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MASTER THESIS

Investigation of Deep-freeze Refrigeration Systems in Supermarket Application

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SUMMARY

A thorough investigation on Deep-Freeze Refrigeration Systems in Supermarket Application is Purposed in the report. The objective is achieved by the combination of two separated parts, an intensive project study as well as an extensive refrigeration alternative and optimization survey.

Within the first part, completely indirect systems and partially indirect systems with secondary condensing cycles were widely analyzed and compared through two representative plants, and all the research had been carried out with the integration of field experiments. The conclusion is that, in the project study case, the completely indirect refrigeration system performed much better than the partially indirect (secondary condensing) system as to system performance, energy consumption, total investment and environmental impact points of view. This indicates that, in supermarket application, Indirect Refrigeration Systems generally perform better than Direct Refrigeration Systems, as to the aspects.

Refrigeration systems, refrigerant alternatives as well as system optimization methods were extensively explored in the second part. The results indicate that many kinds of methods and technologies could be used to improve system performance and maximize system efficiency in the field of supermarket refrigeration, for both retrofitting and new systems. Minimizing environmental impact was of special concern throughout the investigation. Further experiments and practical experiences are expected to verify the theories and conclusions.

ACKNOWLEDGEMENT

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LIST OF CONTENT

LIST OF SYMBOLS

DX	Tested Partially Indirect (Secondary Condensing) Deep-Freeze Refrigeration System
SQD	Tested Completely Indirect Deep-Freeze Refrigeration System
HQL_{beef}	High Quality Life with Beef as an example (day)
T, t	Temperature (°C)
H, h	Enthalpy (kJ/kg)
COP_2	Coefficient of Performance (Cooling)
C	Specific Heat Capacity (kJ/kg * K)
m	Mass flow (kg/s)
ΔT	Temperature Difference (°C)
Δh	Enthalpy Difference (kJ/kg)
Q_2	Cooling Capacity (kW)
W	Energy Consumption (kWh)
N	DX System Night Cover Factor
T	SQD System Time Factor
E	Power (kW)

Subscript

b.	Beef
2k	Compressor Outlet
1k	Compressor Inlet
1s	Condenser Outlet
20	Temper 20
40	Temper 40
original	(Pressure – Enthalpy) Diagram Based Condition
real	Actual Condition for Calculating COP_2
2kr	DX Cabinet Outlet
1sr	Before Expansion Valve
d.	Weekday, everyday
e.	Weekend
y.	Year
de.	Defrost
w.	Week
dry	Dry Cooler
fan	fan
comp.	Compressor
cond.	Condenser
sc.	Subcooler
cir.	Circulation
ptotal	Pumps in Total

INTRODUCTION

1) BACKGROUND

The developing process of human society is a process full of continuous creating and progressing. Refrigeration Technology is one of the most outstanding creations. Instead of depending on natural environment, humans can truly establish artificial conditions to fulfill basic needs by using Refrigeration. In brief, Refrigerating engineering is a technology to create and maintain temperatures lower than the surrounding [1]. Food storage (handling and transport) as well as air conditioning are two typical applications.

Staring from 1960's, the development of refrigeration got a rapid progress. From chilling and chilled storage to freezing and frozen storage, from refrigerated transport to retail distribution, accompanied with the development of air conditioning technology, refrigeration engineering places a more and more important role in food and building industry as well as other fields.

However, along with the wide utilization of refrigeration technology, the corresponding energy consumption also increases quickly. Most of it is consumed in the form of electrical power. Nowadays, the power demand created by refrigerating facilities takes a large part of total power generation capacity.

Another result of the widespread use of refrigeration is environmental pollution. Two representatives are Ozone Depletion that is caused by chloro-fluoro-carbons (CFCs) plus hydro-chloro-fluoro-carbons (HCFCs), and Greenhouse Effect that further result in Global Warming Effect. The first serious environmental impact was not found until 1973. Between 1930's and 1980's, CFCs and HCFCs were widely used in modern refrigeration systems [1].

A thorough investigation on foodstuff storage refrigeration systems, especially on deep-freeze refrigeration systems, was presented in this paper, mainly from energy saving, efficiency improving and environment protection points of view. In order to get practical experience about system performance and direct comparisons between different systems, field experiments were carried out within the investigation.

2) GENERAL PURPOSE

The purpose for this paper is to process a thorough investigation on Deep-Freeze Refrigeration Storage Systems with Display Cabinets used in current supermarkets. The investigation is achieved from four main aspects which are temperatures in cabinets, system energy consumption and efficiency, economic evaluation, as well as environmental impact. All of them are highlights for supermarket refrigeration systems.

The purpose is accomplished from two separated aspects. In first part, integrated field experiments were intensively analyzed from technical point of view. Within the experiments, the two target systems were chosen as representatives for typical partially indirect and completely indirect systems used in current supermarkets. In second part, different alternative refrigerants and systems, as well as optimization methods were widely discussed through theoretical studies and literature survey. With the combination of the two components, the purpose of exploring supermarket deep freeze refrigeration systems is completed by both detailed technical analysis and thorough overview.

3) DEFINITIONS

According to the definition in refrigerating engineering, there exists a clear distinction between rooms for chilled and for frozen products. In the first application one is aiming for temperatures in the range 0 to +8 ˚C, while for frozen products the temperature arrange is from –18 ˚C to –25 ˚C [2]. Within this paper, the raised investigation and discussions are mainly focus on deep-freeze refrigeration systems for frozen foodstuff in supermarkets.

Swedish regulations define that the deep frozen food should be stored at –18 ˚C or lower. The normal chilled food temperature should not exceed +8 ˚C while some product, such as meat, egg and fish chilled temperature should not exceed +4 ˚C. [3]

REVIEW OF THE LITERATURE

A literature survey was widely carried out as a part of the investigation. A lot of articles and papers were found dealing with deep freeze display cabinet refrigeration systems, but concentrate on specific areas, such as system economic evaluation, refrigerant properties, and compressor performance. Many documents containing latest system improving technologies were also found during the survey. Some representative papers were quoted and discussed in the second theoretical study part, while three reports were especially used as supplementary of the on the spot test in first part of the thesis.

The first one is “Performance of a Display Case at Low Temperatures Refrigerated with R404A and Secondary Refrigerants” [4]. The paper presents a similar test performed according to ASHRAE Standard 72. Within the test, different cabinet temperature profiles were recorded as a function of the refrigerant inlet temperature to the heat exchanger for directly and indirectly refrigerated display cases. The information is very useful for a further comparison on cabinet performance with different refrigerants at different inlet temperature, but such test can not be done in the on-spot measurement. Moreover, the paper presented secondary loop and warm liquid defrost systems could also be studied for comprehending the indirect Soft and Quick Defrost system (SQD) which was going to be tested in the experiment, since some of the mechanisms are alike.

The effect of changing compressor power and temperatures in the display case caused by varied outdoor temperature and moisture, is analyzed in the paper named “Field Experiences in Three Supermarkets in Sweden” [5]. Such kind of effect was not discussed in the experiment. Furthermore, the influence of night cover was studied in [5], and reaching the same conclusions as this thesis.

The paper, “Indirect Cooling with Ammonia in Supermarkets” [6], presents a comparison between a traditional direct expansion HCFC plant and an indirect two stage ammonia plant. Such similar field experience is very helpful for confirming the results and conclusions achieved from the test.

INTENSIVE PROJECT STUDY

1 DIRECT AND INDIRECT REFRIGERATION SYSTEM INTRODUCTION

There are many types of refrigeration systems for supermarket cabinets. The same system can be put into different categories depending on varied definitions. For example, a plant can be defined as either a HCFC system (based on the charged refrigerants) or a two-stage system (based on the system configuration). However, the distinction of direct and indirect systems is the most important one among all of them, because a completely new concept has been introduced by the definition. Due to its advantages, more and more supermarkets are trying to use indirect systems for both retrofitting and new installation. Following is a schematic of two basic system arrangements.

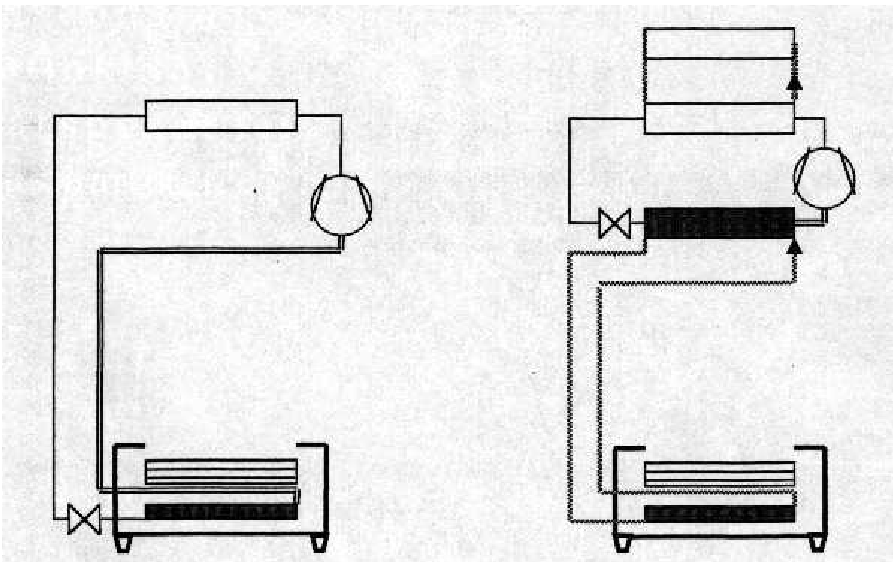


Figure 1. Schematic of two basic arrangements – direct and indirect systems [7]

1.1 DIRECT SYSTEM

Direct refrigeration storage systems are designed based on the basic vapor compression cycle. Four processes, which are evaporation, compression, condensation and throttling, maintain the system operation. Normally, compressors are located in a machine room and driven by electric motors. Condensers can be cooled either by air, or central cooling towers through secondary circuits. Together with circulating fans, evaporators are placed at the bottom of cabinets, which are set in the shopping area. Expansion devices are situated close to cabinet groups. In most of supermarkets, there is a long distance between machine room and shopping area, therefore, extensive pipes have to be added to systems, which leads to the problems of increasing capital cost for both piping and insulation, as well as potential leakage and cold loss sources. Moreover, most refrigerants are expensive and environmentally damaging when released into the atmosphere.

1.2 INDIRECT SYSTEM

The most important distinction between a direct and an indirect system is that a secondary circuit is introduced to the basic system on evaporating side. The working principle and the temperature profile of secondary refrigerant circuit are indicated in the following figure:

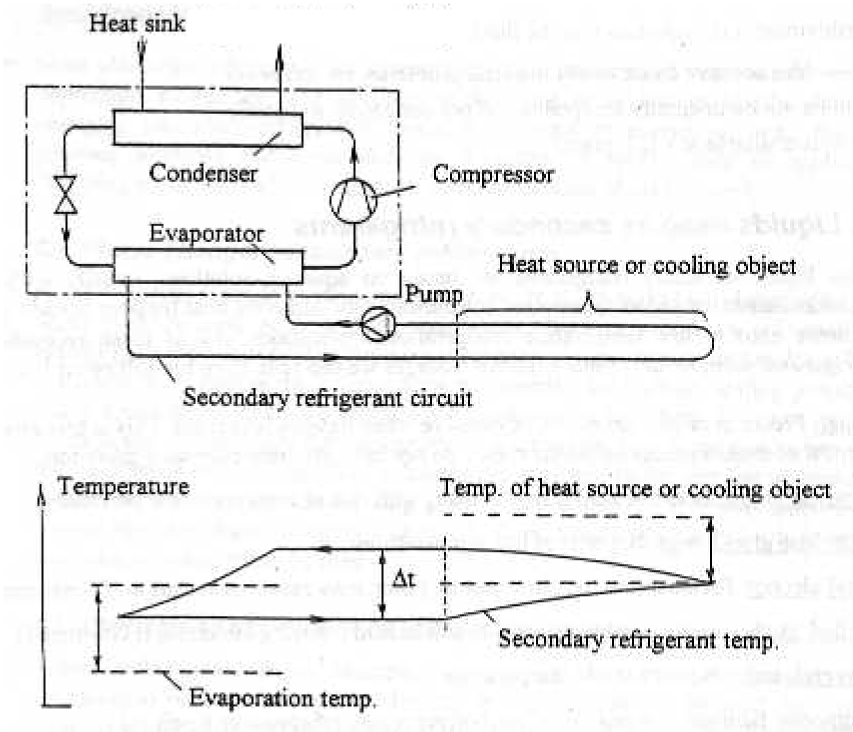


Figure 2. Indirect refrigeration system with temperature profile of secondary refrigerant circuit [1]

✓ **Advantages**

- Refrigerant charge can be minimized in the primary circuit, which can lead to the reduction of both capital cost and refrigerant leakage.

- Compact factory built units can be used for different kinds of purpose, the installation work can be simplified, which leads to a further reduction of initial investment.

✓ **Disadvantages**

- One more heat exchanger and several circuit pumps are added to systems, which will increase costs. An added temperature difference may result in a higher energy demand.

- The difficulty of selecting a suitable secondary refrigerant. Aqueous solutions and non-aqueous heat transfer liquids are two main categories of liquid secondary refrigerants that are used. However, they have both positive and negative effects. On the contrary, some phase changing secondary refrigerants, such as Carbon Dioxide, look very promising.

✓ **Highlight – System Energy Consumption**

More energy consumption compared with direct systems is normally treated as a negative effect for indirect systems. Nevertheless, other technical papers report that in practice the annually total energy consumption of a well-built indirect system is often lower than that of a direct system. This issue will also be investigated again within this paper.

2 PURPOSE

Although two stage and cascade systems are already commercially available for today's supermarket refrigeration, basic direct and indirect chilling and freezing separated plants are still the most commonly used systems. To better understand the different system performance from technical point of view, field measurements had been designed and processed for testing two typical deep freeze refrigeration systems in two Swedish supermarkets.

The first test system is a traditional direct expansion refrigeration system with a secondary condensing cycle. Electric heater defrost is used in the system. The second one is a completely indirect system with secondary condensing and evaporating cycles. Warm liquid defrost is employed in the system. These two targets can be treated as representatives for partially indirect and completely indirect systems that are widely used in today's supermarkets. Therefore, most of the experimental based discussions and analysis are suitable to all of the refrigeration systems which are similar to those two representatives.

3 FIELD EXPERIMENT LAYOUTS

3.1 METHODS AND PROCEDURES

The field measurement was projected under the cooperation of Department of Energy Technology, **Royal Institute of Technology** (KTH) and **FRIGOTECH AB**, a Swedish refrigeration system installation company. Two types of deep freeze systems were chosen from the company's installation works at two HEMKÖP supermarkets. HEMKÖP is a large scale Swedish supermarket chain enterprise. The first traditional direct expansion (DX) deep freeze refrigeration system belongs to the supermarket at Sundbyberg Centrum, Stockholm; while the second one installed at Mörby Centrum, Stockholm, is an indirect system with warm liquid defrost, Soft Quick Defrosting (SQD) mechanism which is patented by FRIGOTECH AB. (DX was used as the direct expansion system and SQD was used as the secondary system with the patented warm liquid defrosting in the following parts.)

✓ Test Period

- DX System at Sundbyberg Centrum

1. 2 days (Nov. 17th ~ Nov. 19th, 2000)
2. 7 days (Feb. 26th ~ Mar. 5th, 2001)

- SQD System at Mörby Centrum

1. 2 days (Nov. 14th ~ Nov. 16th, 2000)

7 days (Feb. 12th ~ Feb. 19th, 2001)

In the first short test period (Nov. 14th ~ Nov. 19th, 2000), Plastic Chickens®, which are specially designed for food storage refrigerating cabinet temperature monitoring, were used for logging temperature data in the cabinets, and the test cabinet sections were fully loaded with real food. (See pictures in Appendix D) Because of the too short logging interval as well as the test period, unstable temperature distributions were found in all test results. Therefore, the tests were repeated again in longer periods. Due to the demand of the supermarket administrators, smaller Tinytalk® temperature loggers were used in the second test period, and the test cabinet sections were allowed to be artificially fulfilled by neither real nor dummy food. Only normal food loads were maintained in both two cases. (See pictures in Appendix D) Consequently, the food temperatures, especially the food temperatures in the top position of the cabinets, were better reflected in the first test period. Unfortunately, only the data gathered in second test period can be used for analyzing. Such problem is very typical in field measurements. However, the purpose of monitoring the practical conditions without disturbing supermarket's normal operation was accomplished in the field tests.

Due to the short test periods as well as the short interval, the outdoor climate can be treated unchanged during the whole period, hence the influence of the changing outdoor temperature and moisture can be neglected. With such assumption, the comparison could be made under the same condition.

All important parameters which can be used for system evaluation were measured and recorded in two separated groups. The first group of data was gathered from the compression systems in machine rooms. The second group of data was collected from the display cabinets.

Machine Room

Measurements had been carried out on compressor energy consumption, discharge and suction line temperature, condenser outlet temperature, and inlet, outlet condensing temperatures on dry cooler side. More temperatures were measured in SQD system for its sub cooler and brine inlet, outlet. Other data such as time and mass flow were also recorded for investigating defrosting process. All the data were logged by ETM® and Woodly® instruments.

Display Cabinet

Two similar sections in the display cabinet groups were chosen at different tested places. Seven thermal sensors were placed in the cabinets. The schematic of the positions is shown in figure 3. The same positions were remained identical for two testing. Three Tinytalk® temperature loggers were placed at position [1] [2] [3] which stand for bottom, middle, and top temperature in the cabinet. The four other probes connected with a Woodly logger were set at position (1) (2) (3) (4) which stand for air outlet, air inlet temperature, and top inlet, top outlet temperature. The sensors were arranged for collecting temperatures at different places in the tested cabinets. Position [1] [2] [3] (3) (4) were fixed on a plastic board for standstill. Position (1) (2) were directly placed at cabinet air outlet and inlet. (See pictures in Appendix D)

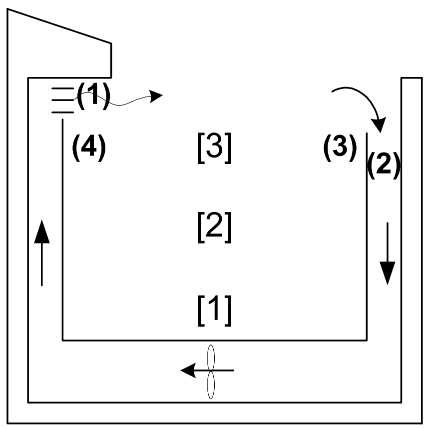


Figure 3. Position of Temperature Loggers and Probes in Display Cabinets

One special point differences from laboratory testes is that the temperature loggers and probes ([1] [2] [3] (3) (4)) were designed surrounded by real food, instead of dummy food package. The real food load was around 75% of full load. The purpose is to record the real cabinet performance with normal customer purchase and collect the real food temperatures at different positions within the cabinets at the same time.

Environment Condition

Although the influence of environment was neglected during the comparison, the indoor temperatures as well as moisture were also logged during the testing period for a complete comparison.

✓ Supermarkets' Operation Time

- Hemköp at Sundbyberg Centrum

Monday to Friday	8:00 ~ 21:00
Saturday	8:00 ~ 19:00
Sunday	10:00 ~ 19:00

- Hemköp at Mörby Centrum

Monday to Friday	9:00 ~ 20:00
Saturday	9:00 ~ 18:00
Sunday	11:00 ~ 18:00

The different supermarket operation times may contribute to different cabinet temperatures and energy consumption distribution. The significant time difference also exists between weekday and weekend for each place. The first impact is noticed in the analysis part, and the second impact is eliminated by the calculation.

✓ Data Source

In the rest of the thesis, all the data used in calculations and analysis are coming from the second test results.

