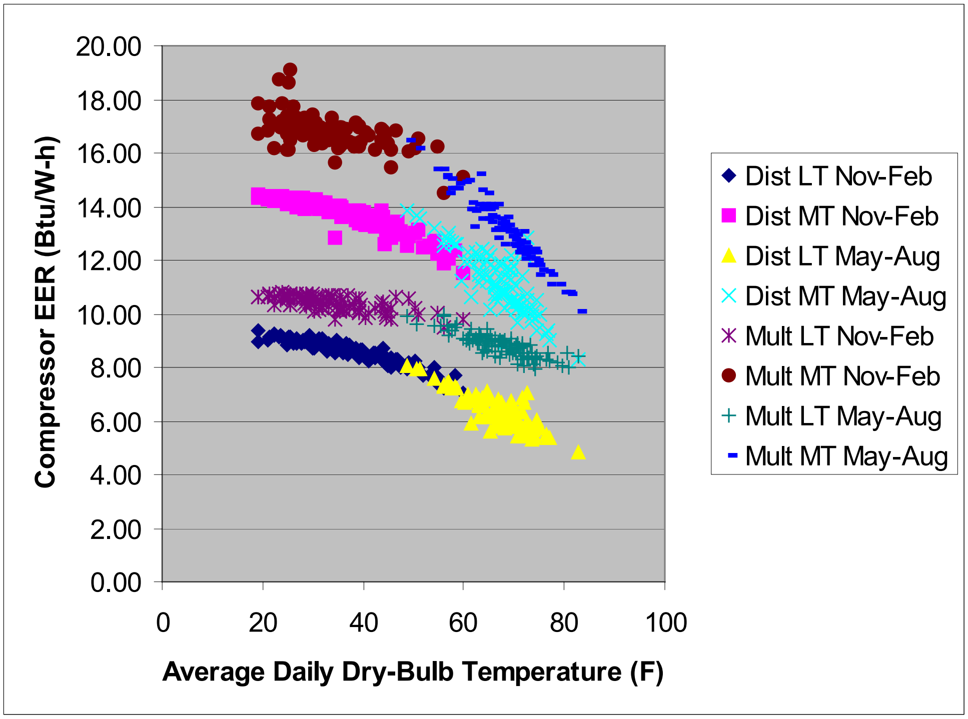

Figure 2 - EER Comparison between the Multiplex and Distributed Refrigeration

A TEWI analysis of each test system was conducted and the results of this analysis are given in Table 9. The charge sizes for the two systems were estimated at 3,000 lb. of refrigerant for the multiplex system and 1,000 lb. for the distributed system. Actual refrigerant leakage rates for the systems are not known. The estimated leakage rates employed in the analysis are 20 and 5 % for the multiplex and distributed systems, respectively. The analysis showed that the distributed system had a lower TEWI for both the summer and winter periods despite the higher indirect value due to the higher energy consumption.

Table 9 - TEWI Analysis for the Field Test Results

	Refrigerant Leakage (lb)		Energy (kWh)	TEWI (kg CO2)		
	R-404A	R-22		Direct	Indirect	Total
May - August						
Multiplex	67	133	381,300	202,055	247,845	449,900
Distributed	17		443,169	24,895	288,060	312,954
Difference						136,046
Nov - Feb						
Multiplex	67	133	234,960	202,055	152,724	354,779
Distributed	17		255,600	24,895	164,034	188,929
Difference						165,850

Multiplex leak rate estimated at 20%/yr Distributed leak rate estimated at 5%/yr

The field test results for the water-source heat pumps are given in Table 10 for the winter period from November 2001 through February 2002. The table lists the average recovery rate of heat from the fluid loops. The amount of heat recovered was estimated at 25.9% of the total heat rejection of the refrigeration system. This amount is considerably less than the capability of the heat pumps, which can recover as much as 663,200 Btu/h, or 59.8% of the rejected heat from the refrigeration system. The ambient conditions during this time period were very mild so that the amount of space heat needed for the store was much less than normally seen. For much of the time, the large heat pumps had only one compressor operating. The small heat pump had a run fraction of approximately 30% during this time period.

Table 10 - Field Test Results for the Water-Source Heat Pumps Space Heating Performance (November 2001 - February 2002)

	Average Recovered Heat (Btu/h)	Average Space Heat (Btu/h)	Heat Pump Energy (kWh)	Gas Displaced (therms)
November 01	248,400	326,100	17,236	2,966
December 01	293,500	381,000	20,531	3,602
January 02	301,900	394,700	21,004	3,702
February 02	307,200	404,700	20,295	3,446
Total			79,066	13,716

Despite limited operation, the water-source heat pumps were able to displace approximately 13,716 therms of natural gas. The value of this displacement is dependent upon the utility rates for electric and gas. Figure 3 shows the estimated energy cost savings for a range of utility rates. For this particular site, the average commercial electric rate is approximately $.09/kWh and the gas rate is $.75/therm. The estimated energy cost savings seen over this 4-month period was $3,171.