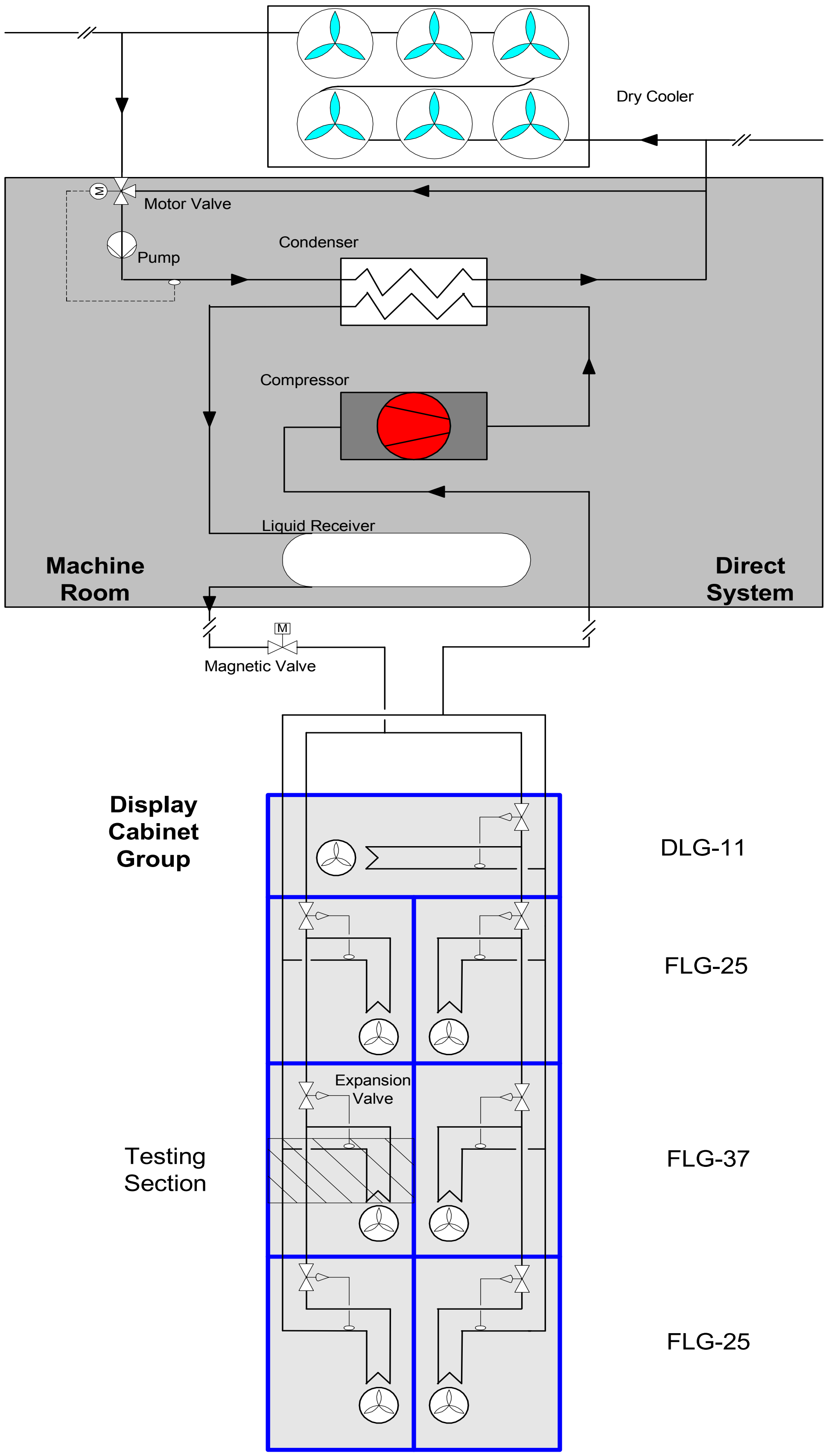


Figure 4a. DX System Configuration at Sundbyberg Centrum

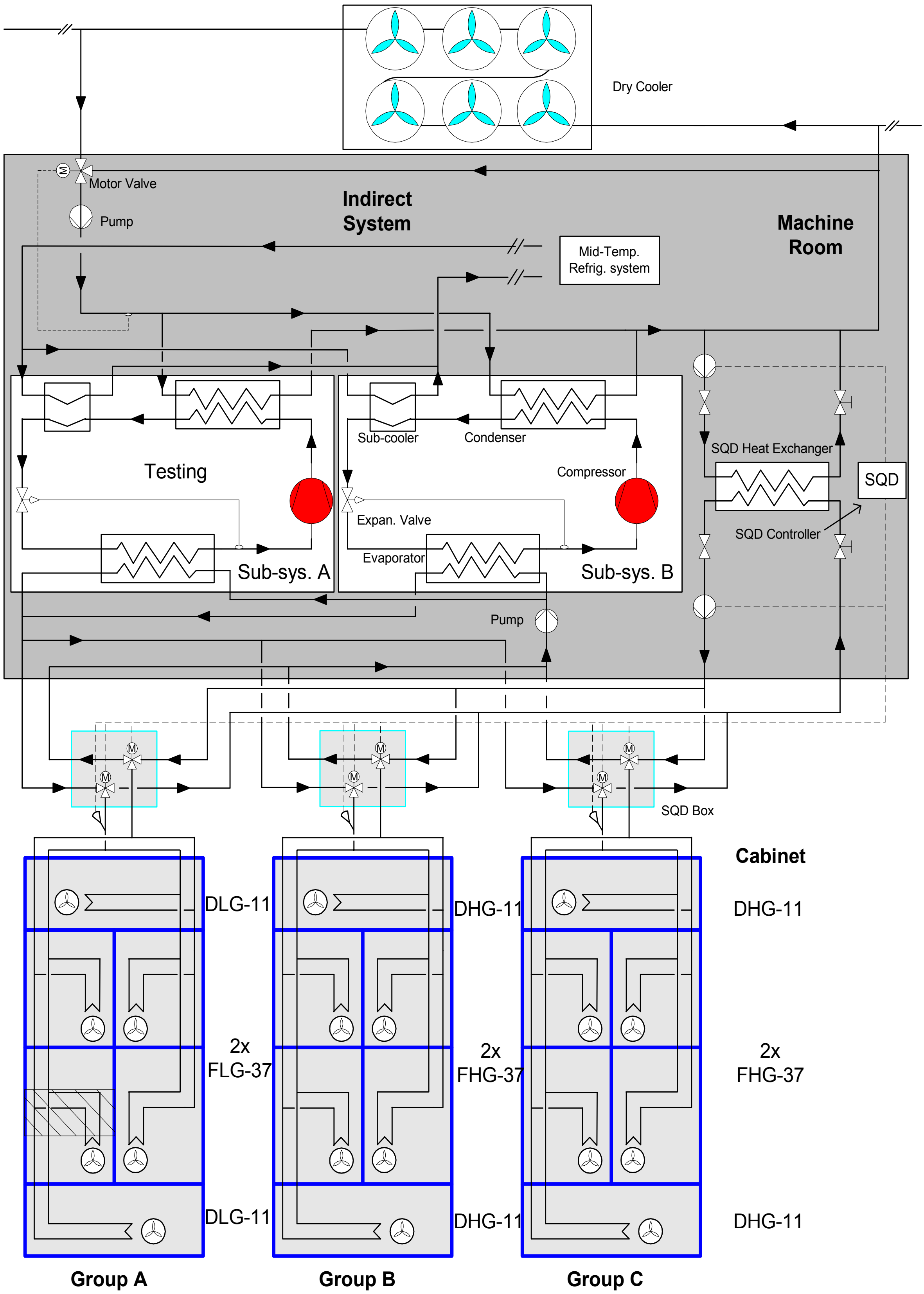


Figure 4b. SQD System Configuration at Mörby Centrum

3.2 SIMPLIFIED SYSTEM CONFIGURATION

The configurations of the testing direct (DX) and indirect (SQD) refrigeration systems are shown in figure 4a and 4b.

3.3 INSTALLATION DESCRIPTION

For achieving closest test condition, the selected refrigeration system installations in two supermarkets have same scale and similar capacity.

Installation Scale

Installation Area: 1 300 m^2
Cold Store: 7
Deep Freeze Store: 1
Refrigeration Cabinet: 84 m
Deep Freeze Cabinet: 30 m

Where: 1 m stands for 1 meter's long standard cabinet with two parallel sections

Rated Installation Capacity (100%, 24hr)

Indirect System (SQD): 6 090+ 6 840x2=19 770 W
Direct System (DX): 6 065 W
19 770/ 6 065=3.26

Cabinet refrigeration capacity for the SQD system (19 770 W) is 3.26 times the capacity of DX system (6 065 W). Therefore, the later energy consumption and investment comparisons were based on the SQD rated cabinet capacity (19 770 W), which means the data coming from DX system needed to be multiplied with 3.26 for getting comparable results.

3.3.1 TESTED DIRECT SYSTEM AT SUNDBYBERG CENTRUM (DX)

The target deep-freeze plant is a typical direct expansion system with a semi-hermetic motor compressor using R404A as refrigerant. One cabinet group is served by the compressor. All the plants (including chilling plants) in the machine room are connected to a main roof-mounted dry cooler outside the building. The defrosting is based on defrost timers and traditional electrical heat elements placed in each cabinet.

3.3.2 TESTED INDIRECT SYSTEM AT MÖRBY CENTRUM (SQD)

The deep-freeze SQD system has two indirect condensing and evaporating circuits, and special design for defrosting and subcooling. The system contains two similar primary cycles circulated by two compressors with refrigerant R404A. The compressors are the same semi-hermetic types (as the one in the DX system), and they have the same capacity. All the plants share one outdoor placed dry cooler. Three cabinet groups are connected by a common loop which is the secondary evaporating cycle.

The recycling of waste energy for defrosting is one of the cardinal points in the construction concept of the SQD system. Instead of electrical heaters, warm liquid will go through the cabinets during the defrosting period, and the energy is coming from the secondary condensing cycle. Such design can use recovered waste energy and decrease the food temperature changing (Details discussed in the later part under the special frosting and defrosting topic). The same heat exchangers are placed after the condensers in two primary cycles and act as sub-coolers. The cooling source is from a nearby chilling plant (see SQD system configuration in figure 4b).

The detailed parameters for the two systems are listed in Appendix A.

4 TEST RESULTS

4.1 RESULTS FROM DISPLAY CABINET

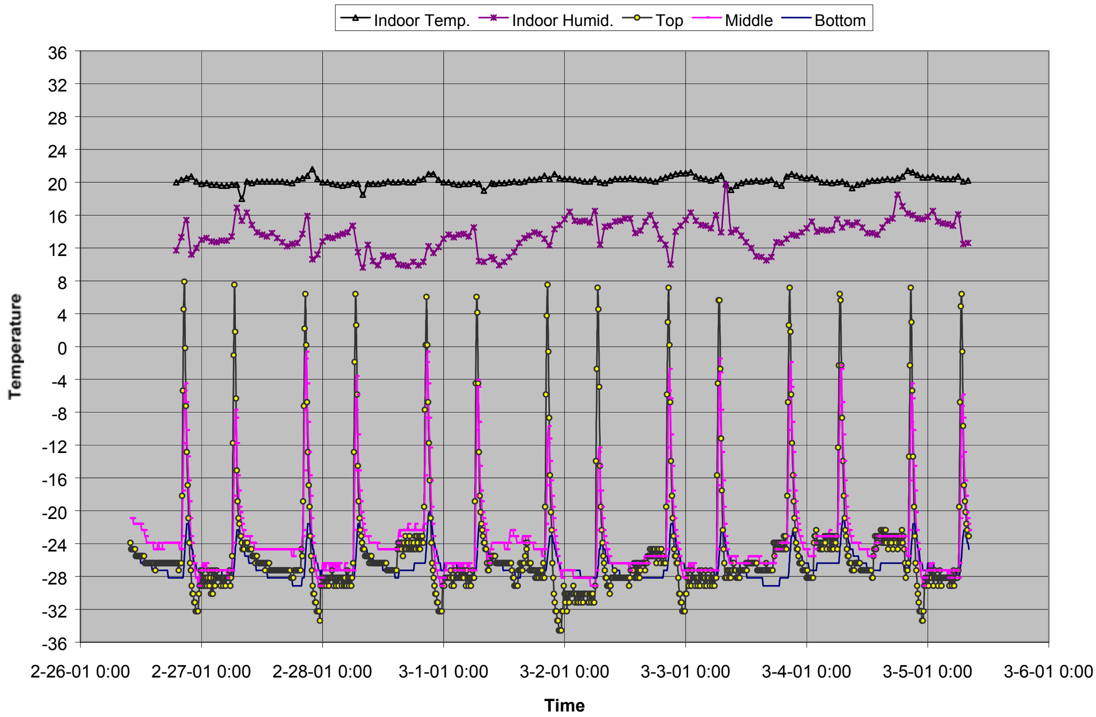


Figure 5a. DX Cabinet Temperature Overview

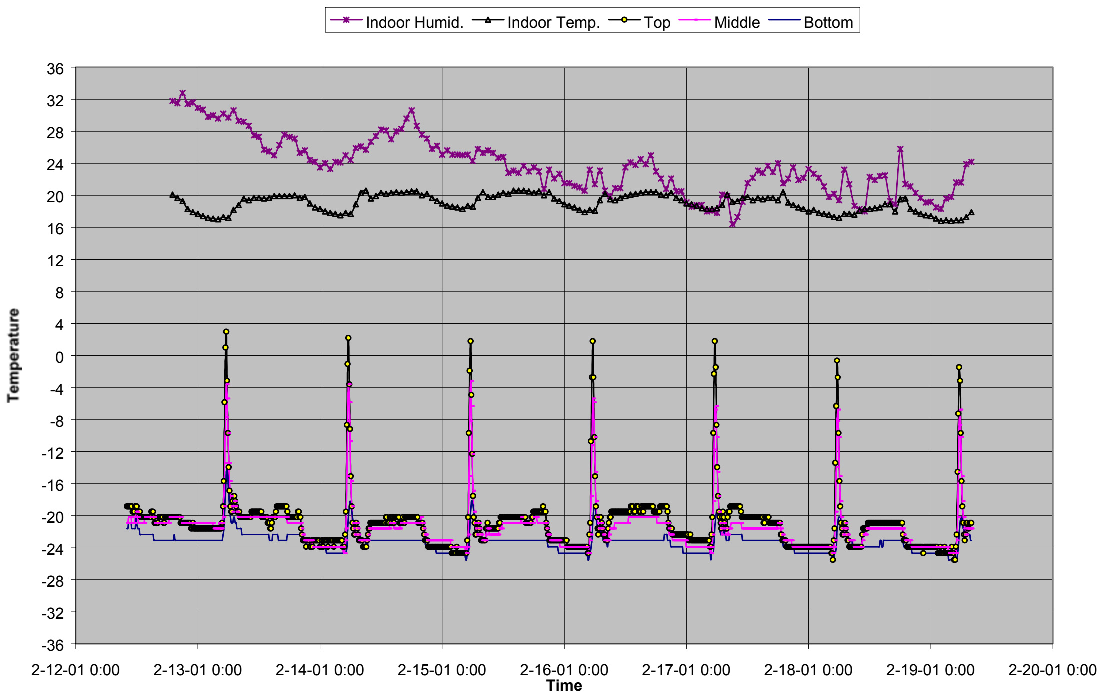


Figure 5b. SQD Cabinet Temperature Overview

The figures 5a and 5b show the overall indoor temperature, moisture, and the temperatures in the cabinets at bottom, middle, top positions along the 7 days test period. (These three temperatures are called as bottom, middle and top temperature in the rest part of the thesis.) It can be seen that the indoor temperature almost remained constant and the changing of indoor moisture had no influence on cabinet food temperatures for the both two cases. Because of the introducing of new technology, the defrosting frequency is reduced to once per day in SQD system, instead of twice per day in DX system and the frequency of significant temperature lifting is consequently reduced to half in SQD system. Less considerable temperature changing contribute to better food quality during the same storage period.

The influence of using night cover can also be clearly seen from the figures. The night covers are normally put on during the close time for the two supermarkets. In order to test the effect, the night covers were designed not to use on Monday evening within each test period for the comparing. However, due to the insufficient cooperation from the supermarket manager, only the SQD night covers were not used on Monday, while the same test was arranged on Saturday evening in DX cabinets. According to the test results, the food temperatures in the cabinets are much higher when the night cover were not put on during the evening. The compressor energy saving can also be achieved from using night covers, and this will be detailed discussed in the later part.

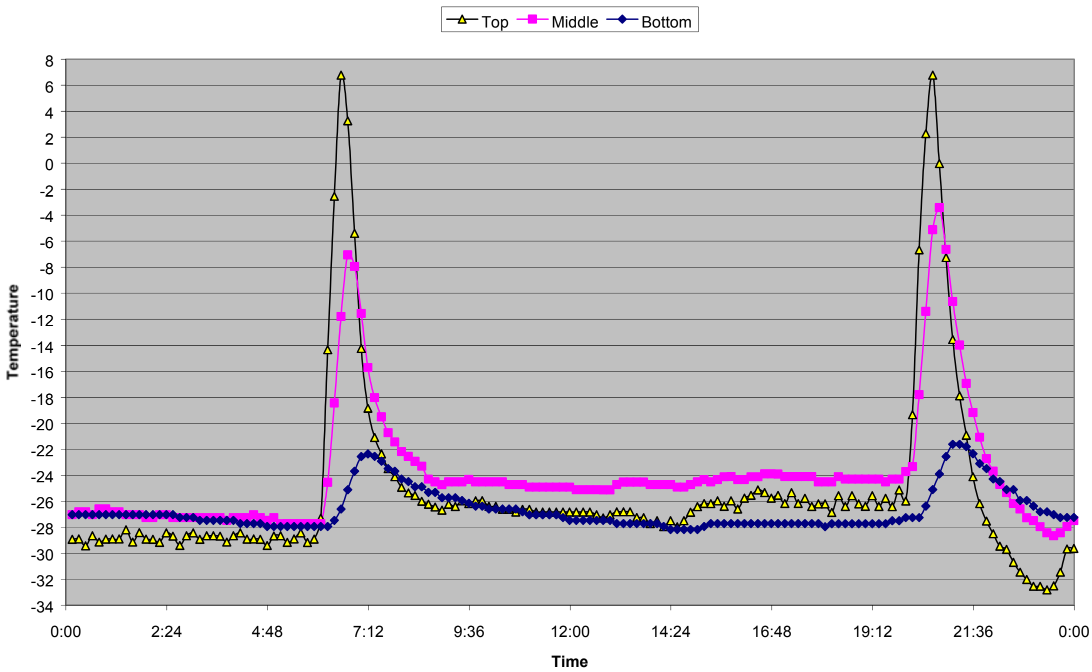


Figure 6a. DX Tinytalk Average Temperature

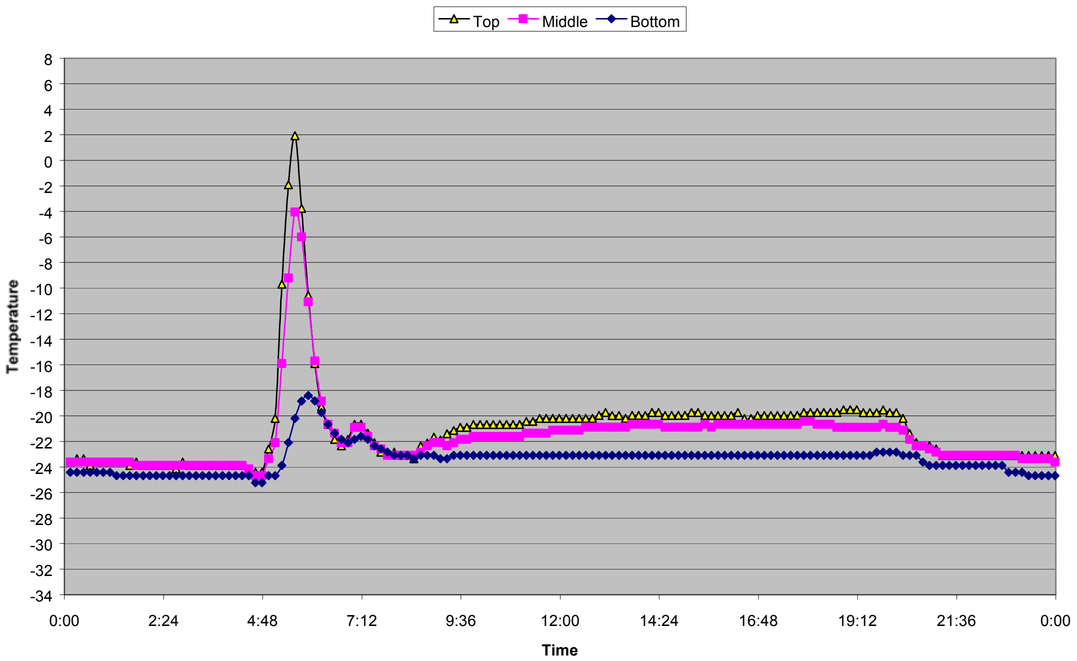


Figure 6b. SQD Tinytalk Average Temperature

The figures present the 24 hours average temperatures for tinytalk logger at different cabinet position. Because of the varied night cover test time, the DX average temperatures were calculated based on four days from Tuesday to Friday, while the SQD average temperatures were taken from Wednesday to Friday for three days. The temperature data in weekend were not counted into the average due to the different operation time on weekend.

Theoretically, the food temperature should decrease from top to bottom in frozen cabinets because of the decreasing of cold losses. However, the measuring results show that the DX top temperature is lower than middle's during the daytime, and even lower than bottom temperature in the night. It was mainly caused by the insufficient food load in DX cabinets, which made the top temperature logger expose in the cold air coming from the cabinet air outlet. Therefore, the logged data at top position in DX cabinets reflected cold air outlet temperature more than the top food temperature. While, the SQD top temperature curve is higher than the middle temperature curve during the operating time and it overlap on the middle curve in the evening. The main reason is the using of night cover decrease the infiltration and radiation losses.

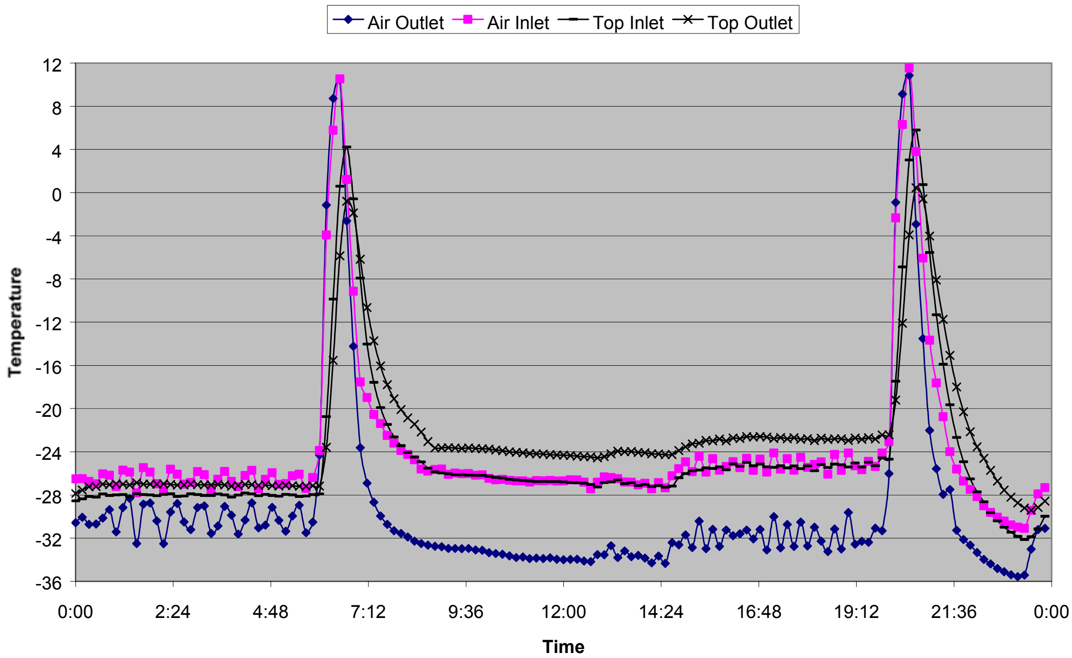


Figure 7a. DX Woodly Average Temperature

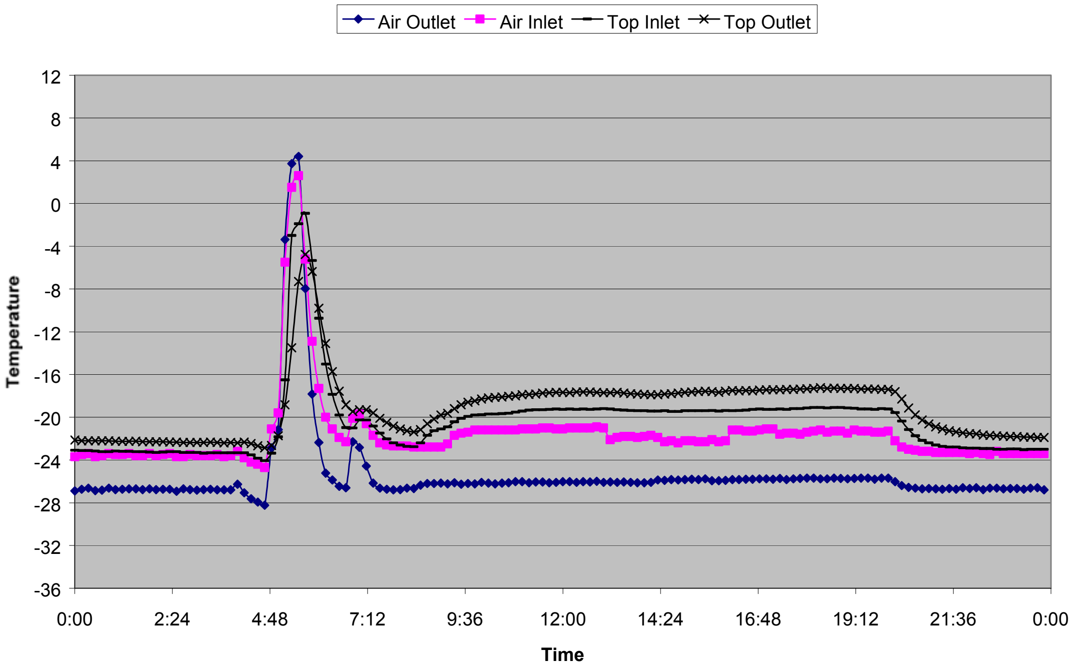


Figure 7b. SQD Woodly Average Temperature

The figures 7a and 7b present the curves from woodly recorded data. The same periods were used in both tinytalk and woodly average calculations. The two "top outlet" and "top inlet" probes were placed at the level of cabinet highest food load line and covered by plasticine to simulate the temperature changing time delay. In figure 7b, the two dummy food temperatures (top outlet, top inlet) are higher than air (outlet, inlet) temperatures all the time. This can be explained by heat is delivered from source (food) to sink (air). The effect of using night cover can be seen from the clear temperature drop at all test positions around 8:30 PM. The dummy food group temperatures go closer to the air group temperatures during the night because of the less cold losses. The top inlet dummy food has the same temperature as air inlet in DX test cabinet in the daytime, and both two food temperature curves go into the range of air temperatures at the night. These results again proof the influence of insufficient food load. The effect of temperature changing caused by the use of night cover is covered by the second defrost cycle which happens also after the supermarket closing time, but such influence can be seen from the temperature drops during the rest of night.

4.2 RESULTS FROM MACHINE ROOM

ETM, a commercial refrigeration measure instrument, was mainly used for monitoring and recording the system performance. Following are the main test results. All the numbers shown in table 1, 2 and the temperature data in table 3, 4 were achieved from 7 days based average calculation.

Table 1. System Pressures and Power Input

	Low Pressure	High Pressure	Power
	P abs (bar)		E (KW)
DX	1.409	16.081	6.252
SQD	1.591	16.339	7.832

Table 2. System Temperatures

	DX	SQD
	T (°C)	
Suction Line	-12.511	-28.965
Discharge Line	89.156	75.146
		Subcool Out
Condenser Out	32.514	-5.554

Following are some other data that were measured and used for further system efficiency calculation.

Table 3. Condensing Data on Dry Cooler Side

	Volume Flow	Inlet	Outlet	Temp. Diff
	In water (l/s)	Propylene Glycol		
DX	0.1	16.363	41.779	25.416
SQD	1.88	32.69	34.536	1.846

Table 4. SQD Temperature Data

	Inlet	Outlet	Temp. Diff	Volume Flow	Mass Flow
	°C			Tem.40 (l/s)	Tem.20 (kg/s)
Sec. Evap.	-27.137	-29.291	2.154	3.1	--
Subcooler	-5.654	0.482	6.136	--	0.321

5 TEMPERATURE PROFILES AND FOODSTUFF QUALITY

5.1 TEMPERATURE COMPARISON

5.1.1 TINYTALK TEST RESULTS

After the previous overview, a serious of comparison had been made for the DX and SQD temperatures at different positions. All the comparisons were based on the average of the weekdays in each test period.

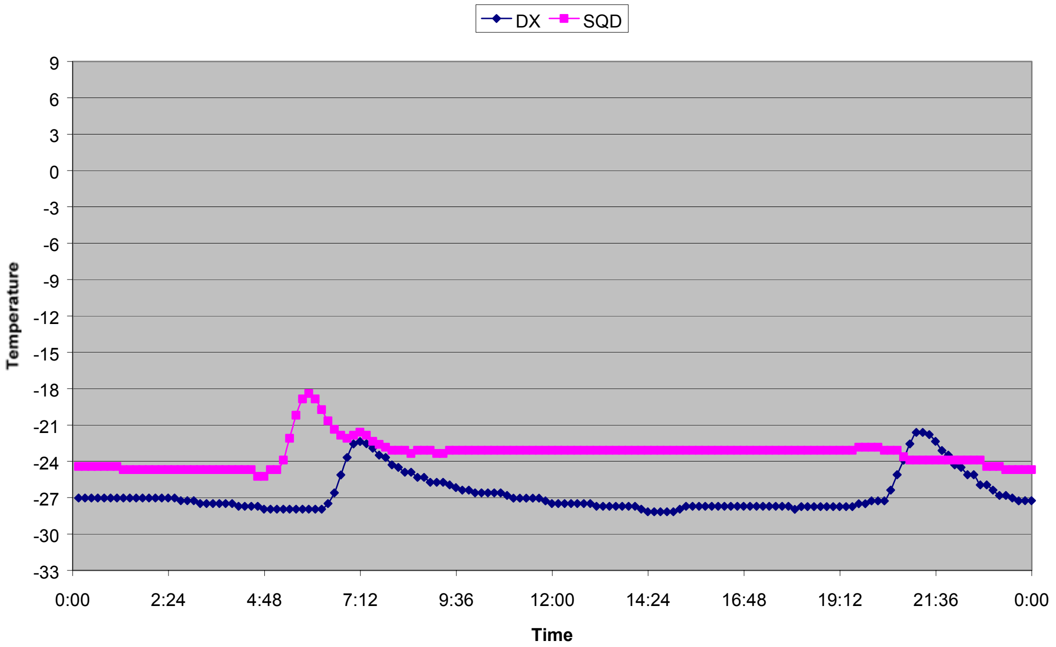


Figure 8a. Tinytalk Bottom Temperature Comparison

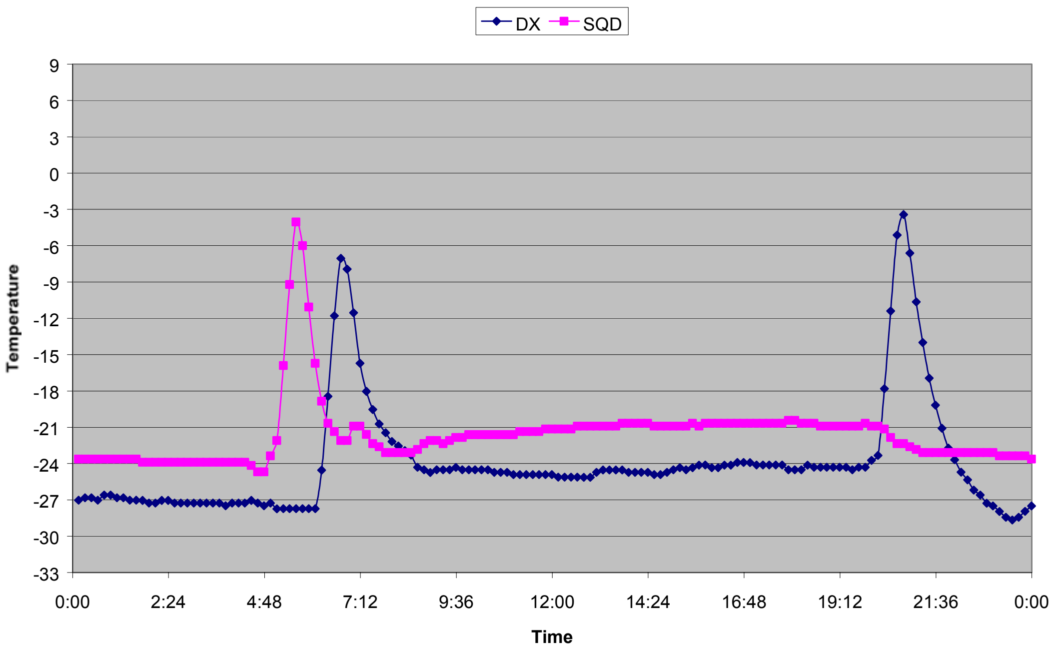


Figure 8b. Tinytalk Middle Temperature Comparison

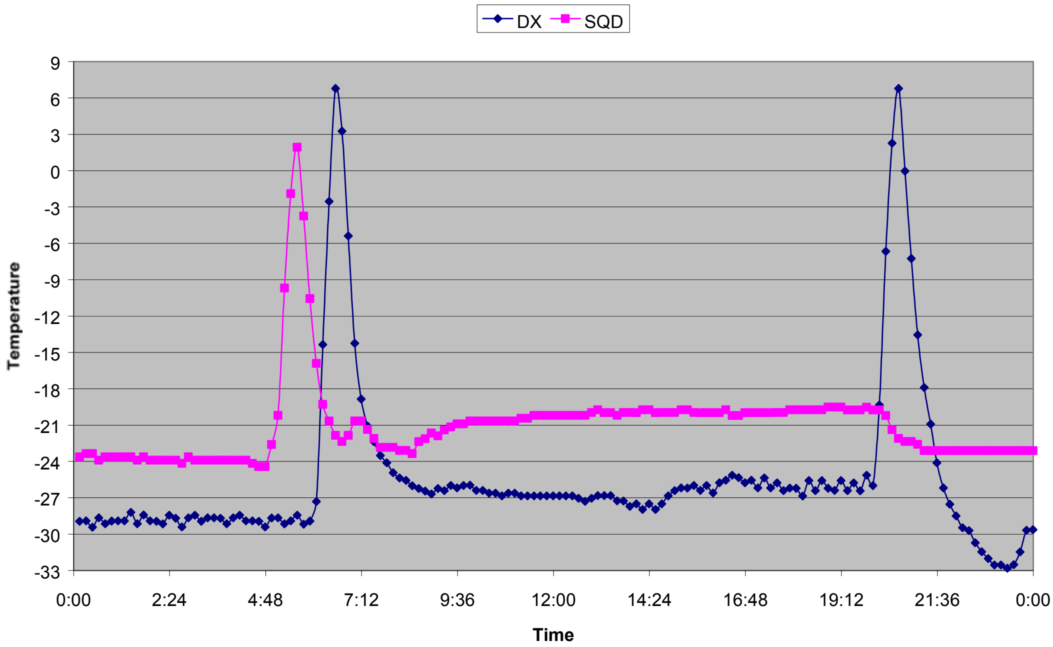


Figure 8c. Tinytalk Top Temperature Comparison

Figure 8a 8b and 8c show the temperature comparison at the three positions in the cabinets. The curves indicate more even temperature distribution can be achieved in SQD cabinets. This is mainly caused by the using of single-phase secondary refrigerant. Therefore, providing uniform storing conditions can be attributed to an advantage of indirect systems. Theoretically, higher single-phase secondary refrigerant specific heat capacity will lead to more even temperature distribution.

Another conclusion is that DX cabinet temperature is much lower than SQD's at all the test positions. A tinytalk average temperature based calculation is shown in table 5.

Table 5. Tinytalk Average Temperatures at Different Cabinet Positions

Average	Bottom	Middle	Top
	Temperature (˚C)		
DX	-26.7	-23.9	-25.3
SQD	-23.4	-21.6	-21.0
Temp. Diff	3.3	2.3	4.3

The results indicate that the tinytalk mean temperatures at each position for DX cabinet are around 2 ~ 4 °C lower than SQD mean temperatures.

5.1.2 PLASTIC CHICKEN TEST RESULTS

In order to make a comparison, the first short period test results are also presented in figure 9a 9b and 9c. The different points are that the first tests were made by plastic chicken temperature loggers and the test cabinet sections were better loaded with real food.

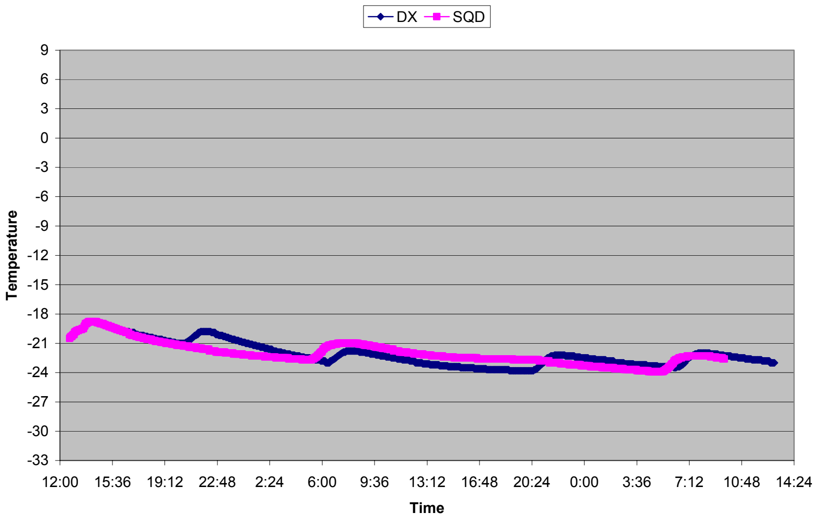


Figure 9a. Plastic Chicken Bottom Temperature Comparison

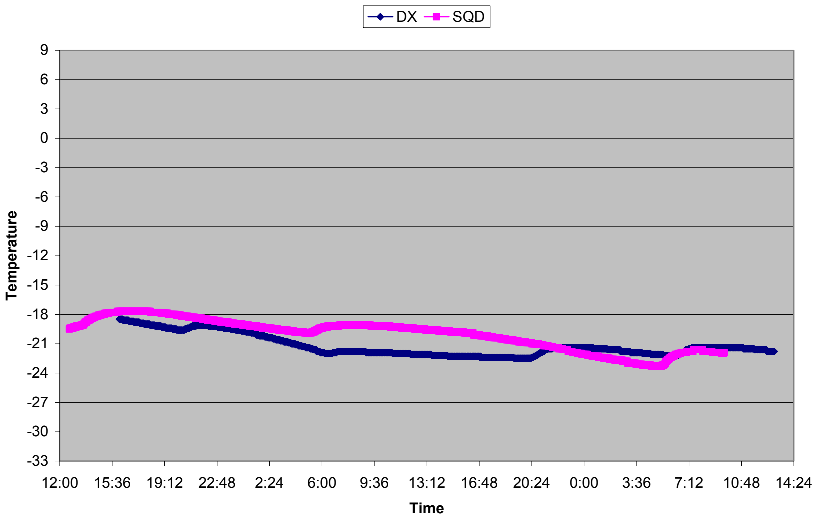


Figure 9b. Plastic Chicken Middle Temperature Comparison

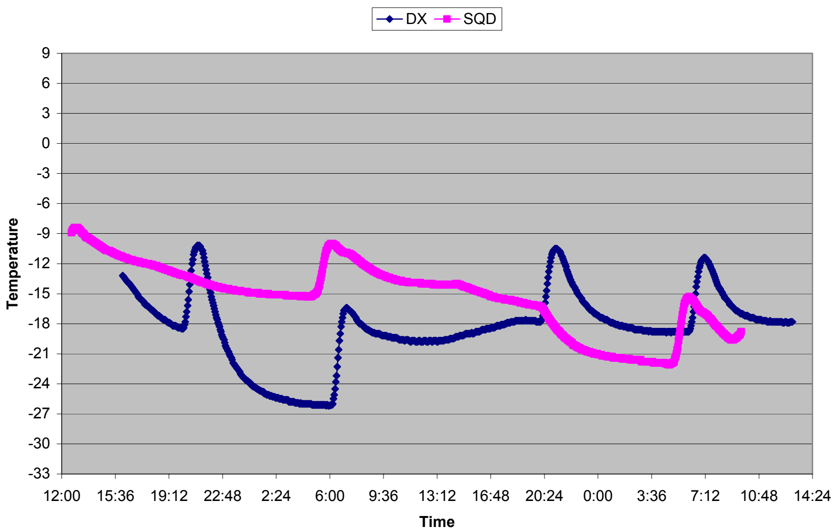


Figure 9c. Plastic Chicken Top Temperature Comparison

It can be seen from figure 9c that lowest top position food temperatures appeared at different points in two systems, and it was caused by the different night cover test period. In the first plastic chicken test, the effect of not using night cover was tested in the second night during the DX system test period, while the same test was arranged to be done in the first night of the SQD system test period. The plastic chicken average temperatures at different cabinet positions were also calculated in table 6.

Table 6. Plastic Chicken Average Temperatures at Different Cabinet Positions

Average	Bottom	Middle	Top.
	Temperature (˚C)		
DX	-22.3	-21.2	-18.4
SQD	-22.0	-20.0	-15.3
Temp. Diff	0.3	1.2	3.1

5.1.3 DISCUSSION

The same conclusion can be made from both plastic chicken and tinytalk test results is that DX cabinet temperatures are lower than SQD's at all test positions. In the plastic chicken case, the temperature differences between the systems vary from 0.3 to 3.1 ˚C. Smaller differences at bottom and middle positions (compared with tinytalk results) may be caused by varied test durations, different (indoor, outdoor) environmental conditions as well as man-made influences. Plastic chicken test top food temperatures are much higher than tinytalk's in both DX and SQD cases, and this is mainly caused by the different food load conditions. (See pictures in Appendix D for comparison) Due to the insufficient food load within the second longer test period, tinytalk logged data more reflect cabinet cooling air temperatures than food temperatures at the top positions. The varied instrument design mechanisms may also influence the test results.

According to the Swedish regulation, the frozen food temperature should not exceed –18 °C. maintaining unnecessary low temperature in the cabinets will lead to refrigeration system extra energy consumption. According to both of the test results, DX cabinet temperature is excessively maintained. However, the DX system temperature control was set in the correct range. (Air outlet control in the range of –25 ~ –28 °C) The reason for the large cabinet temperature difference is caused by the varied system control mechanism. Indirect suction line pressure control is used in DX system. The air temperature signals will control the magnetic valve's on/off stage. The valve is placed on the central refrigerant inlet pipe (see figure 4a, DX system configuration). The temperature signals are coming from each cabinet section's air outlet within the whole group. However, only the worst section's temperatures really control the valve's operation. Therefore, if only there exist load difference, the magnetic valve will keep open to serve the worst section's cooling demand, while most of the rest sections already reach the preset condition. And that is why DX system provides excessive cooling to the whole cabinet group that lead to unnecessary low cabinet temperature. On the contrary, the compressor control signal is coming from secondary refrigerant outlet on the central pipe in the machine room (see figure 4b, SQD system configuration). Such signal can exactly reflect the cooling load for the whole cabinet groups, instead of any individual section. One advantage of SQD control mechanism is that the prescribed temperature can be maintained by the minimum energy consumption.

5.1.4 FURTHER ANALYSIS

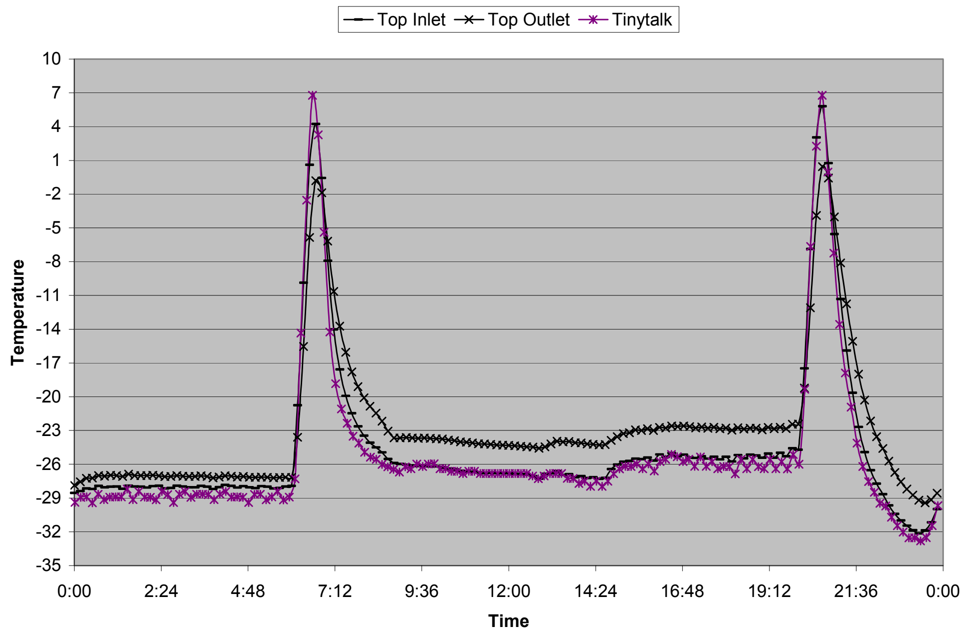


Figure 10a. DX Top Position Temperatures

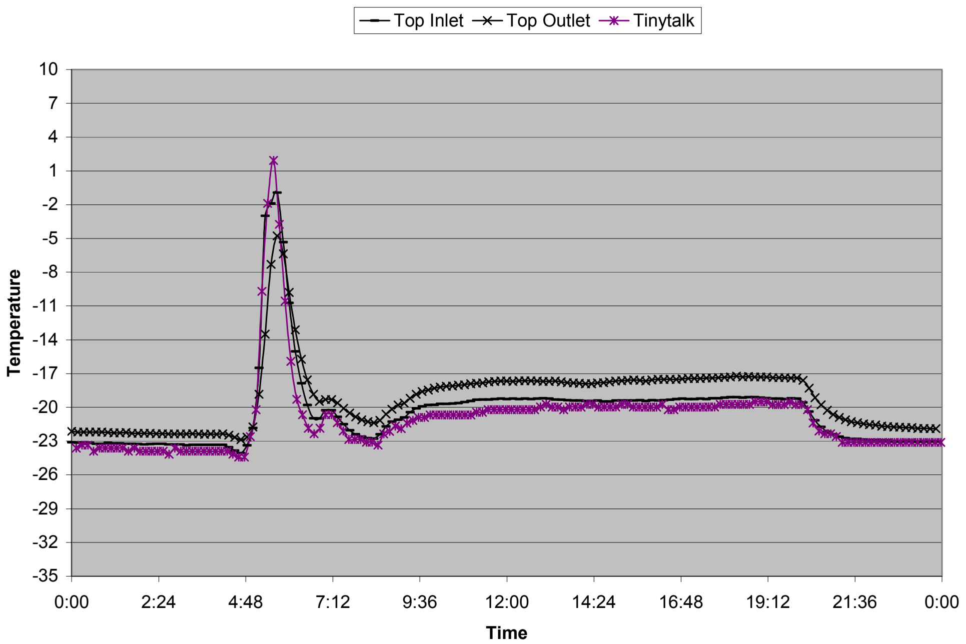


Figure 10b. SQD Top Position Temperatures

Dummy food temperatures at top outlet (4), top middle [3], and top inlet (3) for DX and SQD systems are presented in figure 10a and 10b individually. Top outlet and top inlet should have lowest and highest temperature theoretically. But the test results show that the middle part has the lowest temperature while the outlet part has the highest one. This temperature distribution is clearer in the evening when less loss influences the results. Such effect can be explained from cabinet constructing point of view. Due to the direction of the cold air orientation board, the out-coming cold air could not reach top outlet part that locates just under the board. Although there are some holes opened close to the top outlet part, the out-going air is too little to maintain the low temperature. On the contrary, the top inlet position is totally covered by the in coming cold airflow, and that is why it has even lower temperature than top outlet. Such temperature difference is more definite in figure 11a.