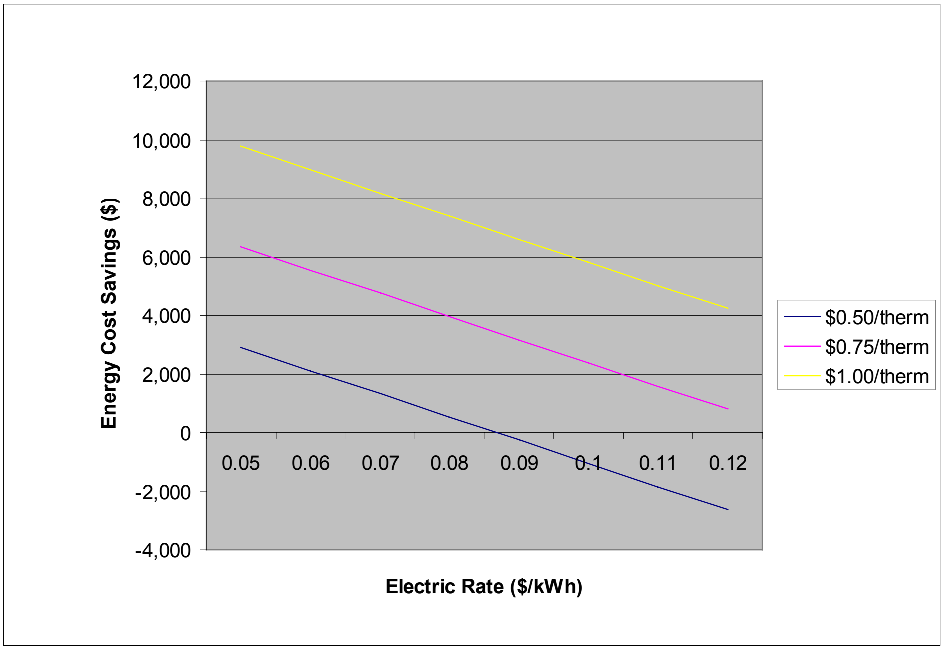

Figure 3 - Water-Source Heat Pump Energy Cost Savings (Nov '01 - Feb '02)

2.3 Conclusions and Recommendations

The distributed refrigeration system consumed much more energy and operated at much lower EER values relative to the baseline system than expected based on the projections from the earlier analytical studies [2,3]. This increased energy use can be attributed to several factors:

- Use of dry fluid coolers for heat rejection - Lower energy consumption should be expected if evaporative heat rejection was used for the fluid loops. This is particularly true for summer operation. With evaporative heat rejection the fluid loop temperature could be dropped below the ambient dry-bulb temperature resulting in lower condensing temperatures at the compressor cabinets. Evaporative heat rejection would also require less fan power than dry heat rejection, which would help to lower energy consumption. It should be noted that evaporative heat rejection can be used with any supermarket refrigeration system but usually is not due primarily to the increased maintenance requirements.
- The performance of the scroll compressors - Comparison of manufacturer's data for the scroll and reciprocating compressors suggests that the EER of the scroll compressors used by the distributed system is less than that of the reciprocating compressors used by the baseline system at the same saturated suction and discharge temperatures. This observation is true for both low and medium temperature compressors. The methods needed to overcome this performance difference include operation of the scroll compressors at lower saturated discharge temperature and at higher saturated suction temperature. The differences in these values obtained by the distributed system tested were not adequate to overcome the performance difference between these two compressor types. It is also desirable to maintain as low a return gas temperature as possible to eliminate excess refrigeration load. The return

gas temperature values were about the same for both the multiplex and distributed refrigeration systems despite the shorter piping runs used by the distributed system.

- Extensive mechanical subcooling of the multiplex systems coupled with ineffective vapor injection subcooling for the distributed system compressors - The multiplex system used mechanical subcooling extensively for both the low and medium temperature compressor racks to maintain low liquid refrigerant temperature. This added subcooling produced more of a performance enhancement for the multiplex system than had been originally expected. Subcooling by mid-scroll vapor injection was installed on the scroll compressors in the low temperature compressor cabinets of the distributed system. The subcooling obtained was found to be very limited because the injection ports on the scroll compressors were sized for liquid injection and were too small to allow adequate vapor flow.

Some of the performance difference between the two systems can also be attributed to the use of R-22 for medium temperature refrigeration on the multiplex system and R-404A on the distributed system. This difference will disappear in later installations after the phase-out of R-22.

TEWI analysis for the two systems indicates that the distributed system has less environmental impact with lower total CO_2 production than the multiplex system. This result was obtained even though the indirect CO_2 production of the distributed system was greater due to higher energy consumption.

The distributed system that was tested did not have compressor cabinets located in the sales area of the supermarket. The compressor cabinets were located either in backrooms around the perimeter of the store, or above several of the walk-in coolers. This fact suggests that other compressor types, such as reciprocating could have been used in the cabinets in this installation without concern about excessive noise in the sales area. Small rooftop refrigeration units may also be an effective alternative for a distributed refrigeration system. The use of more efficient reciprocating compressors could help to reduce the energy difference between the distributed and multiplex systems.