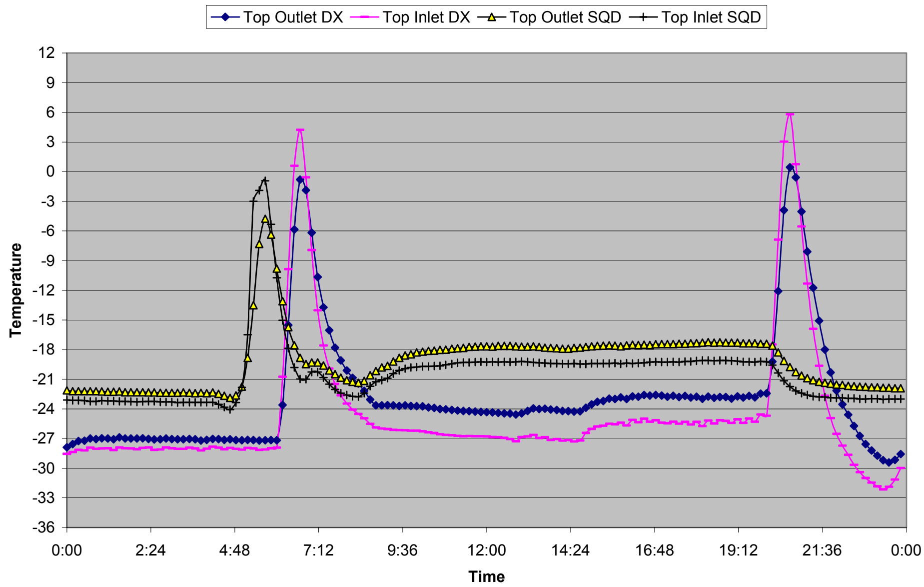


Figure 11a. Top Position Dummy Food Temperature Comparison

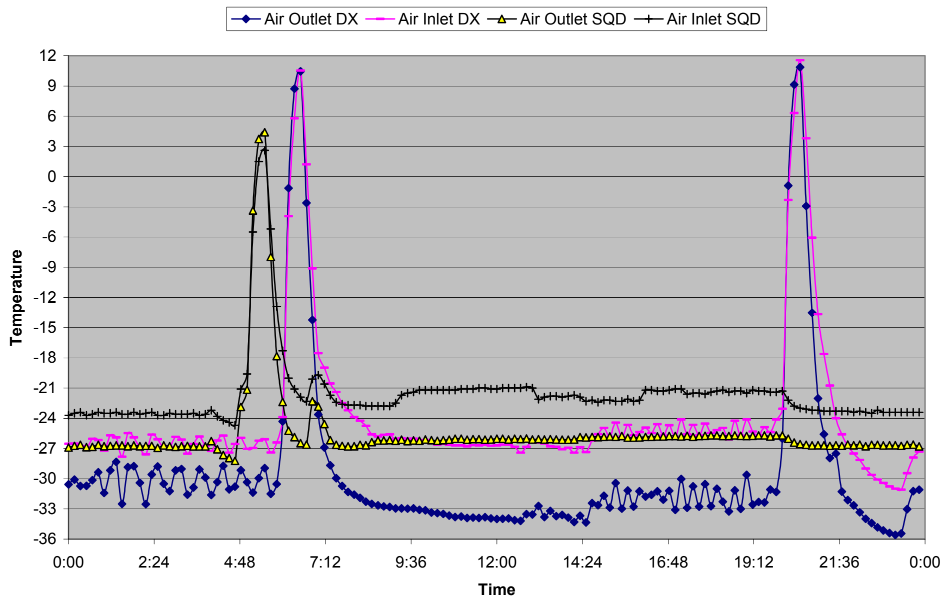


Figure 11b. Air Temperature Comparison

Figure 11b indicates the air temperatures in two test cabinets. SQD air outlet temperature almost equals to DX air inlet temperature all the time. Much energy is saved in SQD system because of the higher temperature distribution range. The character of SQD cabinet temperature even distribution can also been clearly seen from the curves.

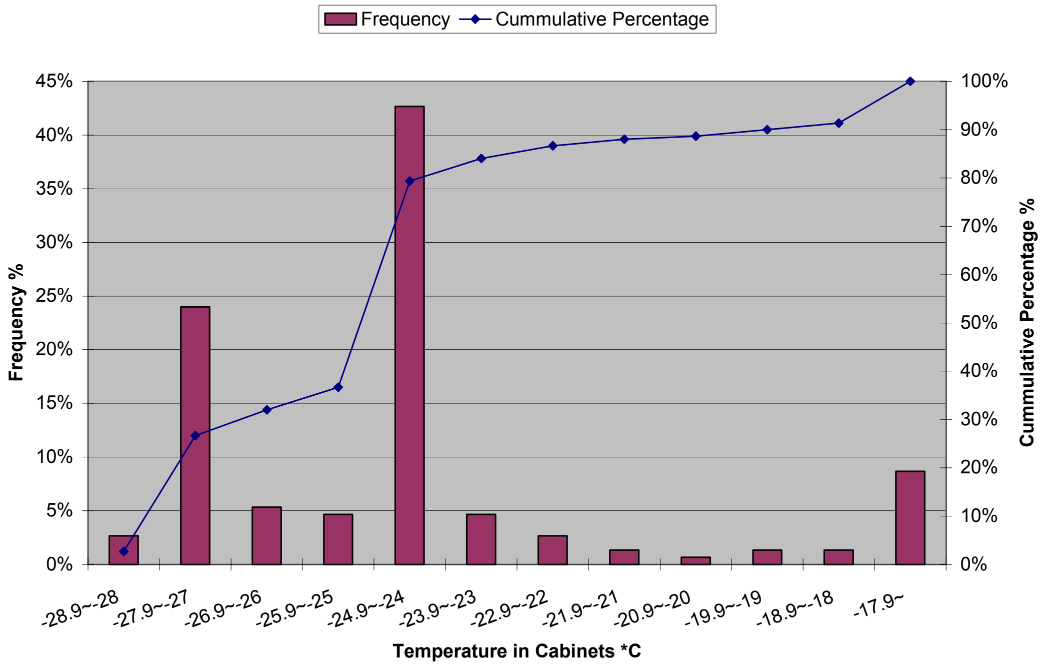


Figure 12a. DX Middle Temperature Distribution

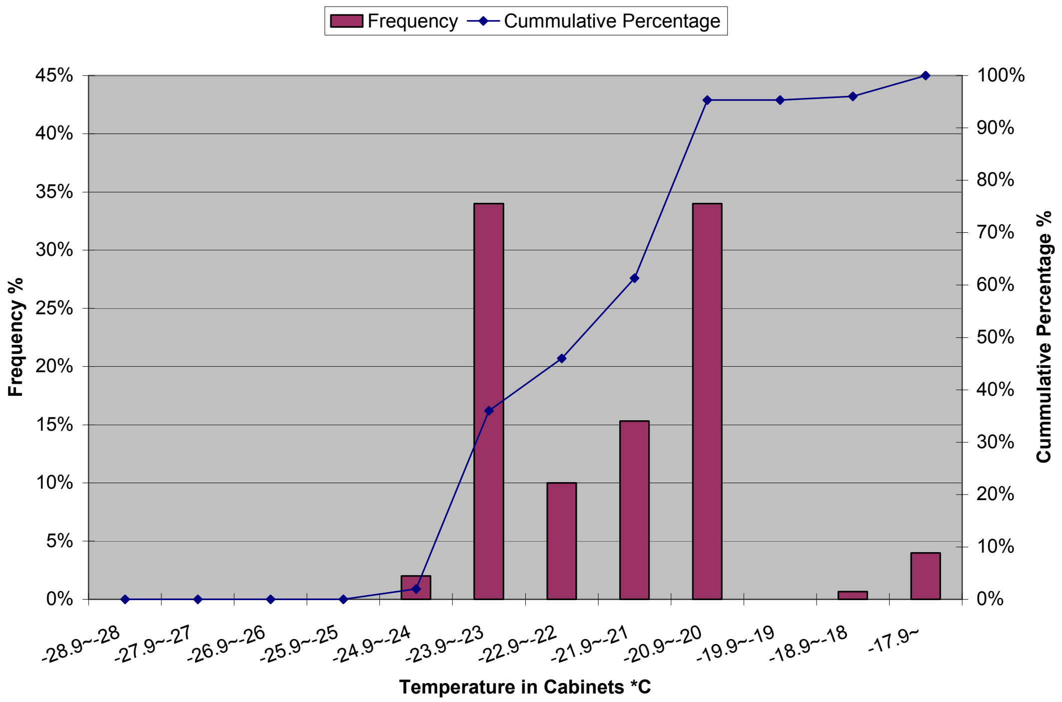


Figure 12b. SQD Middle Temperature Distribution

Figure 12a and 12b were created for indicating the middle position temperature distributions in the two cases. According to the figures, the products in DX cabinets are mainly stored in the temperature range from –28 °C to –24 °C, while SQD cabinets keep food stored in temperature range from –24 °C to –20 °C. The time that products were stored at -18 °C or less is almost the same in two system cabinets within the test period, more high temperature distribution (above –18 °C) in DX cabinet may be caused by the extra defrosting. Keep in mind that such result was achieved by much higher secondary refrigerant temperature in SQD system. (R404A evaporate at –38 °C in DX cabinet evaporator, while the mean secondary refrigerant temperature for cabinet evaporator in and outlet is –27.3 °C in SQD system)

A relevant laboratory test was found for further improving the benefits of secondary coolant. Within the test, dummy food temperatures were logged as a function of the refrigerant inlet temperature to the heat exchanger in the cabinet for R404A and Potassium Formate. The result shows that the same product temperature could be achieved by significantly higher temperature of the secondary refrigerant at the evaporator inlet. [4] This is mainly caused by the better heat transfer performance and higher heat capacity. The SQD system used secondary refrigerant, Temper, also has excellent thermal properties. It is not so hard to deduce that a higher evaporator inlet temperature can also be achieved in SQD cabinets, if the food temperature is maintained on the same level in both DX and SQD cabinets.

5.2 FOODSTUFF QUALITY

Keep product in good quality and store them as long as possible are the purpose of frozen food storage. However, chemical process is continuously going on along the time. High Quality Life (HQL) is therefore introduced to calculate product quality and storage time. HQL is defined as the time of storage that can be accepted before the first signs can be detected of any deterioration in taste or quality. [1]

HQL is decided by storage temperature which may have large difference along the cold chain from manufacturers to consumers. One typical cold chain for the distribution of frozen products is presented in figure 13.

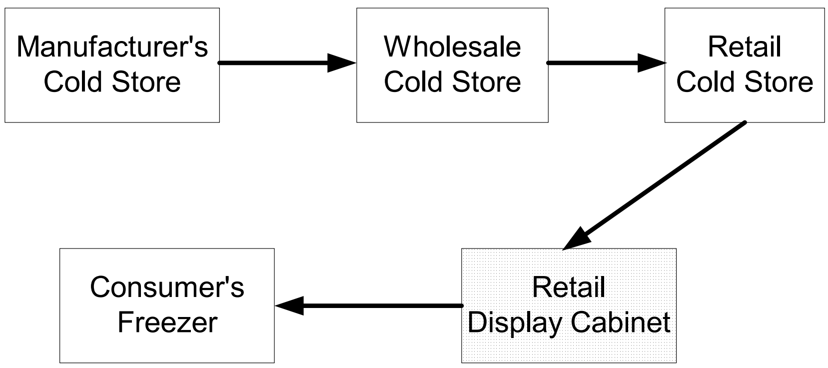


Figure 13. Schematic of Frozen Products Distribution Cold Chain

Focusing on retail display cabinet, HQL was calculated for the test DX and SQD cabinets. Temperatures at cabinet middle position were taken as examples for representing the mean temperature in each cabinet. The following equation used beef’s HQL as an example.

According to the figure in [1]

$$HQL_{beef} \cong 400 \times \exp\left(\frac{-18-t}{5.36}\right)$$

Where t is beef storage temperature

If beef is put into DX or SQD cabinets, it will have following High Quality Life.

t_{DX}= -23.9 °C $HQL_{b.\ DX}$= 1202 days
t_{SQD}= -21.6°C $HQL_{b.\ SQD}$= 782 days

The results can only provide a general idea about the storage period. Although the DX high quality life is 1.5 times than SQD's, the daily large temperature changing caused by different defrosts can not be reflected from the calculation, and such temperature lift will have significant influence on product quality.

6 REFRIGERANTS CHARGING AMOUNT COMPARISON

Table 7. Refrigerant Capital Cost for different Systems

Direct System (DX)				
	Ref. Type	Amount	Unit Price	Total Price
		Kg	SEK/kg	SEK
Primary Cycle	R404A	18	352	6 336
x3.26				
		58.68	Total	20 656
Indirect System (SQD)				
	Ref. Type	Amount	Unit Price	Total Price
		Kg	SEK/kg	SEK
Primary Cycle	R404A	7	352	2 464
Sec. Eva. Cycle	Temper 40	732	76	55 632
			Total	58 096

(Bulk) Price Sources:
R404A – AHLSELL, 1999
Temper # – ONINNEN PRO KYL, 1999

Table 7 shows that the current price of charging such an indirect system (58 096 SEK) is around 3 time more than direct system's (20 656 SEK), if the comparison based on the same capacity. Although a higher cost is got from the calculation, in practice, the indirect system charging cost can be reduced to a competitive position due to the discount based on the larger scale purchase. Within the example, the required amount for Temper® 40, a type of proprietary coolant, is almost 14 times more than required R404A for the same system capacity.

Moreover, indirect systems are more competent in case of accidental leakage. A direct system is one pressurized close circuit, and the normal primary refrigerant is in vapor phase when exposed to atmosphere. If leakage happens, direct systems will easily loose their entire charged refrigerant within a short period. While indirect systems are composed of two separated cycles. The charging amount of primary refrigerant is much smaller. Temper and similar kinds of secondary refrigerants are in liquid form. Therefore, only a small amount of refrigerant will be lost before leakage can be detected. Hence, the capital cost can be reduced for indirect systems from maintenance point of view. And smaller leakage will lead to less environmental impact.

7 COMPRESSION CYCLE ANALYSIS

7.1 COEFFICIENT OF PERFORMANCE (COP)

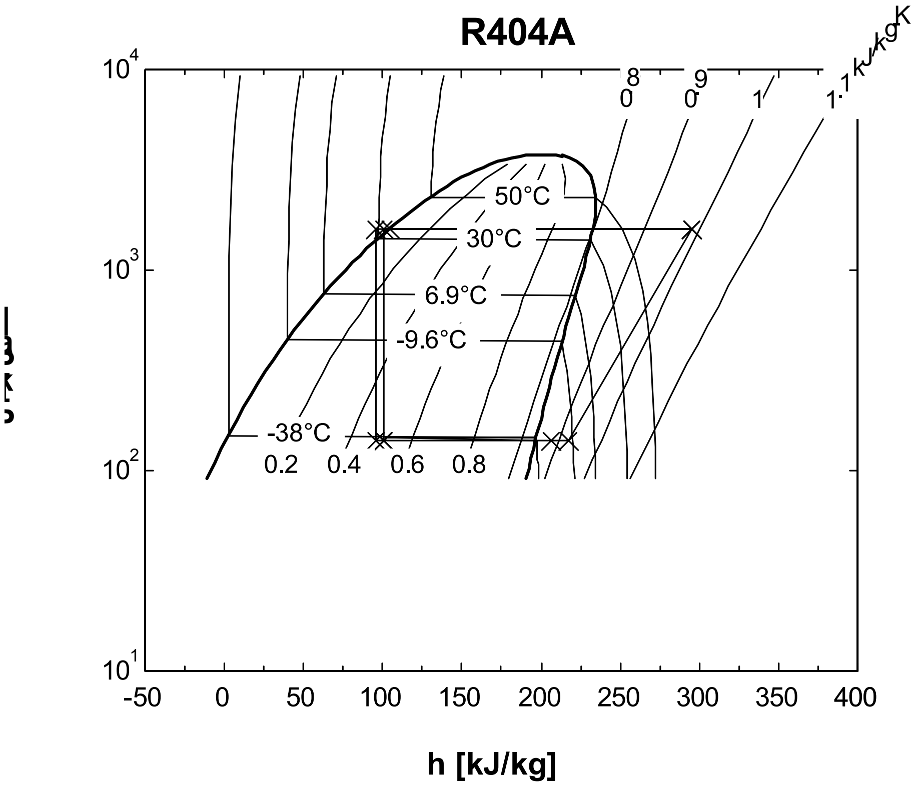


Figure 14a. DX Compression Cycle on P-H diagram

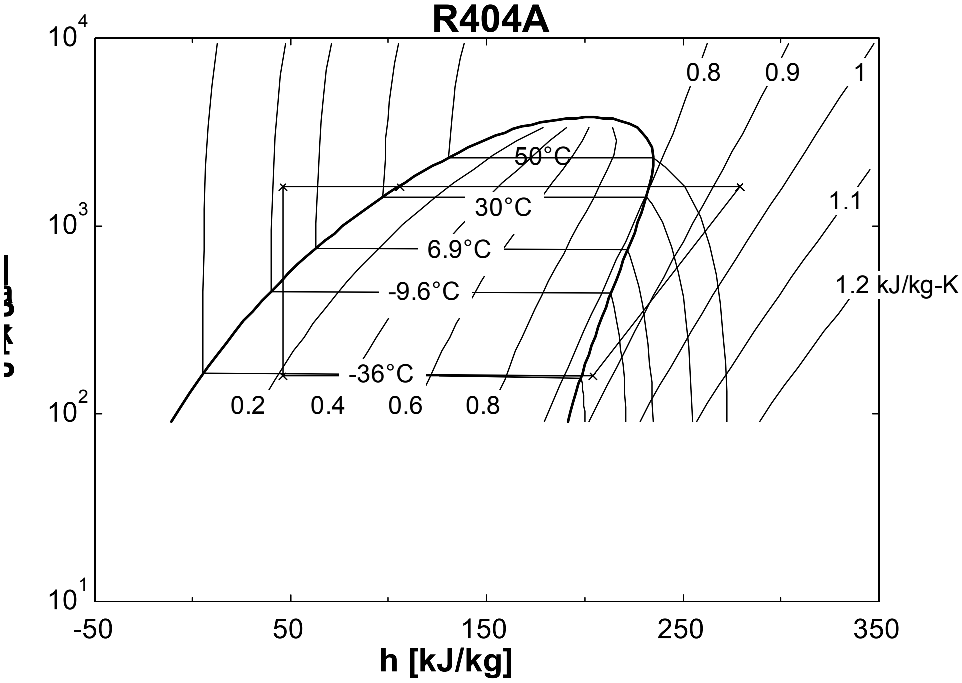


Figure 14b. SQD Compression Cycle on P-H diagram

Compression cycles were drawn and COPs were calculated on basis of ETM recorded data. A software, Engineering Equation Solver (EES), was used for programming the COP calculations. (See Appendix B for the programs) The basic formula $COP_2 = \frac{h_{2k} - h_{1s}}{h_{1k} - h_{2k}}$ was used for calculating. Due to the long distance between machine room and shopping area, temperature drops exist along the refrigerant deliver pipes. Too much pipeline cold loss will decrease system performance. The losses were measured and the results are presented in table 8.

Table 8. DX and SQD Pipeline Temperature Drop

Temperature (°C)		
DX	Comp. Inlet	-12.511
	Evap. Outlet	-25.598
	Temp. Diff.	13.087
	Cond. Outlet	32.514
	Bef. Exp. Val	28.824
	Temp. Diff.	3.69
Sec. Evap. Cycle		
SQD	Evap. Inlet	-27.137
	Cab. Outlet	-26.903
	Temp. Diff.	-0.234
	Evap. Outlet	-29.291
	Cab. Inlet	-27.7
	Temp. Diff.	1.591

It can be seen from table 8 that SQD pipeline temperature drop is much smaller than DX's, this is caused by the use of double layer insulation in SQD refrigerant deliver pipes. The test evaporator inlet point reflects the temperature of refrigerant coming from group A, B and C, while the cabinet outlet point only represents the refrigerant outlet temperature at group A, and that why a minus temperature drop was got from the table.

Modified COP_2s were also calculated for two systems. In DX modification, evaporator outlet and before expansion valve temperature were used to count the system cooling capacity. While the SQD cooling capacity was calculated depending on the formula, $Q_2 = C_{40} \times m_{40} \times \Delta T_{40}$. All the data were gathered from the central pipes of secondary evaporating cycle in the machine room. Instead of using ETM data, the SQD compressor power input was calculated based on the formula, $E_{comp.} = m \times \Delta h$, because the ETM measured power supply is expected including approximately 5 ~ 10 % loss which loose to the surrounding. [8]

DX

Original COP_2

$$COP_2 = \frac{h_{2k} - h_{1s}}{h_{1k} - h_{2k}} = \frac{217.6 - 100.9}{294.9 - 217.6} = 1.509$$

Real COP_2

$$COP_{2real} = \frac{h_{2kr} - h_{1sr}}{h_{1k} - h_{2k}} = \frac{206.9 - 95.27}{294.9 - 217.6} = 1.444$$

m_{R404A}=247 kg/h=0.0686 kg/s (manufacturer data)

SQD

Original COP_2

$$COP_2 = \frac{h_{2k} - h_{1s}}{h_{1k} - h_{2k}} = \frac{203.7 - 46.09}{279.1 - 203.7} = 2.091$$

$$Q_2 = m \times \Delta h = 0.0964 \times (203.7 - 46.09) \times 2 = 30.387 \text{ kW}$$

Real COP_2

$$COP_{2real} = \frac{C_{40} \times m_{40} \times \Delta T_{40}}{m_{R404A} \times (h_{1k} - h_{2k})} = \frac{21.246}{0.0964 \times (279.1 - 203.7) \times 2} = 1.46$$

m_{R404A}=347 kg/h=0.0964 kg/s (manufacturer data)

Table 9. DX and SQD Compressor Power Comparison

Comp. Power (kW)	ETM	$m \times \Delta h$
DX $E_{comp.}$	6.252	5.303
SQD $E_{comp.}$	7.832	7.269

$COP_{2.\ DX}$= 1.444 $COP_{2.\ DX\ Original}$ = 1.509
$COP_{2.\ SQD}$= 1.460 $COP_{2.\ SQD\ Original}$ = 2.091

No distinct difference exists between DX original and real COP_2 because temperature drops occur on both liquid and suction line in DX compression cycle. While SQD real COP_2 is lower than original one, and this is mainly caused by the use of modified cooling capacity. The Q_2 calculated in original SQD COP_2 could be treated as the cooling capacity provided by the primary compression cycle. Therefore an energy loss in the form of cooling capacity decrease (around 9 kW in this case) does exist in indirect systems, because of the introduction of one extra heat exchanger.

But the SQD COP_2 can not be used to judge the system performance, because two extra subcoolers are introduced in SQD system for further cooling the refrigerant after the primary condensers. The subcooler consumed energy is coming from a nearby middle temperature refrigeration system, and such energy consumption can not be reflected by COP calculation. But in general, SQD system still has higher system efficiency, because of the similar condensing temperature (around +34 °C) and the higher evaporating temperature (-36 °C for SQD and –38 °C for DX).

7.2 COMPRESSOR ENERGY CONSUMPTION COMPARISON

The compressor energy consumption was calculated on the basis of ETM logged compressor power input data. The time interval for logging was set at every one minute in both cases. Due to the different night cover test time and supermarket operation time through one week, the weekday's and weekend's energy consumption were calculated separately. The purpose of calculation is to reflect two systems' average weekly compressor energy consumption.

DX System

Energy consumption for four days from Tuesday to Friday

W= 489.537 kWh

Average energy consumption for each weekday

$W_{DX.\,d}$= 122.384 kWh

Energy consumption during weekend (without night cover)

Saturday: W= 123.419 kWh

Sunday: W= 121.340 kWh

Total: W= 244.759 kWh

Night cover factor for DX system (got from nigh cover energy consumption calculation)

N= 0.873

Energy consumption during weekend (with night cover)

$W_{DX.\,e}$= Total * N = 213.796 kWh

DX system weekly energy consumption

W_{DX}= $W_{DX.\,d}$ * 5+ $W_{DX.\,e}$= 825.717 kWh

SQD System

Group A

Energy consumption for three days from Wednesday to Friday

W= 346.426 kWh

Average energy consumption for each weekday

$W_{SQD.\,d}$= 115.475 kWh

Energy consumption during weekend

Saturday: W= 115.079 kWh

Sunday: $W = 107.175$ kWh

Total: $W_{SQD.\,e} = 222.254$ kWh

SQD Group A weekly energy consumption

$W_{SQD.\,A} = W_{SQD.\,d} * 5 + W_{SQD.\,e} = 799.630$ kWh

Group B

Two timers were added on SQD system for counting the compressor work time.

A timer: Calculated working time: 101.95 hr

Real working time: 102.14 hr

Accuracy: 0.19%

B timer: Real working time: 116.46 hr

Time factor: T= Time B/Time A=1.1402

Group B weekly energy consumption

$W_{SQD.\,B} = W_{SQD.\,A} * T = 911.738$ kWh

SQD system weekly compressor energy consumption

$W_{SQD} = W_{SQD.\,A} + W_{SQD.\,B} = 1711.368$ kWh

The results should be compared based on the same system capacity. The modification is shown in table 10.

Table 10. DX and SQD Compressor Weekly Energy Consumption Comparison

	DX	SQD
Energy Consu. (kWh) 1.	825.717	1711.368
System cap. (kW)	6.065	19.77
Cap. Factor 2.	3.26	1
Modi. Ener. Consu. (kWh) 1.* 2.	2691.837	1711.368
Energy Saving	0	36.4%

Table 10 shows a compressor energy saving of 36.4% is reached by indirect SQD system. The reasons are mainly contributed to the advanced SQD compressor control mechanism and the usage of high performance secondary refrigerant which can reduce energy consumption caused by the extra temperature difference in the primary evaporator. The extra SQD subcooler energy consumption was not taken into account, which also influence the SQD energy saving potential. One special point is (DX) Sundbyberg supermarket has two more daily open hours than (SQD) Mörby supermarket's, which may reduce the SQD system energy saving potentials, but the indirect system's saving result is still substantial.

Moreover, the above calculation was made only from compressor energy consumption point of view. If take into account defrost and SQD subcooling energy consumption, the two systems' total energy consumptions will be different from compressor energy consumption results, and such analysis was done in the following section.

Outdoor temperature and moisture are the factors which can influence compressor energy consumption. However, the effect can only be seen from a long period, such as several months. Due to the short time schedule of the field test (two weeks), the outdoor climate influence was not considered during the analysis, but such test had been done in another field measurement. And the conclusion is that higher outdoor temperature and moisture will increase the compressor energy consumption in supermarket case. [5]

7.3 ENERGY SAVING POTENTIAL ANALYSIS

In order to reflect the total energy saving potential of test indirect system (SQD), compared with test partially indirect system (DX), the total energy consumptions for two different systems were further analyzed. Weekly total energy consumptions were calculated and all the calculations are based on the logged data and the FRIGOTECH AB provided technical information. Modifications were made in the processing according to the assumed similar cooling capacity (19.77 kW).

DX System

Compressor Weekly Energy Consumption

$W_{comp.\ w}$ = 2 691.837 kWh

Defrost Weekly Energy Consumption

W_d = 20.86 kWh/day

$W_{de.\ w}$ = 20.86 * 3 * 7 = 438.06 kWh

Dry Cooler Weekly Energy Consumption

$W_{dry.\ w}$ = 25 000 * (7/365) = 479.452 kWh

DX System Total Weekly Energy Consumption

$$W_{DX.total} = W_{comp.w} + W_{de.w} + W_{dry.w}$$

$$= 2\,691.837 + 438.06 + 479.452 = 3\,609.3 \text{ kWh}$$

SQD System

Compressor Weekly Energy Consumption

$W_{comp.\ w}$ = 1 711.368 kWh

Subcooler Weekly Energy Consumption

$E_{sc.}$ = 5.510 kW (See Chapter 9.2 COSP)

$W_{sc.\,w} = E_{sc}$ * Load Factor * 24 * 7

= 5.510 * 0.75 * 24 * 7 = 694.26 kWh

Defrost and Circulating Pumps Weekly Energy Consumption

$W_{de.\,d}$ = 0.608 kWh/day

$W_{de.\,w}$ = 0.608 * 7 = 4.256 kWh

E_{cir} = 0.7303 kW (See Chapter 9.2 COSP)

$W_{cir.\,w}$ = 0.7303 *24 * 7 = 122.69 kWh

$$W_{ptotal.w} = W_{de.w} + W_{cir.w}$$

= 4.256 + 122.69 =126.946 kWh

Dry Cooler Weekly Energy Consumption

$W_{dry.\,w}$ = 25 000*(7/365) = 479.452 kWh

SQD System Total Weekly Energy Consumption

$$W_{SQD.total} = W_{comp.w} + W_{sc.w} + W_{ptotal.w} + W_{dry.w}$$

= 1 711.368 + 694.26 + 126.946 + 479.452 = 3 012.0 kWh

Total Energy Saving Potential

$$Saving = \frac{W_{SQD.total} - W_{DX.total}}{W_{DX.total}} \times 100\% = -16.6\%$$

it can be seen from the calculation that the indirect SQD system achieves a 16.6% total energy saving over the partially indirect DX system. The large SQD compressor (as well as SQD defrost) energy saving result is decreased by the big SQD subcooling energy consumption. But SQD system is still superior from less system total energy consumption point of view.

The cabinet lighting and frame heating energy consumption are not counted in the calculation, since they are same in the two systems.

A similar field test was found through the literature survey. The report presents a refrigeration system retrofitting work processed in a Danish supermarket. The existing HCFC plant was replaced by a new indirect two-stage ammonia plant. The power consumption was monitored and corrected before and after the retrofitting. The result shows that a 14% energy saving was obtained when operating with the new indirect ammonia plant. [6] Such test, again, indicates that a large amount of energy can be saved from well-constructed indirect systems, compared with traditional direct expansion systems.

8 DISPLAY CABINET & RELATED ENERGY CONSUMPTION ANALYSIS

8.1 ENERGY BALANCE AND LOSSES

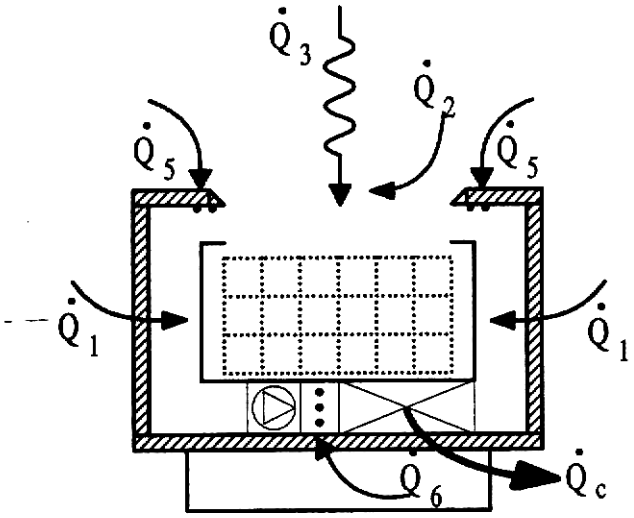


Figure 15. Frozen Cabinet Energy Losses [9]

Q_1: Heat transfer through cabinet walls

$$Q_1 = U_w * A_w * (t_{amb} - t_i)$$

Q_2: Heat transfer by infiltration

$$Q_2 \approx X * \dot{M}_{al} * (h_{amb} - h_{al})$$

Q_3: Radiant heat transfer

$$Q_3 \approx \alpha_{r0} * \varepsilon_e * F_{amb,i} * (t_{amb} - t_i) * A_{vo}$$

[Q_4: Frame heating]

Q_5: Lighting and Fan

Q_6: Defrost

<u>*Energy Balance for Display Cabinet:*</u>

$$Q_c = Q_1 + Q_2 + Q_3 + Q_4 + Q_5 + Q_6$$

Figure 15 indicates the energy balance (losses) for a typical frozen cabinet. All the losses can be minimized from cabinet constructing point of view except infiltration Q_2, Radiation Q_3 and defrost Q_6 losses which may varied a lot from different specifics. Moreover, infiltration and defrost are normally the most energy consuming parts among all the items. A reduction on either of them will lead to a notable energy saving result. The details are discussed in the following two sections.

8.2 FROSTING AND DEFROSTING

Ice form in frozen cabinet easily because of the moisture in air. The ice layer will decrease heat transfer and influence the circulating airflow, which will further increase the compressor power input. A specific research indicates that, compared with traditional R404A direct expansion served cabinet, the cabinet served by secondary coolant (such as potassium formate) has less frost and more uniform frost deposition, because it has higher coil surface temperature and the secondary refrigerant has almost constant heat transfer coefficient along the heat exchanger. [4] These advantages lead to a decrease on both defrost frequency and defrost time. The test SQD cabinets also have the same benefits thanks to the excellent Temper thermal properties. And that is one of the reasons why the defrosting frequency can be set once per day, instead of twice per day for typical DX cabinets.

The test DX cabinets use traditional electric heater to defrost, while SQD cabinets use warm liquid defrost. The cardinal point is that SQD defrost energy is from the waste heat on secondary condensing side. During defrost period, one special cycle is pumped to circulate warm liquid to each cabinet group through the two three-way valves at the inlet and outlet of the group. The energy goes from secondary condensing circuit to defrost circuit through a heat exchanger. Therefore, the only energy consumption is coming from the operating of two circulating pumps during the whole defrost period.

8.2.1 ENERGY CONSUMPTION ANALYSIS

DX and SQD defrost energy consumption data were listed in table 11a and 11b. The DX evening defrost was picked up for comparison, because it got less environment influence, the DX and SQD defrost lasting times were counted from two extra manual set defrosting within the test period. Defrost is terminated by the cabinet air outlet temperature signal (around +7 °C) in DX system, and by cabinet refrigerant outlet temperature signal (around +12 °C) in SQD system.

Table 11a. DX Defrost Energy Consumption

DX	Electrical Heating Defrosting			Cabinet group Cap. 6.065 kW		
Type	Heating Rods	Tested	Heating Cap.	Defro. Time	Energy Consu.	Energy Consu.
		Cabinet Group		From Air Outlet	Per time	Twice per Day
	W		KW	min	KWh	kWh
DLG-11	3x480	1xDLG-11	1.44	--	--	--
FLG-25	2x3x770	2xFLG-25	9.24	--	--	--
FLG-37	2x3x1200	1xFLG-37	7.2	--	--	--
Total	--	--	17.88	35	10.43	20.86

Table 11b. SQD Defrost Energy Consumption

SQD	Warm Liquid Defrosting			Cabinet Group A Cap. 6.090 kW		
SQD Defrost Pump	Current	Voltage	Power	Tested Defro. Time	Energy Consu. Per Group	Energy Consu. Three Groups
	A	V	KW	min	KWh	KWh
Sec. Cond. Side	1.53	218	0.3335	31	0.1723	--
Sec. Evap. Side	0.27	218	0.0589	31	0.0304	--
Total	--	--	--	--	0.2027	0.608

El.Heater in Test SQD Group		Energy Consu.
Type	KW	KWh
2xDLG-11	2.88	--
2xFLG-37	14.4	--
Total	17.28	20.16

It can be seen from the calculation that, with almost the same capacity cabinet groups, traditional electrical heater defrost consume 100 times more energy than warm liquid defrost's. From other point of view, almost 20 kWh is saved per day only from one SQD cabinet group by using warm liquid defrost, instead of electric heater defrost.

8.2.2 TEMPERATURE ANALYSIS

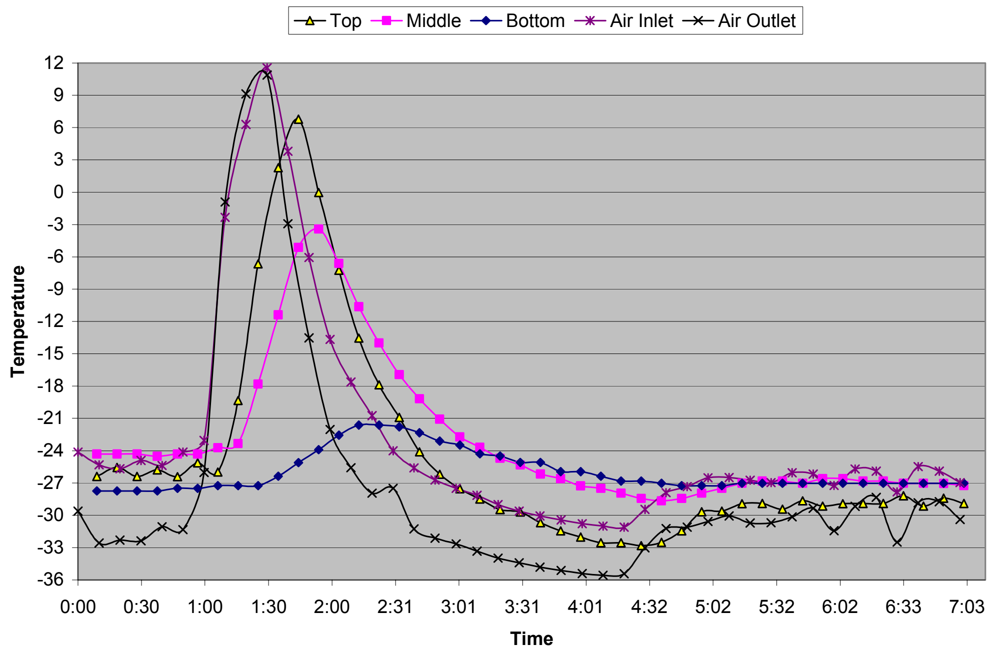


Figure 16a. DX Defrosting Temperature Profile

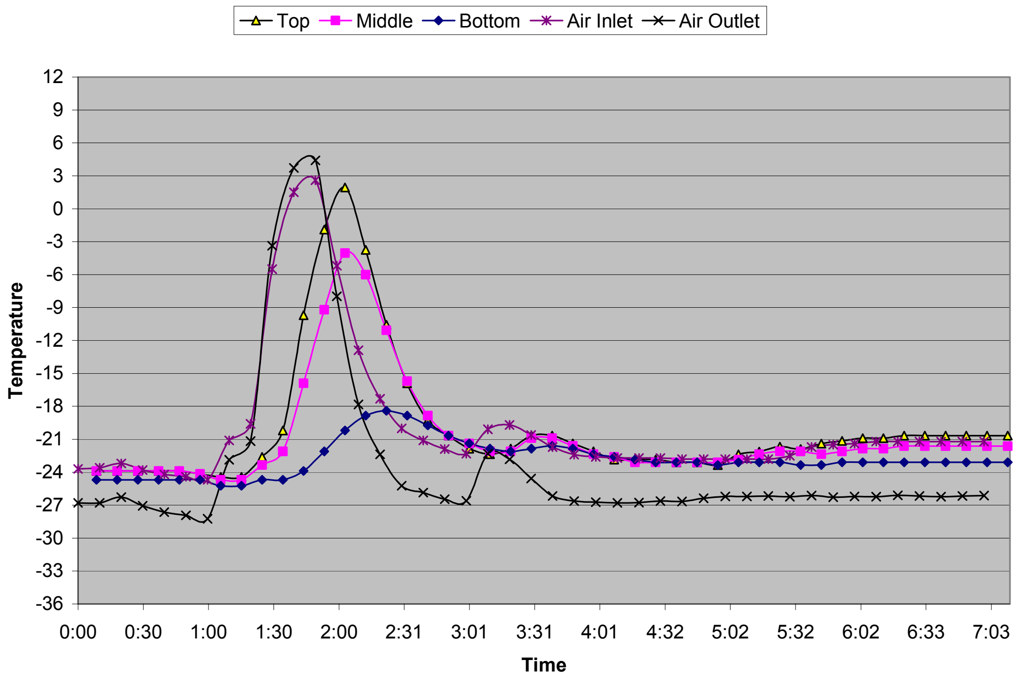


Figure 16b. SQD Defrosting Temperature Profile

Figure 16b shows that there is a small temperature lift in SQD defrost cycle after the main temperature changing peak, and it is caused by the nearby cabinet group's defrost termination. Because the three cabinet groups defrost in a sequence of C, A, B, the test group A did not get sufficient cooling supply, when there was a peak demand after the group B defrost termination point.

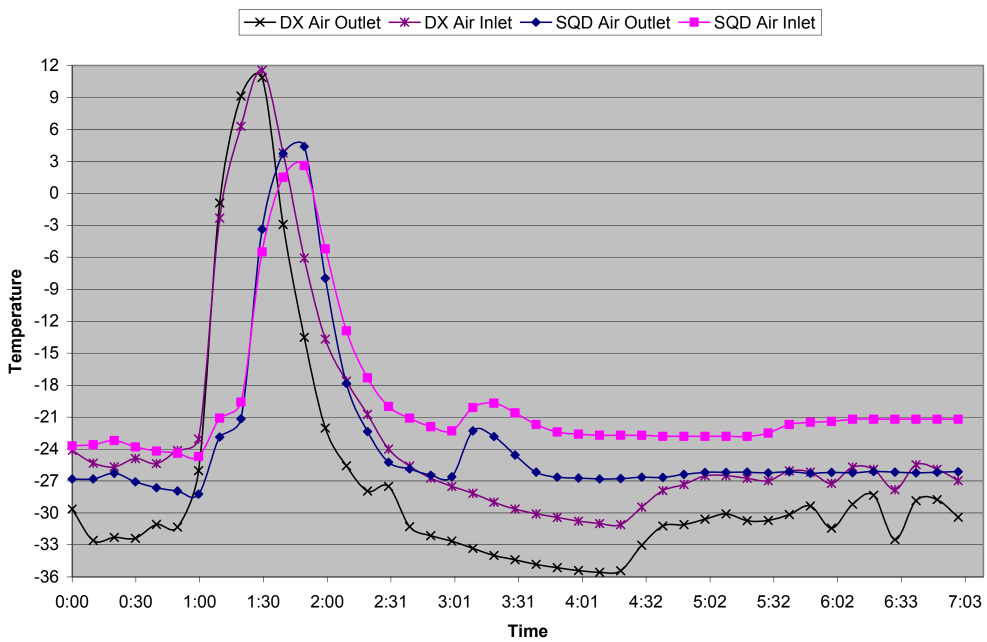


Figure 17a. Air Temperature Comparison during Defrost Time

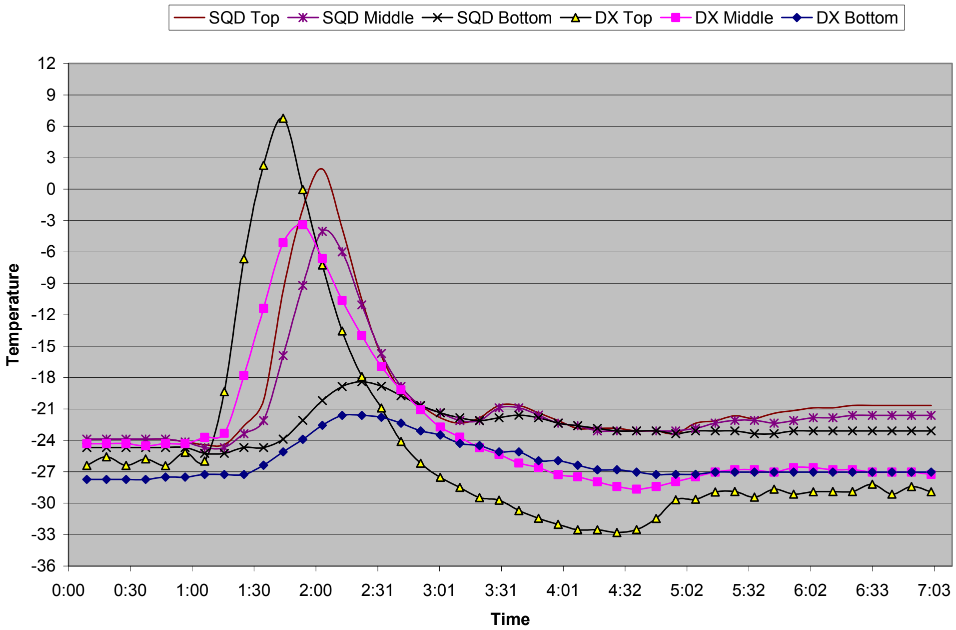


Figure 17b. Dummy Food Temperature Comparison during Defrost Time

The DX evening defrost is set to initiate at 7:00 PM and the SQD group A defrost is set to start at 4:40 AM. In order to make a clear comparison, the analyzed data were taken from 1 hour before the defrost initiation for both two cases, and the curves were drawn on the same time axis starting from zero. The comparison was based on the average data within the test period.

The air comparison indicates there exists a time delay for SQD air temperature raising, compared with DX air temperature curves'. The reason is that electric heater temperature is higher than SQD heat exchanger surface temperature within the defrost period, and the cold air in DX cabinets get a quick raising just after the defrost initiation.

One special point is that SQD defrost lead to a smaller temperature changing (for both food and air temperatures) during the whole process. The highest air temperature was only +5 °C while the correspondent temperature went to +12 °C in DX cabinets. This is caused by a better location of heat source in SQD cabinets. In warm liquid defrost, the heat directly reach the heat exchanger which has heavy ice formation. While in electric heating defrost, the air is heated first, and it acts as the heat source to heat the ice secondly. The same smaller temperature changing can also be found from the dummy food curves. Tinytalk test results indicate that the highest food temperature is around +7 °C in DX cabinet, and the food in SQD cabinet only has highest temperature as +2 °C.

One special point is that plastic chicken test results should also be analyzed because the plastic chicken recorded top position data could better reflect the real food temperature than tinytalk's. A temperature analysis is shown in table 12a and 12b.