Operation of the water-source heat pumps in conjunction with the distributed refrigeration system was shown to be effective at recovering reject heat from the refrigeration system for space heating. Energy cost savings were found for the test site due to this heat recovery by the heat pumps. It should be noted that this heat recovery approach could be applied to any supermarket refrigeration system that employs a fluid loop for heat rejection and is not limited to the distributed system.

3. References

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4. Acknowledgements

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Table 1 - Description of the Multiplex Refrigeration Test Store

Circuit	Case Length, No. of Doors, or Walk-in Floor Area	Design Refrigeration Load (Btu/hr)
Rack 1 Low Temperature - Sat. Suction Temp -20°F Refrigerant - R404A		
Reach-in Frozen Food	76 Doors	114,000
Multi-Deck Frozen Meat	40 ft	60,520
Multi-Deck Frozen Fish	18 ft	27,234
Multi-Deck Bakery	12 ft	18,156
Walk-in Freezers	925 ft^2	69,375
Ice Maker		36,000
Rack 1 Low Temperature - Sat. Suction Temp -30°F		
Coffin Ice Cream	104 ft	74,600
Coffin Shrimp	16 ft	5,840
Walk-in Ice Cream	180 ft^2	17,100
Rack 2 Medium Temperature - Sat. Suction Temp 20°F Refrigerant - R22		
Reach-in Dairy	46 Doors	62,560
Single-Deck Produce	56 ft	53,200
Multi-Deck Produce	24 ft	32,160
Service Meat & Deli	64 ft	23,692
Multi-Deck Deli	48 ft	69,920
Coffin Cheese & Deli	16 ft	24,800
Tables Fish	20 ft	10,000
Cooler Floral	140 ft^2	10,500
Floral Display	8 ft	10,336
Rack 2 Medium Temperature - Sat. Suction Temp 15°F		
Bakery Display	15 ft.	12,550
Multi-Deck Meat	68 ft	91,800
Isle Cheese & Deli	56 ft	63,360
Isle Ready Meals	52 ft	57,200
Rack 3 Medium Temperature - Sat. Suction Temp 20°F Refrigerant - R22		
Walk-in Coolers	2,195 ft^2	197,475
Rack 3 Medium Temperature - Sat. Suction Temp 35°F		
Meat Prep Room	1,010 ft^2	141,400
Produce Prep Room	520 ft^2	39,000
Mechanical Subcooling		361,672

Table 2 - Description of the Distributed Refrigeration Test Store

Circuit	Case Length, No. of Doors, or Walk-in Floor Area	Design Refrigeration Load (Btu/hr)
Cabinet A Sat. Suction Temp -25°F		
Walk-in Freezer	1,013 ft^2	76,000
Cabinet B Sat. Suction Temp -25°F		
Walk-in Freezer	506 ft^2	38,000
Cabinet B Sat. Suction Temp -15°F		
Reach-in Frozen Food	23 Doors	34,500
Cabinet C Sat. Suction Temp -15°F		
Reach-in Frozen Food	52 Doors	67,500
Multi-Deck Frozen Food	8 ft	11,880
Cabinet D Sat. Suction Temp -25°F		
Multi-Deck Frozen Meat	56 ft	57,120
Multi-Deck Frozen Fish	28 ft	23,740
Walk-in Frozen Fish	144 ft^2	12,000
Cabinet E Sat. Suction Temp 20°F		
Multi-Deck Dairy	120 ft	160,800
Multi-Deck Cheese	14 ft	18,760
Walk-in Dairy	857 ft^2	62,000
Cabinet F Sat. Suction Temp 17°F		
Service Deli	84 ft	25,920
Multi-Deck Cheese	20 ft	28,800
Isle Deli	16 ft	38,388
Walk-in Deli	153 ft^2	15,300
Walk-in Raw	48 ft^2	4,800
Cabinet F Sat. Suction Temp -20°F		
Walk-in Deli Freezer	81 ft^2	6,075
Cabinet G Sat. Suction Temp 17°F		
Single-Deck Meat	24 ft	10,680
Multi-Deck Meat	68 ft	91,800
Floral Display	24 ft	31,008
Multi-Deck Deli	28 ft	42,420
Single-Deck Fish	16 ft	17,600
Walk-in Fish	110 ft^2	11,000
Cabinet H Sat. Suction Temp 35°F		
Meat Prep Room	6,144 ft^2	153,600
Cabinet H Sat. Suction Temp 5°F		
Ice Maker		36,000
Cabinet I Sat. Suction Temp 20°F		
Multi-Deck Produce	116 ft	122,680
Table Fish	15	7,500
Walk-in Produce	984 ft^2	66,520
Cabinet J Sat. Suction Temp 15°F		

Walk-in Meat	1,056 ft^2	79,200
Walk-in Bakery	120 ft^2	19,800
Cabinet J Sat. Suction Temp -20°F		
Walk-in Freezer Meat	100 ft^2	7,650
Walk-in Freezer Bakery	144 ft^2	10,800
Multi-Deck Bakery	14 ft	15,780
All Compressor Cabinets employ R-404A as the refrigerant		