Table 12a. Temperature Distribution from Tinytalk Test Results

Tinytalk			
DX	Bottom	Middle	Top
	Temperature (°C)		
Lowest	-29.1	-29.1	-34.6
Highest	-20.2	-0.6	7.9
SQD	Bottom	Middle	Top
	Temperature (°C)		
Lowest	-25.5	-24.7	-25.5
Highest	-14.0	-3.2	3.0

Table 12b. Temperature Distribution from Plastic Chicken Test Results

Plastic Chicken			
DX	Bottom	Middle	Top
	Temperature (°C)		
Lowest	-23.8	-22.5	-26.2
Highest	-19.7	-18.5	-10.2
SQD	Bottom	Middle	Top
	Temperature (°C)		
Lowest	-23.9	-23.3	-22.1
Highest	-18.8	-17.7	-8.4

The minimum and maximum temperatures were chosen from all the logged data (including not using night cover period) in both tinytalk and plastic chicken cases. It can be seen that the food temperature in SQD system has a higher highest temperature than DX's in plastic chicken tests. While the previous tinytalk test result analysis shows that DX cabinets have higher highest food temperature. Two reasons may cause these opposite results. First, tinytalk top temperatures reflect air temperature more than food temperature. DX air temperatures are do higher than SQD's in defrost cycle, and this can be proved from woodly test results. Second, the plastic chicken data could not be used to fully reflect the real circumstance because of the too short test period.

From above analysis, it can be concluded that DX defrost air temperature is higher than SQD defrost air temperature. However, because lower DX store-food temperature is maintained in the normal operation cycle, DX and SQD have similar highest food temperatures during defrost cycles. Obviously, smaller temperature lift will lead to better food quality and less compressor load raised by the defrost.

Although the temperature curves showed the SQD defrost a relatively longer time than DX's, the SQD cabinet need a shorter time to restore the normal condition (excluding the small temperature changing caused by the nearby cabinet group's defrosting). It can be seen from the air temperature curves that the DX cabinets need almost two more hours to restore the normal temperature condition after the temperature changing peak during the defrost period. Cooling was excessively supplied within the two hours, which leaded to extra compressor load. This is again, caused by the defeat of DX compressor control mechanism.

Laboratory-based tests show that warm liquid defrost need shorter time (10 to 15 min) than traditional electric heater defrost (12 to 21 min). [4] The relatively longer SQD defrost period may be caused by the limitation of the waste heat temperature, which is the secondary condensing refrigerant temperature around +31 °C. The warm liquid defrost time can be decreased by increasing the fluid velocity, because the increase of fluid velocity in the heat exchanger improves the heat transfer and consequently the rate of defrost water formation. However, too short defrost time will lead to a insufficient melting on the drain pan, and blocked drain will further cause water overflow and spill out of the case. Therefore, the related parameters should be carefully adjusted to reach an optimum defrost period.

Another defrost improving method is to use demand control, instead of traditional preset time control. Defrost frequency can be reduced by the optimum defrost initiation, and the further benefits will be better temperature control, increased product quality or life and energy savings. However, the initiation signal is very hard to be chosen. The methods such as measuring the ice thickness or the thermal conductivity of ice are complex and unreliable. One recent research indicates that a suitable parameter that could provide an indication for optimum defrost initiation is the divergence that occur between the temperature of the air just after the cabinet coil and at some point in the back flow tunnel after a period of operation between defrosts. This divergence is mainly caused by the reduction in the airflow through the coil arising from excessive frost built-up. Another parameter that could be used for optimum defrost initiation is the duration of the previous defrost cycle. [10] Only laboratory tests has been made for demand defrost control. The final commercial demand defrost control system should be easily operated, self-adaptive and non-cabinet and refrigeration system specific.

One important point is that the indoor temperature and humidity have strong influence on the rate of ice formation. Therefore, a tight indoor humidity as well as temperature control is recommended at least for cold storage area within supermarkets.

8.3 INFILTRATION AND NIGHT COVER

Infiltration is the biggest energy loss part among all the items, the easiest and most efficient method to decrease the loss is installing insulated glass on the open area of the cabinet. However, it will make purchase inconvenient, glass windows are therefore not popular for frozen cabinets, only few types of horizontal cabinets can be found in supermarkets equipped with glass windows.

Instead of insulated glass, night covers are widely used in current supermarket display cabinets to minimize the infiltration as well as radiation losses. A layer of thin and reflecting insulation material is simply used to cover the cabinet open area in the evening. The test results indicate significant energy savings and tight temperature controls can be achieved by using night covers.

8.3.1 ENERGY CONSUMPTION ANALYSIS

The compressor energy consumption had been measured both while using and not using night covers. The calculation was based on the average of ETM logged compressor power input data during the night cover on period, and one tested cover off period. Such night cover off test was arranged in Saturday evening (03.03 ~ 03.04, 2001) in DX system and on Monday evening in SQD system (02.12 ~ 02.13, 2001).

Table 13. Compressor Energy Consumption Comparison on the Effect of Using Night Cover

DX	With NC.	No NC.
Energy Consu. (kWh)	42.749	48.941
Net Saving	6.191	
Saving Poten.	12.7%	
SQD	With NC.	No NC.
Energy Consu. (kWh)	52.415	65.849
Net Saving	13.434	
Saving Potent.	20.4%	

A higher saving potential can be achieved by using night cover in SQD cabinets. The possible reason is that the supermarket at Mörby Centrum (SQD) has shorter opening time (2hours less per day) than that of Sundbyberg Centrum supermarket, and the longer period of using night cover will obviously lead to a larger energy saving. The field inquires showed that normally night covers are put on 10 ~ 15 min after the close time and put off one hour before the opening time in both places.

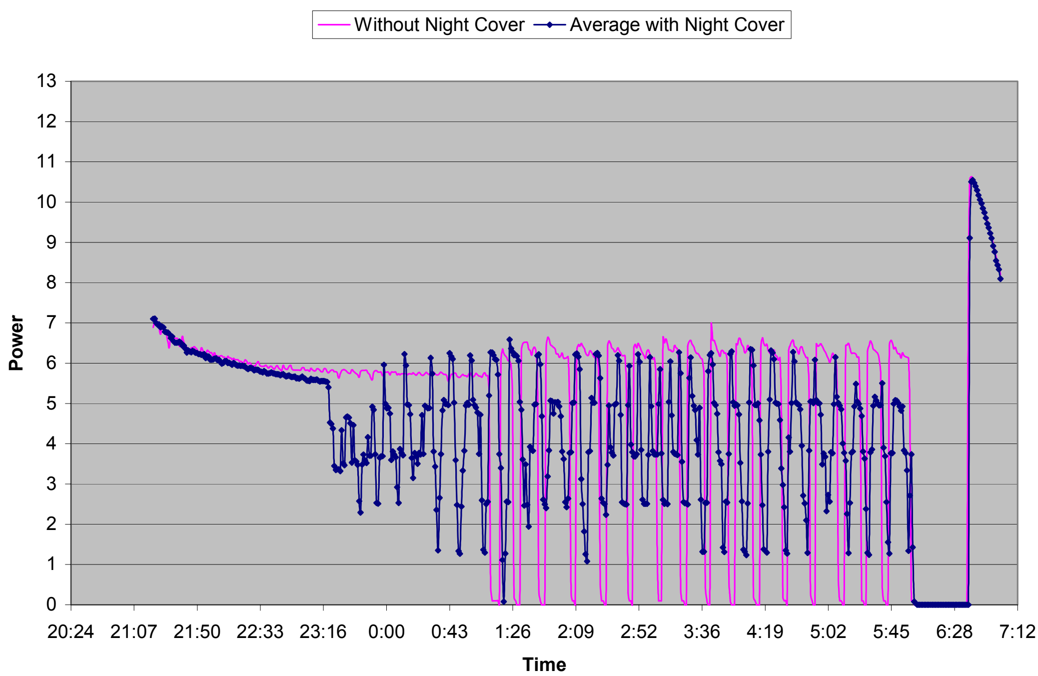


Figure 18a. DX Compressor Power Input Comparison on the Effect of Using Night Cover

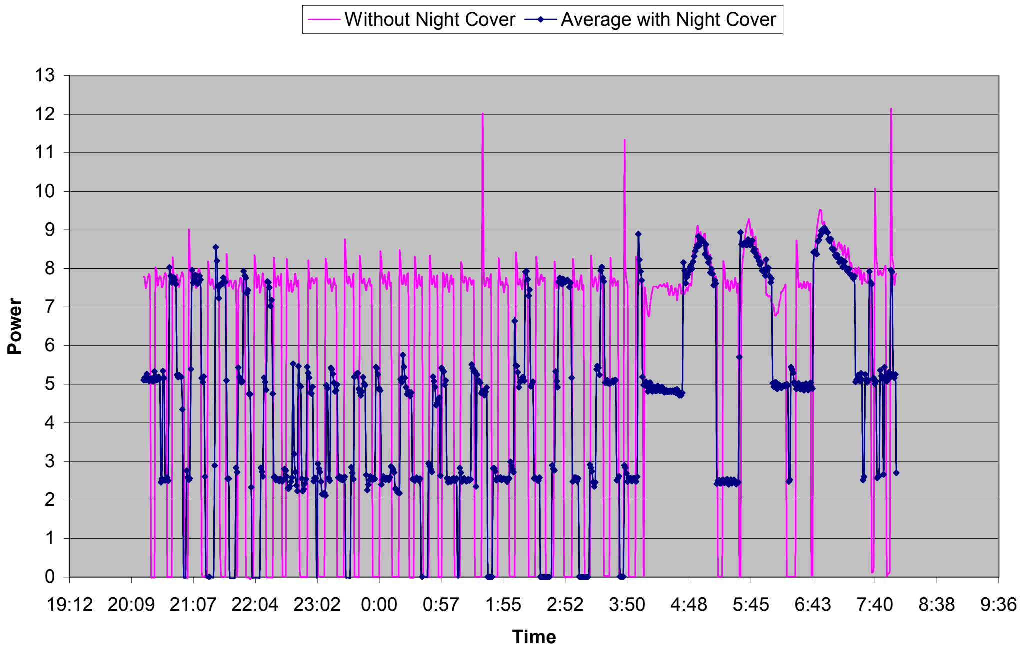
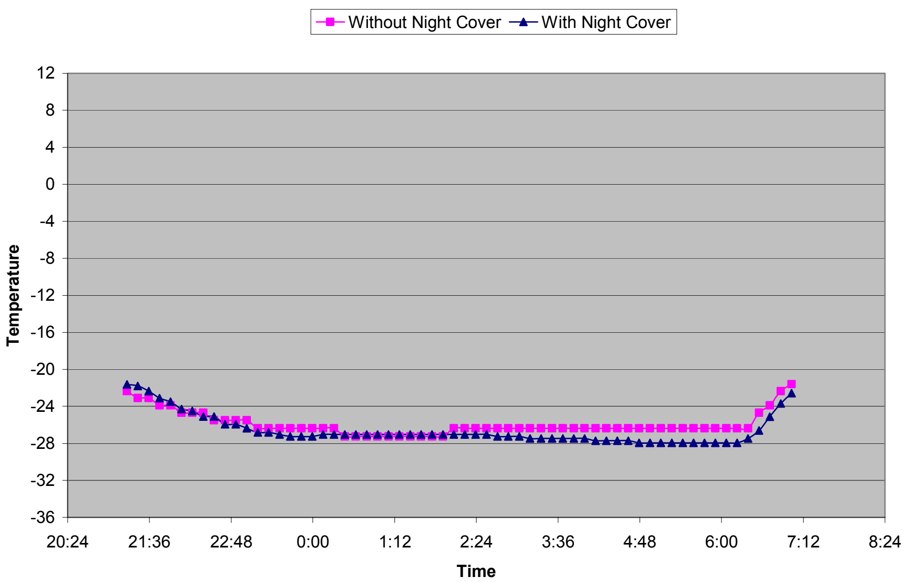


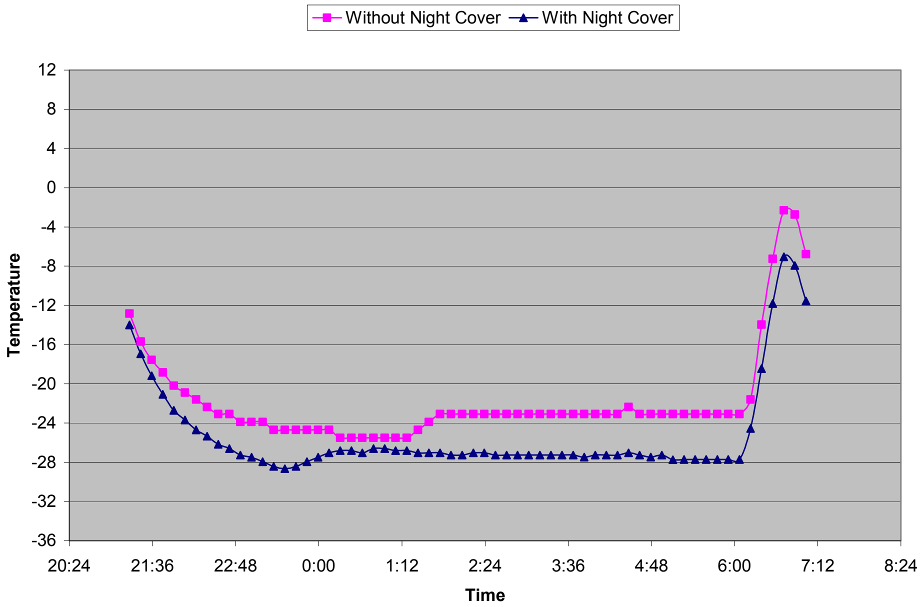
Figure 18b. SQD Compressor Power Input Comparison on the Effect of Using Night Cover

It can be seen from the both figure 18a and 18b that the compressor power input was much lower when night covers were put on in the evening. Longer compressor off intervals were also achieved, and that is why many lower average power inputs (around 2 ~3 kW) appeared on the average curves.

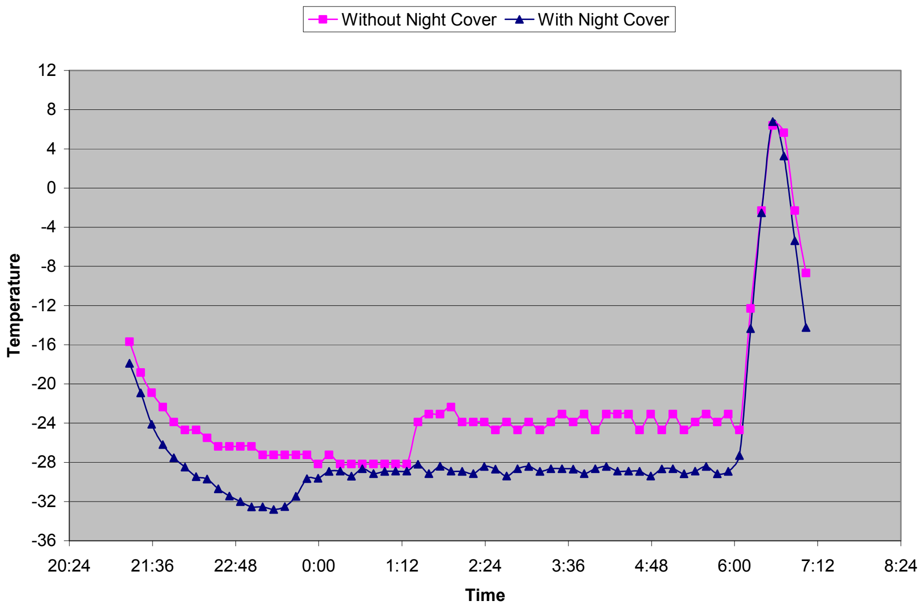
8.3.2 TEMPERATURE ANALYSIS



Figures 19a. DX Bottom Temperature Comparison on the Effect of using Night Cover

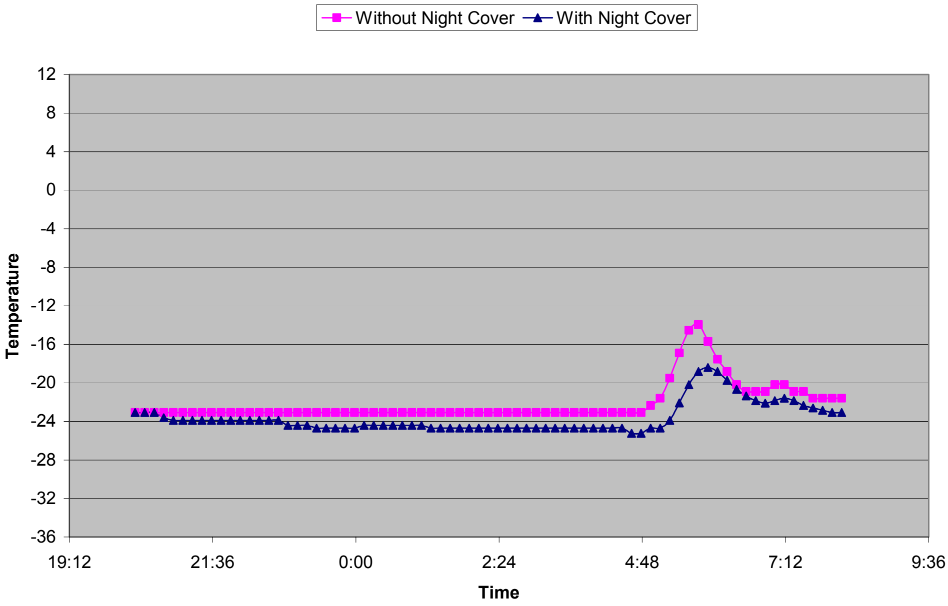


Figures 19b. DX Middle Temperature Comparison on the Effect of using Night Cover

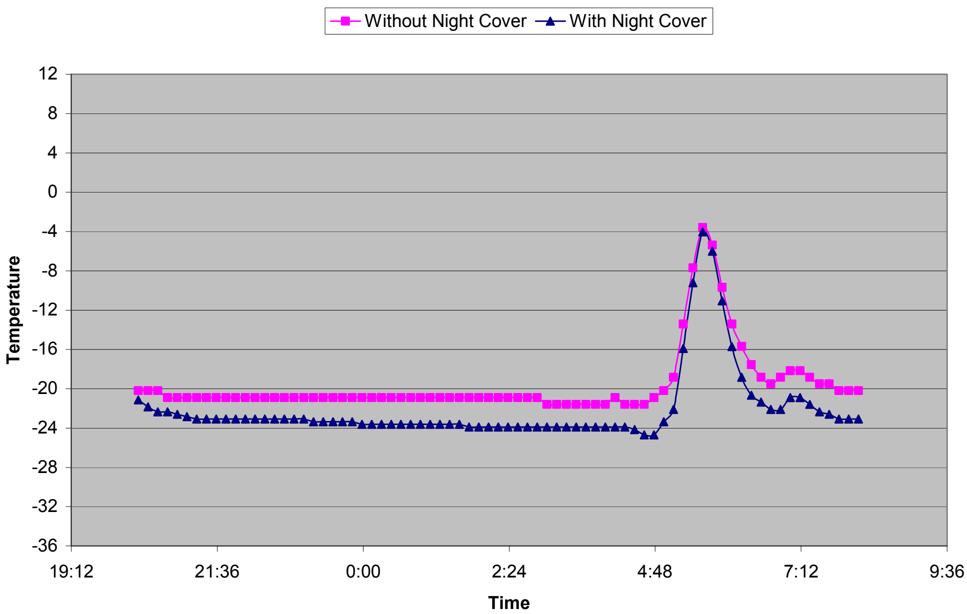


Figures 19c. DX Top Temperature Comparison on the Effect of using Night Cover

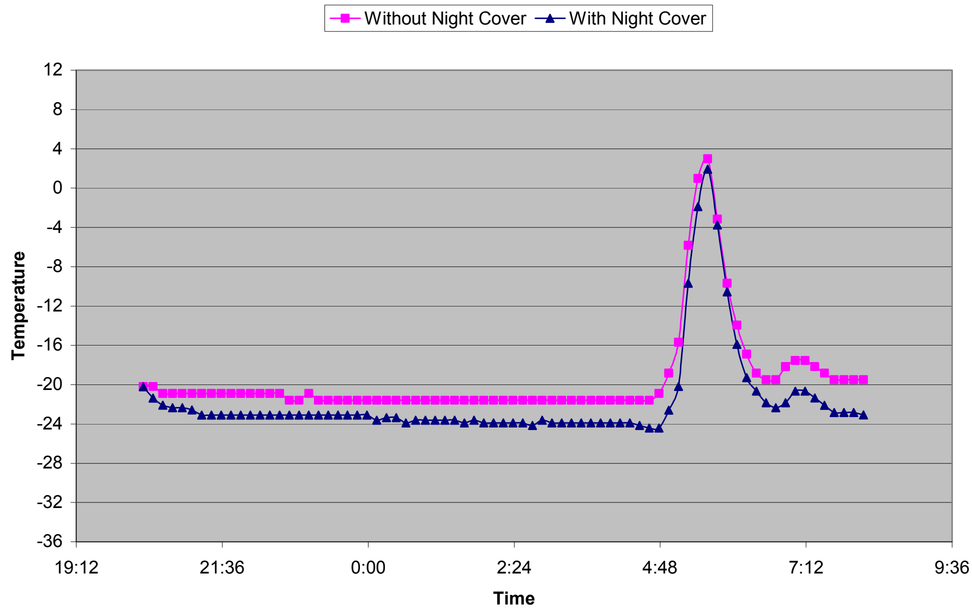
The effect of using night cover can be easily observed from the temperature decreases in all of the three cases. Better results were accomplished on the top part, because most of infiltration and radiation losses that happen around the top open area are avoided by the using of night cover.



Figures 20a. SQD Bottom Temperature Comparison on
the Effect of using Night Cover



Figures 20b. SQD Middle Temperature Comparison on
the Effect of using Night Cover



Figures 20c. SQD Top Temperature Comparison on the
Effect of using Night Cover

Figures 20a, 20b and 20c indicate that the similar results were accomplished once more from SQD cabinet test. More stable food temperatures were also achieved at all the test positions in SQD cabinet. Such results again proof the advantage of indirect systems.

Table 14. Dummy Food Temperature Comparison on the
Effect of Using Night Cover

DX	With NC.	NO NC.	Temp. Diff.
Temp.	°C		
Top	-26.5	-22.5	3.9
Middle	-25.2	-21.7	3.5
Bottom	-26.6	-25.9	0.6
SQD	With NC.	NO NC.	Temp. Diff.
Temp.	°C		
Top	-21.7	-19.5	2.2
Middle	-22.1	-19.6	2.5
Bottom	-23.6	-22.1	1.5

The temperature differences are clearly shown in table 14. A general temperature decrease happened in SQD cabinet when night covers were put on, while such difference was very weak in the bottom of DX cabinet.

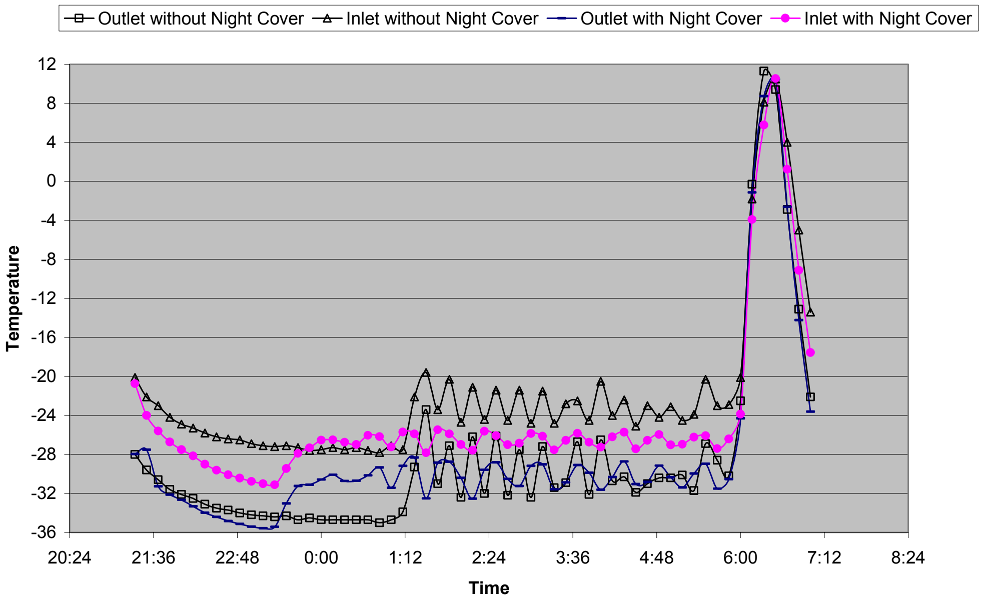


Figure 21a. DX Air Temperature Comparison on the
Effect of Using Night Cover

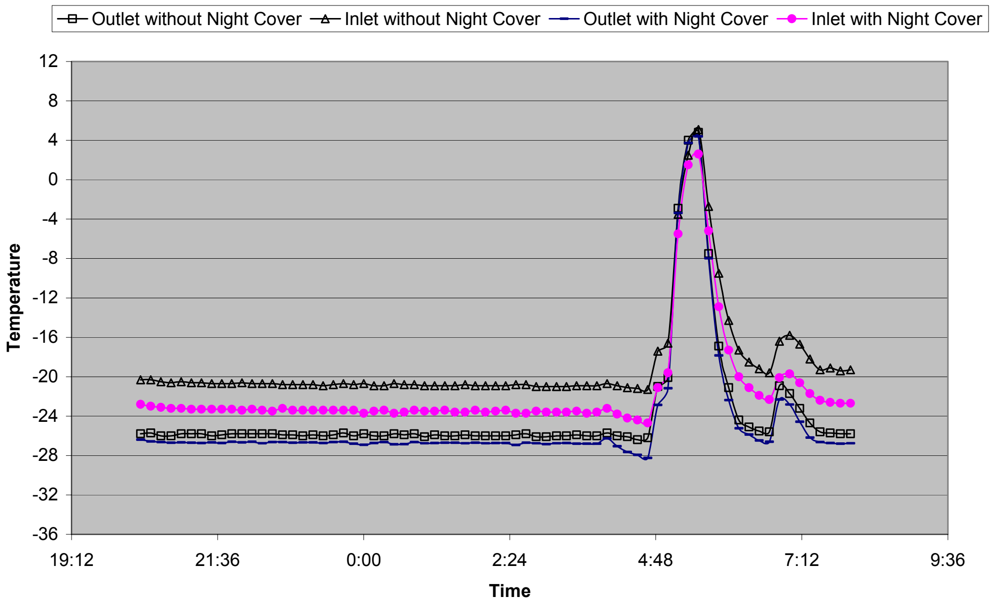


Figure 21b. SQD Air Temperature Comparison on the
Effect of Using Night Cover

First, the air inlet and outlet temperature differences were smaller in both cabinets when night covers were used, which means cooling loads were decreased by minimized infiltration loss. Second, SQD cabinet outlet air was colder in cover used condition, while the air outlet temperature remain the same in DX cabinet, and this is caused by the usage of single phase secondary refrigerant. Third, the character of uniform temperature distribution in SQD system was distinctly shown in the comparison of the two figures. The related temperature data were further calculated in table 15.

Table 15. Air Temperature Comparison Data on the Effect of Using Night Cover

DX	With NC.	NO NC.	Temp. Diff.
Temp.	°C		
Air Inlet	-24.4	-21.8	2.6
Air Outlet	-28.1	-28.3	-0.2
Temp. Diff	3.7	6.5	--
SQD	With NC.	NO NC.	Temp. Diff.
Temp.	°C		
Air Inlet	-21.6	-18.8	2.8
Air Outlet	-24.8	-23.9	0.9
Temp. Diff	3.2	5.1	--

9 COST AND EFFECTIVENESS ANALYSIS

9.1 LIFE CYCLE COST ANALYSIS (LCCA)

The same like technical evaluation, economic analysis is very important and necessary for the thorough understanding about refrigeration systems. Therefore, Life Cycle Cost Analysis (LCCA) is introduced to accomplish the task. Life cycle cost analysis is an economic method of project evaluation in which all costs arising from owning, operating, maintaining, and ultimately disposing of a project over a given study period (usually related to the life of the project) are considered to be potentially important to that decision. [11] The life cycle cost formula for refrigeration systems can be expressed as follows.

$$LCC = I + E + Med + OM\ \&\ R + \mathrm{Re}\,pl - \mathrm{Re}\,s + Ins + Env$$

Where

LCC:	Total LCC in present value kronor of a given alternative
I:	Present value investment costs
E:	Present value energy costs
Med:	Present value capital costs for the media in use
OM&R:	Present value non-fuel operating, maintenance, and repair costs
Repl:	Present value capital replacement costs
Res:	Present value residual value
Ins:	Present value insurance costs
Env:	Present value capital environmental costs (due to emissions)

The insurance costs vary a lot among specifics, and currently there is no refrigeration systems related emission costs. The last two factors were consequently not counted within this discussion. Both test DX and SQD system's life cycle costs were calculated depending on the example from Department of Energy's Federal Energy Management Program (FEMP), United States. [11] All the calculations were based on logged data and information provided by FRIGOTECH AB on March 2001.

<u>Life Cycle Cost Calculation</u>

Discount Rate: Current FEMP discount rate: 3% real for constant kronor analysis
Energy Prices: Fuel Type: Electricity at 0.26 SEK/kWh for both DX and SQD systems.
Rate Type: Commercial
Discount Factor: FEMP Uniform Present Value (UPV) factor based on a 3% (real) discount rate
Useful lives of systems: 15 years
Study period: 15 years
Base Date: March, 2001

The two systems' economic data and the calculating process are presented in table 16a and 16b.

Table 16a. DX LCC Calculation Sheet

Cost Items (1)	kWh	Base Date Cost (2)	Year of Occurrence (3)	Discount Factor	(4) Number	Present Value (5)=(2)*(4)
Initial Investment Cost		450 000	Base Date			450 000
El. Compressor	140 360	36 494	Annual	FEMP UPV*15	12.02	438 653
El. Dry Cooler	25 000	6 500	Annual	FEMP UPV*15	12.02	78 130
El. Defro.	22 842	5 939	Annual	FEMP UPV*15	12.02	71 386
OM&R		20 000	Annual	UPV15	11.94	238 800
Repl.		200 000	10	SPV10	0.744	148 800
Res		-30 000	15	SPV15	0.642	-19 260
Total						1 406 509

Table 16b. SQD LCC Calculation Sheet

Cost Items (1)	kWh	Base Date Cost (2)	Year of Occurrence (3)	Discount Factor	(4) Number	Present Value (5)=(2)*(4)
Initial Investment Cost		600 000	Base Date			600 000
El. Compressor	89 236	23 201	Annual	FEMP UPV*15	12.02	278 880
El. Dry Cooler	25 000	6 500	Annual	FEMP UPV*15	12.02	78 130
Subcoolers	36 201	9 412	Annual	FEMP UPV*15	12.02	113 135
Pumps	6 620	1 721	Annual	FEMP UPV*15	12.02	20 689
OM&R		10 000	Annual	UPV15	11.94	119 400
Repl.		0				0
Res		-100 000	15	SPV15	0.642	-64 200
Total						1 146 035

FEMP UPV*	FEMP Uniform Present Value discount factors adjusted for fuel price escalation, by end use sector and fuel type
UPV	Uniform Present Value discount factors (non-fuel)
SPV	Single Present Value discount factors

(The used factors can be found from factor tables in Appendix C.)

- Energy Calculation

DX

Compressor
W_w = 2 691.837 kWh/week
$W_y = W_w \times 365/7$= 140 360 kWh

Defrost
$W_{de.\ d}$ = 20.86 kWh/day
$W_{de.\ y}$ = 20.86×365×3=22 842 kWh

SQD

Compressor
W_w = 1 711.368 kWh/week
$W_y = W_w \times 365/7$= 89 236 kWh

Subcooler
Esc = 5.510 kW
Wsc. y = Esc×load factor×24×365
= 5.510×0.75×24×365
=36 201 kWh

Pumps
$W_{de.\ d}$ = 0.608 kWh/day
$W_{de.\ y}$ = 0.608×365 = 222 kWh
$W_{cir.\ d}$=0.7303×24=17.527 kWh/day
$W_{cir.\ y}$ = 17.527×365 = 6 398 kWh
$W_{ptotal.\ y} = W_{de.\ y} + W_{cir.\ y}$
=222 + 6 398 = 6 620 kWh

- Explanation

✓ The refrigerant costs are already included in total initial investment cost.
✓ Component replacement happens at the end of year 10 in DX system, while there is no replacement in SQD system due to the better design and installation.
✓ Th higher OM&R in DX are mainly caused by electric heater defrosting related problems.
✓ The pumps energy consumption in SQD system includes the energy used by warm liquid defrost pumps and the circulating pump on secondary evaporating cycle.
✓ LCC was calculated on the basis of same system cooling capacity that is 19.77 kW.

SQD system is obviously the first choice from economic point of view because its LCC of 1 146 035 SEK is lower than the LCC of 1 406 509 SEK of DX system. The Net Saving can also be easily calculated out.

NS = LCC DX – LCC SQD = 1 406 509 – 1 146 035 = 260 474 SEK

This means the advanced indirect refrigeration system saves 260 474 SEK in present value kronor over 15 year study period, over and above the 3 percent minimum acceptable real rate of return already taken into account through the discount rate.

9.2 SYSTEM EFFECTIVENESS EVALUATION (COSP)

COP (Coefficient of Performance) is traditionally used to give refrigeration engineers an idea about the system efficiency and the compressor power input, but such parameter can not exactly reflect the system power input. COSP (Coefficient of System Performance) is consequently introduced. It is also a ratio of the heat extracted by the evaporator divided by the work input to the system, and the different point is the definition of the work input to the system. Instead of compressor power input in COP, the power input in COSP includes the needs for driving compressor, fans as well as pumps and any other heat transfer assists, such as subcooler. [12] The COSP is therefore a ratio of the useful work to the total energy that has to be paid for. It can provide a more useful index of performance. The DX and SQD COSP were calculated as follows.

DX System

$COSP_{DX} = \dfrac{Q_2}{E_{comp} + E_{cond}}$, Where $Q_2 = m_{DX} \times (h_{2kr} - h_{1sr})$ and $E_{cond} = E_{fan} + E_{pump}$.

The data as well as parameters were listed in table 17a.

Table 17a. DX COSP Data

DX				
Cab. Eva. Outlet	kJ/kg	h_{2kr}	206.9	Measured or Calculated Data
Before Exp. Valve	kJ/kg	h_{1sr}	95.27	
Comp. Power	kW	E_{comp}	6.252	
Ref. Mass Flow	kg/s	m_{DX}	0.0686	Manu. Data
Dry Cooler Fan Power	kW	E_{Fan}	0.53	
Pump Power	kW	E_{Pump}	0.18	

$$COSP_{DX} = \frac{0.0686 \times (206.9 - 95.27)}{6.252 + 0.53 + 0.18} = 1.1$$

SQD System

$$COSP_{SQD} = \frac{Q_2}{E_{comp} + E_{cond} + E_{cir.pump} + E_{sc}}$$

Where $Q_2 = c_{40} \times m_{40} \times \Delta T_{40}$ $\qquad E_{cond} = E_{fan} + E_{pump}$

$E_{sc} = \dfrac{Q_{2sc}}{COP_2}$ $\qquad Q_{2sc} = c_{20} \times m_{20} \times \Delta T_{20}$

The data as well as parameters were listed in table 17b.

Table 17b. SQD COSP Data

SQD			
Subcooler Temper 20	kJ/kg C	c_{20}	3.263
	kg/s	m_{20}	0.344
	°C	ΔT_{20}	6.136
		Nor. COP	**2.5**
Cooling Cap. Temper 40	kJ/kg C	C_{40}	2.875
	kg/s	m_{40}	3.43
	°C	ΔT_{40}	2.154
	KW	$E_{cir\ pump}$	0.7303
Compressor	KW	E_{comp}	15.665
Dry Cooler	KW	E_{fan}	1.59
	KW	E_{pump}	0.54

Subcooler

$$Q_{2sc} = 3.263 \times 0.344 \times 6.136 = 6.887 \text{ kW}$$

$$Esc = \frac{6.887}{2.5} \times 2 = 2.7548 \times 2 = 5.510 \text{ kW}$$

$$Q_2 = 2.875 \times 3.43 \times 2.154 = 21.246 \text{ kW}$$

$$COSP_{SQD} = \frac{21.246}{15.665 + 5.510 + 0.7303 + 2.13} = 0.884$$

It can be seen from the calculation that, if take into account all the compression system power inputs, SQD has lower efficiency, and that is mainly caused by the extra subcoolers' energy consumption.

9.3 ECONOMIC EVALUATION

Systems with low total cost (including initial cost and operating cost) are pursed by proprietor owners. Traditionally, all indirect systems are treated as more expensive options compared with direct expansion systems. The main reason for higher total cost is coming from the extra pumps, heat exchangers and the related temperature lift.

The previous Life Cycle Cost Analysis (LCCA) shows that, though a higher initial investment is needed in indirect SQD system, the total SQD system cost is smaller than that of DX system over the studied service life. A higher SQD COSP can only be used to reflect the compression cycle efficiency. If take into account two systems' defrost energy consumption, SQD system will have a lower total system energy (electricity) input, compared with DX system's. This result proves the statement that the total system energy consumption of a well-built indirect system can be lower than that of a direct expansion system, or a partially indirect (secondary condensing cycle) system within the test case. Therefore, indirect SQD system is superior to partially indirect DX system from both economic and energy saving points of view.

10 ENVIRONMENTAL IMPACT EVALUATION (TEWI)

Environmental impacts, such as global warming and ozone depletion are more and more of concern since the 1980's. The Total CO_2 Equivalent Warming Impact (TEWI) is a good criterion when judging and comparing the influence of the refrigeration systems on environment. Two factors will influence a refrigeration system's global warming contribution. The first one is the direct CO_2 emission caused by refrigerant leakage, while the second one is the indirect emission of CO_2 from the primary energy (electricity) producing process. A simple program was used to calculated the total emission of CO_2 for the test direct (DX) and indirect (SQD) system. The calculation results are present in figure 22and the program is shown in table 18. [13]

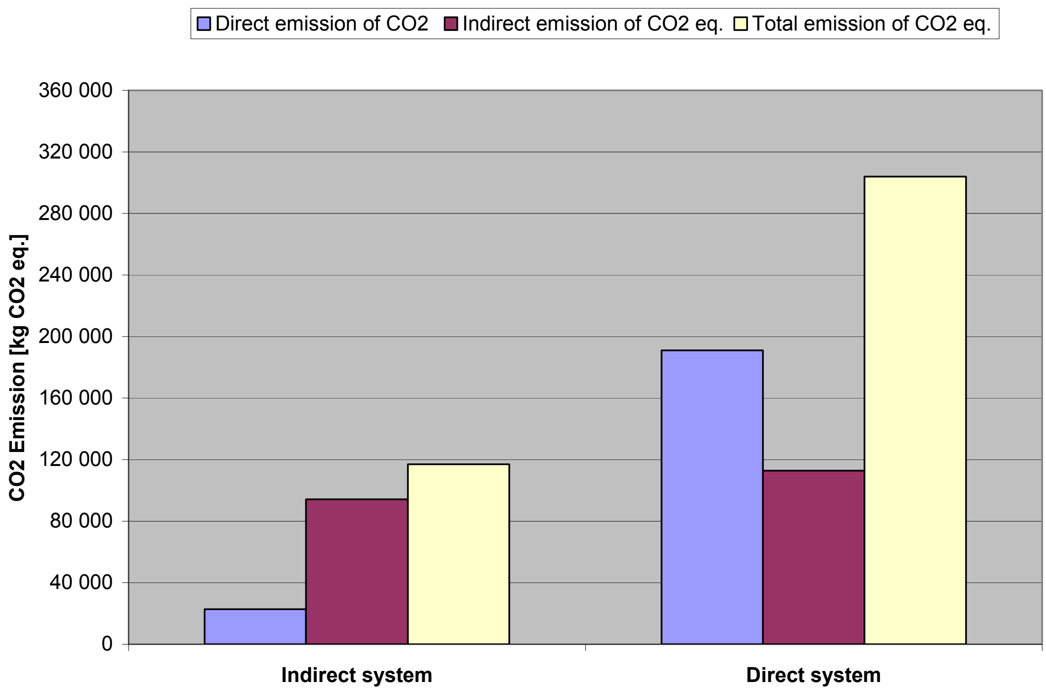


Figure 22. TEWI Comparison (Indirect System VS. Direct System)

Not only the compressor performance life, but other electricity consumption (defrost, dry cooler and pump energy consumption in DX system, defrost, dry cooler, pumps and subcooler energy consumption in SQD system) as well, were considered in the calculating process to reflect the real condition. All the data were taken from the life cycle cost calculation. The similar indirect emissions are achieved from two systems, as shown in figure 25, and the reason is that the energy saved by warm liquid defrost in SQD system is equivalently consumed by the extra subcoolers' need. Due to the less amount of primary refrigerant charging, indirect SQD system is still more environmentally friendly, compared with direct DX system.

Table 18. TEWI for DX and SQD System

TOTAL ENVIRONMENTAL WARMING IMPACT - TEWI				
	Indirect system		Direct system	
Chiller life	15	[Year]	15	[Year]
Refrigerant charge	7	[kg]	58.68	[kg]
Refrigerant	R404A	[ASHRAE N°]	R404A	[ASHRAE N°]
G W P	4 340.0	[CO2 = 1]	4 340.0	[CO2 = 1]
Loss rate	5	[%]	5	[%]
Direct emission of CO2	22785	[kg CO2]	191003.4	[kg CO2]
Nominal cooling capacity	19.8	[kW]	19.8	[kW]
Real COP	1.460	[kW/kW]	1.444	[kW/kW]
Annual Chiller Performance (life)	89 236	[kWh]	140 360	[kWh]
Total Chiller Performance (life)	1 338 540	[kWh]	2 105 400	[kWh]
Other Annual Energy Consu. (El.)	67 821	[kWh]	47 842	[kWh]
Other Total Energy Consu. (El.)	1 017 315	[kWh]	717 630	[kWh]
Total Energy Consu. (El.)	2 355 855	[kWh]	2 823 030	[kWh]
Country	Sweden		Sweden	
Regional conversion factor	0.04000	[kg CO2/kWh]	0.04000	[kg CO2/kWh]
Indirect emission of CO2 eq.	94 234	[kg CO2 eq.]	112 921	[kg CO2 eq.]
Direct emission of CO2	22 785	[kg CO2]	191 003	[kg CO2]
Total emission of CO2 eq.	117 019	[kg CO2 eq.]	303 925	[kg CO2 eq.]
Direct/Total	19.47	[%]	62.85	[%]

11 CONCLUSION

An intensive comparison had been made between a typical direct system with a secondary condensing cycle (DX), and a completely indirect system (SQD) in the field of supermarket deep freeze refrigeration systems. In general, indirect SQD system is much better than partially indirect DX system from following aspects:

- ✓ More uniform product temperature is maintained in SQD cabinet with less compressor energy consumption.
- ✓ Less primary refrigerant charging amount is needed by indirect SQD system, which lead to less environmental impact.
- ✓ Less compressor energy consumption is accomplished by a better compressor control method and the utilization of high performance secondary refrigerant.
- ✓ The total SQD system energy consumption is less than that of DX system, if the same system capacity is considered.
- ✓ Instead of consuming huge amount of energy, waste energy from secondary condensing side is used by the patented warm liquid defrost technology (SQD). A better result from food product quality point of view is also achieved by using such technology.
- ✓ The energy saving potential caused by using of night cover is higher in SQD system.
- ✓ Though DX system's compression cycle efficiency (COSP) is higher than SQD system's, it can not be used to prove that DX refrigeration system is superior to SQD system. On the contrary, SQD refrigeration system is much more attractive from an economic point of view because of the less Life Cycle Cost (LCC) over the total system service life.

Although these conclusions were drawn from comparisons of the specific test systems, they could be used to all the comparisons as well as evaluations between direct and indirect systems that are widely used in today's supermarkets, because of the similar system configurations and design methods. And it indicates that, in supermarket application, Indirect Refrigeration Systems generally perform better than Direct Refrigeration Systems.

The research on direct and indirect systems can only be treated as a part of the thorough investigations on today's supermarket used (deep-freeze) refrigeration systems. Therefore, the second part of the thesis concentrates on study of other types refrigeration systems as well as all the possible system optimization methods, and they were discussed mainly from theoretical point of view.

ALTERNATIVES AND SYSTEM OPTIMIZATION

1 PURPOSE

Looking at current supermarket refrigeration systems from other points of view than the distinction of direct and indirect, a lot of different system configurations could be found. Moreover, If thinking more about refrigeration systems, a different approach can also be made other than cabinets and machine rooms. Alternative refrigerants, varied circulation system configurations, and high tech applications are the components for the “SYSTEM”.

In the following part some alternatives, recent optimization methods for supermarket refrigeration systems were thoroughly studied. Some of them are still tested in prototype, while others can be found in practical installations. The integrated literature surveys can lead to further understandings for different principles. All in all, most of these technologies were analyzed from energy saving, efficiency improving, as well as environmental protection points of view.

2 REFRIGERANTS AND ALTERNATIVE SYSTEMS

2.1 REFRIGERANT EVLUATION

Due to the phasing out of CFCs and HCFCs, the topic of using sustainable, environmental friendly refrigerants as well as achieving excellent performance comes to a highlight. It is not so hard to estimate that HFCs, such as R404A, which are widely used in current supermarket refrigeration systems, may also be phased out in the near future. Within the field of food storage refrigeration, some long used natural refrigerants, such as ammonia (NH_3) and carbon dioxide (CO_2) are reused in many new installations. On the other hand, the research on utilizing zero-pollution refrigerants, such as air and water (H_2O), also achieves rapid development. Some interesting and promising refrigerants are reviewed in the following part for both primary and secondary circuits.

2.1.1 PRIMARY REFRIGERANTS

1. AMMONIA (R717)

Ammonia is a well-known refrigerant with more than 100 years of practical experience. It has many advantages. First, ammonia is a natural refrigerant with no Ozone Depleting Potential (ODP) or Greenhouse Warming Potential (GWP). Secondary, it has high heat of vaporization, excellent heat transfer and thermophysical properties. Ammonia is cheap and can be easily used in systems without suffering the oil separation problems, because it has lighter density than immiscible oils'. But ammonia is toxic and can not be used directly due to the security reasons. Furthermore, it is not compatible with the most commonly used material in refrigeration systems, copper. [1]

Due to its toxic and high discharge temperature properties, ammonia is more suitable for indirect and two stage systems, which just matches both supermarket's chilling and freezing refrigeration load exactly. Because of all these reasons, ammonia is employed in many recent supermarket refrigeration installations.

The ammonia plant is an ideal candidate for system retrofit. A field experience [6] indicates that the existing (H)CFC plant can be replaced by a new indirect two stage ammonia plant without replacing neither any display cabinets nor evaporators in cold storage rooms. Better performance can even be accomplished from both energy saving and prescribed temperature maintain. Ammonia solution can also be used for HFC plant replacing, such as the measured DX system that uses R404A, the HFC refrigerant with high Global Warming Potential (GWP). Due to the complexity and the flexible expense, indirect system retrofitting seldom uses ammonia plan.

2. CARBON DIOXIDE (R744)

Carbon Dioxide is also a widely used refrigerant with many advantages. It has no ODP and no additional direct GWP. Thanks to its high-pressure operation property, CO_2 refrigeration systems require very small dimensions and greatly reduced compression ratio compared to conventional refrigerants. One outstanding point is that CO_2 has excellent thermodynamic properties. If CO_2 is employed as refrigerant, the heat transfer coefficient as well as refrigeration capacity will be considerably higher for the CO_2 evaporator compared with the identical evaporator with halocarbons as working fluid. [14] High performance leads to a reduction on the size of heat exchangers. After being used in display cases, a larger part of the cabinet volume may be used for food storage. In addition, the heat of vaporization for CO_2 is relatively higher than CFCs' and HCFCs', that means a higher refrigerating effect can be achieved with the same mass flow rate.

CO_2 is cheap and has complete compatibility to common machine construction materials. [15] The main problem for CO_2 being a refrigerant is the low critical temperature of +31.1 ˚C with the high critical pressure of 73.8 bar, which means CO_2 is more suitable for low temperature (frozen storage) systems. Safety valves, expansion containers, and auxiliary systems should be used individually or with combinations for security considerations. According to practical experience, CO_2 is mainly used for cascade systems or secondary systems to fulfill supermarket's low temperature cooling demand, due to its thermodynamic and thermophysical properties.

3. HYDROCARBONS (HCs)

Hydrocarbons (HCs), such as methane, ethane, propane and butane, are a group of refrigerants with bright future. They have non-ODP and very low GWP. However, they are all flammable and explosive, which limits the wide application. Hydrocarbons can be used as pure substances, as hydrocarbon blends and as blends with halocarbon refrigerants. They are all ideal candidates for CFCs, HCFCs, and even HFCs replacement. A recent research [16] indicates that all halocarbon refrigerants can be replaced by an equivalent hydrocarbon refrigerant or by an equivalent blend of hydrocarbon refrigerants, such as the group of CARE® fluids, a type of proprietary refrigerants produced by an UK. company. The system performance will maintain the same level or a better result can be achieved. Therefore, it can be easily pointed out that the both DX and SQD system can use HCs to replace their HFC refrigerant for a better system performance and a lower contribution to GWP.

Moreover, hydrocarbon refrigerants can also be used as components of non-flammable, zero-ODP replacement blends, such as R413A®. [16] The presence of the hydrocarbon improves efficiency by increasing the critical temperature of the blend and can also produce improved oil return properties.

4. AIR

The earliest real success in mechanical refrigeration was an air-cycle refrigerating machine built in 1844 and patented in 1851 by the physician John Gorrie in Florida, United States. In fact, air is the ultimate green refrigerant that has really zero environmental impact compared with any other types of refrigerants except water. However, some drawback eliminates its application mainly in aerospace air conditioning. First of all is the low critical point that leads to air remaining as a gas throughout the normal refrigeration process. Second is the rapid COP dropping due to the introduced irreversibilities of practical machinery, such as turbines both for expansion and compression process. [1]

The research on wider air refrigeration utilizing never stops. Some progress is already made on low temperature refrigeration field. One practical cycle is the Joule cycle or the "cold air cycle" that consists theoretically of four processes; two isentropic processes (compression and expansion) and two isobaric (cooling and heating). Such typical supermarket utilization based on the concept of open-air cycle has already been built up and tested in prototype.

2.1.2 SECONDARY REFRIGERANTS

1. CARBON DIOXIDE (R744)

Carbon Dioxide is also a good medium when used as phase changing secondary refrigerant. It has high volumetric heat capacity (e. g. 360 MJ/m^3 for CO_2 (liquid at –40 ˚C) against 3 MJ/m^3 for conventional secondary refrigerants), [15] thus the needed volume flow rate is greatly reduced for a given cooling load. No temperature gradient exist between the in and outlet of the heat exchangers because of its phase changing character. Low viscosity lead to a low pressure drop, moreover CO_2 also has very low temperature gradient due to the pressure drop (the reason is that it has very low viscosity), which is an important advantage for supermarket application, especially when long transport lines are unavoidable. However, as with direct systems, safety control methods must be considered during the design stage.

2. AQUEOUS SOLUTIONS

Aqueous solutions are a group of liquid secondary refrigerants which are widely used in today's low temperature refrigeration field. Generally, they have better performance and are much cheaper than non-aqueous liquids. Individual thermal properties as well as detail comparisons for most of the currently used aqueous solutions (including the following discussed refrigerants) can be found in reference [17].

✓ **POTASSIUM FORMATE**

Potassium Formate (K FO) is the most promising secondary refrigerant among all the water based solutions. Compared with the traditional aqueous solutions of alcohols and glycoles, it has many advantages. First of all, potassium formate has better thermal conductivity, which means a better heat transfer, than ethylene glycol's (E G) and ethyl alcohol's (E A). Heat transfer coefficients for different aqueous solutions are compared in the form of "heat transfer factor for turbulent flow" $F\alpha_{turb}$ in Fig. 6.15 in reference [1]. Second, the better heat transfer will lead to a smaller temperature difference crossing the heat exchanger, and this benefit can improve the energy saving potential in indirect systems. Third, Potassium Formate is also low toxic and non-flammable. Relatively low volumetric heat capacity is the only drawback. The comparison can be found in the Example 6.12 in reference [1]. However, the smaller pressure drop saving potential due to its low viscosity (Fig. 6.10, reference [1]) can cover the extra energy consumption caused by pumping the higher rate of mass flow in potassium formate systems. All of these benefits make potassium formate favorable for supermarket application.

✓ **CALCIUM CHLORIDE**

Calcium Chloride ($Cacl_2$) is the second one with also high thermophysical properties. Calcium Chloride is corrosive in the presence of air. Although it can be avoid by addition of inhibitors, the toxicy prevent its application in the food production and retail industries. Nevertheless, the non-toxic inhibitor is already available from recent technologies, which makes calcium chloride acceptable for many new commercial fields. Such research on calcium chloride based secondary refrigeration systems for supermarket display cabinets is already launched in UK. [18]

✓ **AQUEOUS SOLUTION USED IN TESTS**

Many kinds of new secondary refrigerants, including some traditional water based refrigerants, are patented and commercially produced in recent days. **Aspen Temper**® is such a representative of new refrigerants used in the tested systems. Like other latest commercial refrigerants, temper is a group of blend coolants. Temper –20, Temper –40, and Temper –60 are defined base on different freezing points. They are all composed by carboxylic (potassium acetate and potassium formate), water, and corrosion inhibitors. Temper works with good performance in the SQD system as well as other FRIGOTECH installations.

Compared with ethylene glycol and propylene glycol in the same temperature range, Temper has many advantages. (All the technical data come from company information sheets.) First, reduced liquid flows can be achieved by using Temper due to its high volumetric heat capacity, and it leads to smaller dimensions and pump size. Second, high heat conductivity causes better heat transfer in heat exchangers and energy can be saved from smaller temperature difference. Third, Temper has low viscosity, first cost can be reduced by smaller pumping size and operating cost can also be reduced by low energy consumption. Unlike carbon dioxide, Temper can be used for a wider temperature range, such as medium-temperature refrigeration and secondary condensing cycle in supermarket application. Unfortunately, no comparison data between temper and potassium formate is available from the information sheets. Some further comparisons between Temper systems and other secondary systems are needed from practical experience.

In general, due to the rapid developing technologies, both primary and secondary refrigerants will be continuously updated and improved to fulfill different refrigeration as well as energy and environmental requirements.

2.2 ALTERNATIVE SYSTEMS

Accompanied with the researching of alternative refrigerants, some alternative systems are also introduced to the supermarket refrigeration field. Most of them are indirect or cascade systems. They can be used for both chill and freeze refrigeration with HCs or natural refrigerants charging. Moreover, a modern total air system was also found and discussed in the following part.

1. A NH_3/ CO_2 CASCADE REFRIGERATION PLANT FOR SUPERMARKETS [14]

Norild, a leading contracting company for refrigeration systems for supermarkets in Norway, had built a NH_3/ CO_2 cascade refrigeration plant in a test room in their factory, with ambient conditions of +25 ˚C and 60% relative humidity.

The chilling load is covered by a NH_3 compression cycle, while a CO_2 compression cycle is cascaded into the system by using NH_3 evaporator as CO_2 condenser. The high temperature ammonia circuit is equipped with a heat recovery condenser in addition to the outdoor air condenser. The cooling purposes are provided at intermediate pressure level with a propylene glycol or a binary ice circuit for comparison. The CO_2 evaporators for the freezing room and freezing cabinets are traditional R22 evaporator with specific changing adapted to the CO_2 characteristics.
The evaluation indicates that the plant total efficiency may be improved by the superior air cooler efficiency of CO_2, and the higher CO_2 compressor efficiency.

2. A CENTRAL AMMONIA REFRIGERATION PLANT WITH ICE ACCUMULATION AND ICE/WATER/ETHANOL CIRCUIT FOR COOLING AND CARBON DIOXIDE FOR FREEZING [19]

Ammonia is used for production of ice slurry. The ice slurry is circulated to the cabinets in the supermarket. Carbon Dioxide is used to meet the cooling load at –30 ˚C by means of a CO_2 compressor. Sabroe Refrigeration, Danfoss, the Danish Technical University and the Danish Technological Institute are together investigating new indirect systems for supermarket refrigeration based on the principle as shown in figure 23.

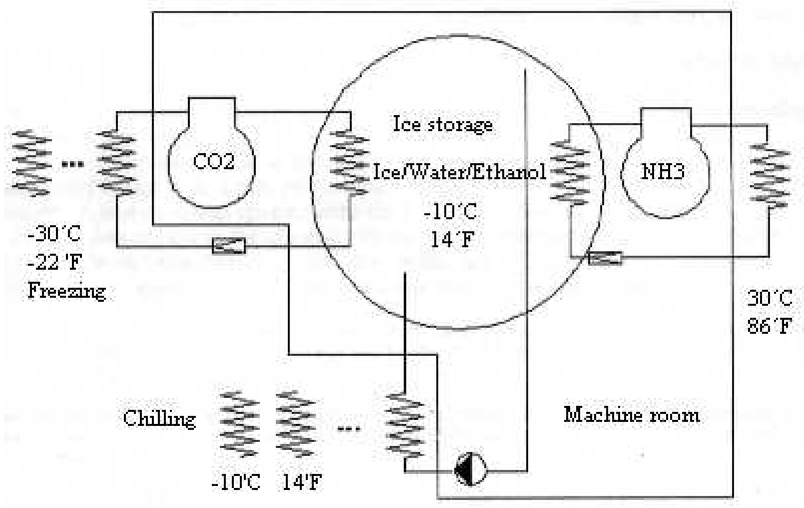


Figure 23. NH_3/CO_2 Plant with Ice Storage

3. A NH_3 CASCADE REFRIGERATION SYSTEM WITH CO_2 AS SECONDARY COOLANT IN LOW STAGE FOR SUPERMARKETS [20]

A supermarket, ICA FOCUS in Lund, Sweden, had been converted to a NH_3 cascade system with secondary propylene glycol for chilling and secondary CO_2 for freezing. The condenser for the low stage compressor is cooled by the glycol system at –8 ˚C. both the high and the low stages use ammonia with gravity-fed evaporators. In the CO_2 circuit, CO_2 liquid is pumped out to the evaporators where a portion of it is boiled off. A mixture of liquid and gas then returns to the condenser and the receiver/pump unit. (See figure 24)

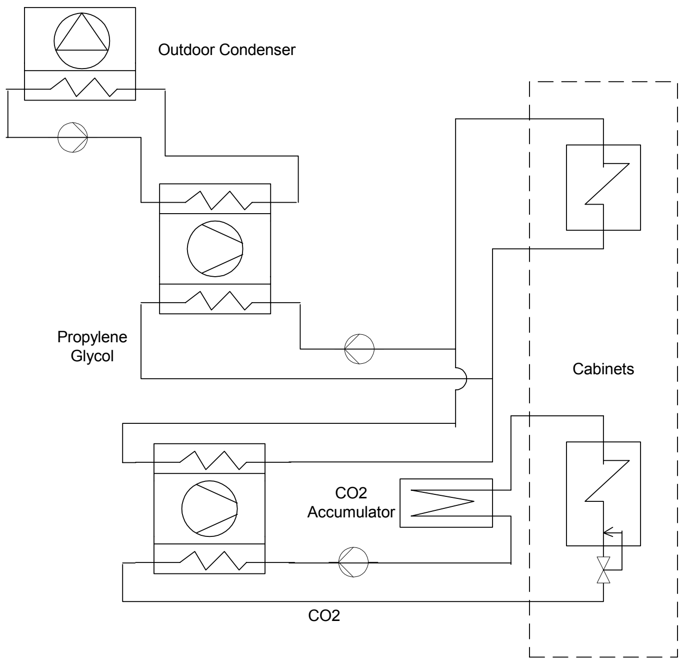


Figure 24. NH_3 Cascade System with CO_2 as Secondary Refrigerant

Investment evaluation was made for the NH_3 system, and the results show that the initial cost for NH_3 system is higher than other's due to the extra environmental risk, but the ammonia plant has better performance than any other solutions, which lead to a lower running cost.

The Carbon Dioxide secondary system is superior to other options from both initial and operating points of view. This part was detail discussed in the economic section.

4. AIR CYCLE REFRIGERATION IN SUPERMARKET APPLICATION [21]

Air cycle refrigeration had been installed in two purpose-built supermarket display cabinets, and one of them is a frozen well. Such "open" system is used in the case by directly delivering the handled air into the cold space. The compressed air is supplied from a remote air compressor. One recuperative heat exchanger is introduced to the system to improve the efficiency. The system configuration is illustrated in figure 25.

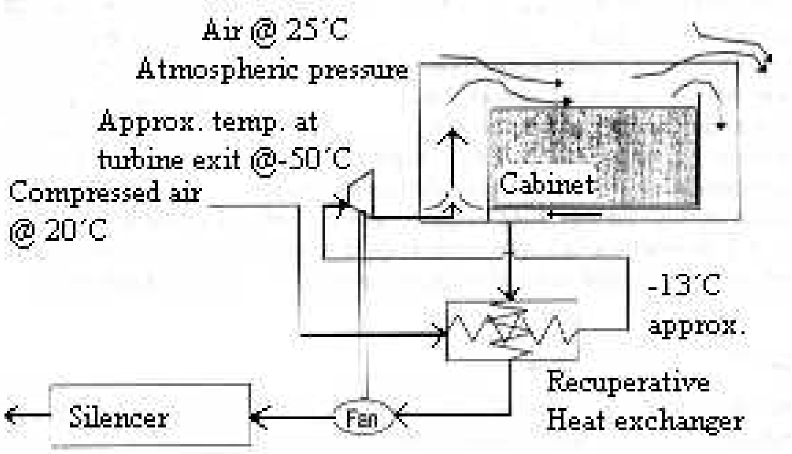


Figure 25. The Air Cycle Frozen Well Cabinet

Two specific problems were solved in the application, which are icing/de-icing and noise control. In the normal operation, the moisture is absorbed by an adsorbent drier. While in the de-icing period, defrosting can be achieved by bypassing the recuperator flow from the well. The formation of ice within the well is inhibited by the flow of very dry air from the cold air supply, and any deposited ice tending to evaporate. The noise generated by the turbine is eliminated by some special designs, such as using cell foam, and a silencer was added at the downstream of the fan to absorb its noise.

The mean air temperatures in the frozen well are in the range –27.4 ˚C to –37.6 ˚C. The effectiveness of the recuperation is 60%. The total system efficiency can be improved by using high performance turbines.

2.3 ECONOMIC EVALUATION

The previous Life Cycle Cost Analysis (LCCA) has proved that test completely indirect system (SQD) is economically better than test partially indirect (secondary condensing cycle) system (DX). Moreover, there also exist some other competitive indirect systems. Carbon Dioxide is chosen as a representative and discussed as follows.

Due to the increasing utilization of carbon dioxide refrigerant, total investments for CO_2 secondary systems are more and more concerned in today's food-storage refrigeration field. From the ICA-FOCUS case and many other field experience, it can be concluded that both initial and operating cost for CO2 secondary systems can be maintained in the same level or even be lower than direct systems and other secondary systems.

First Cost: Compared with direct systems, a balance point can be found where the extra costs for a direct expansion piping system are the same as the costs for the extra CO_2 condenser and the pumping unit. More benefit can be get for larger CO_2 filling. [20] Compared with other conventional secondary system, CO_2 indirect systems are better, because much smaller heat exchangers are required and the size of the transport lines as well as their insulation thickness will be smaller. This is especially obvious in the supermarkets which have long distance between machine rooms and shopping areas.

Operating Cost: The problem of more energy consumption (raised by the extra temperature difference) is general in conventional secondary loops. Thanks to the high volumetric heat capacity and phase changing property, a CO_2 indirect system can afford one more heat exchanger (temperature difference) without consuming extra energy compared with a direct expansion system. And it makes CO_2 systems more competent than other solutions from minimizing operating cost point of view.

Another comparison was also made between CO_2 indirect systems and CO_2 cascade systems. The results indicated that the energetic and economic performances of the CO_2 indirect coolant system and of the CO_2 cascade system are of the same order of magnitude. The simplicity of the indirect secondary system makes it superior. [15]

The main drawback for CO_2 systems is the extra cost raised by security considerations. Workers need to be especially trained to install as well as operate CO_2 systems, and such cost should be added to system total costs. In addition, CO_2 systems' special operation and maintenance costs may vary a lot depending on specifics.

In general, carbon dioxide is a very promising refrigerant for low temperature application, because of its high heat transfer coefficient, heat of vaporization, and low viscosity. It can be used either in the low stage of a cascade system or in the secondary circuit of an indirect system. High-pressure systems and related safety control design as well as safe operation have to be especially concerned when a CO_2 system is built up.

3 COMPRESSION CYCLE MODIFICATION

Compressor is the most important component in the system. Any compressor energy saving methods can lead to a significant total saving result. Besides choosing correct capacity to operate compressors at highest efficiency, some other latest technologies have been introduced to refrigeration systems to reduce the energy usage. Both research and practical experience prove those principles can be used to minimize refrigeration system energy consumption.

3.1 LIQUID REFRIGERANT PUMPING (LRP)

The principle of Liquid Refrigerant Pumping (LRP) technology is to allow the traditional direct expansion systems changing their head pressure from a fixed set-point (minimum head pressure control) to a floating-head pressure control. Energy saving can be achieved whenever LRP system's condensing pressure (temperature) lower than traditional systems'. [22]

In a conventional direct system, head pressure is normally maintained in a high level in order to avoid the generation of flash gas between the condenser and the expansion valve. Therefore, the condensing temperature is fixed neglecting the changing of ambient air temperature. While a liquid deliver pump is installed between the outlet of the condenser and upstream of the expansion valve. The liquid pressure is increased before it enters the expansion valve, and this prevents the formation of flash gas. The mechanism allows the using of floating-head pressure control to decrease the compressor discharge pressure and condensing temperature whenever the ambient temperature becoming colder. The decreasing will lead to a significant energy saving from the compressor by the reduction of both power consumption and operating time. Other benefits can also be made from the extending of equipment life and the reduction of maintenance cost of compressors in LRP systems. The energy saving mechanism can be explained from figure 17. In addition, desuperheating can be minimized in the condenser by injecting liquid refrigerant from the outlet of a liquid pump to compressor discharge line, in order to achieve higher condenser efficiency.

LRP technology can be applied to most of conventional direct refrigeration systems that use thermostatic expansion valves with minimum head pressure control and outdoor placed condensers. It is suitable to cold climate or the place where the cold period is dominant through the year.

An Internet searching found that some companies, such as Hy-save, Inc. UK, could provide such service from system design to installation. Some LRP systems were already installed in US. and Canada. Technical evaluation showed that energy savings typically range from 10 to 30 % with paybacks under two years. [22]

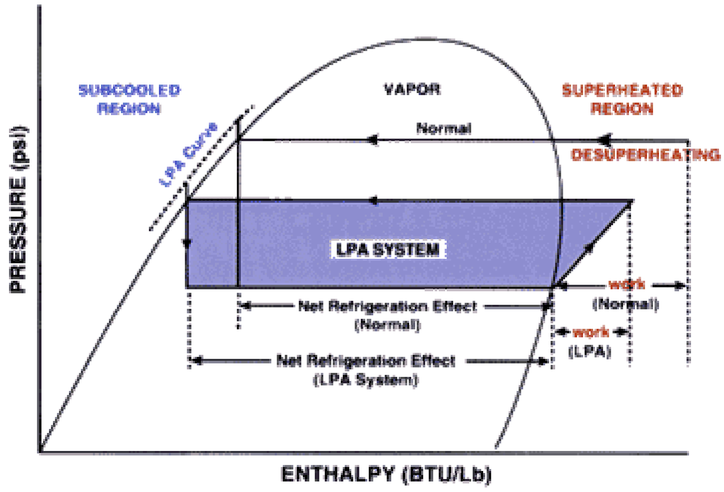


Figure 26. LRP Energy Saving Mechanism [23]

Both tested direct and indirect systems in on-the spot experiment use secondary condensing temperature control to prevent the formation of flash gas. A three-way valve is placed at the inlet of the condenser on secondary dry cooler loop, and the inlet temperature is fixed at a preset point by a control box connected with a thermal sensor. (See DX and SQD system configuration) Therefore, if only the temperature is maintained in a correct range, no vapor will form before the expansion valve. The same secondary loop is shared and the same mechanism is used by other refrigerating plants in the machine rooms. Such LRP technology can be used for current DX system's retrofitting. However, the extra sub-coolers in SQD system can already guarantee the liquid delivering. The effect of installing LRP pumps will be reduced. Therefore, the LRP technology is not necessary in this case. But generally, it can be used on either direct or indirect systems.

- **Liquid Refrigerant Pump for Supermarket Refrigeration**

A case study was taken from a 9-ton refrigeration system in the commissary at Fort Drum, Watertown, New York, US. The retrofitting was made on one of its low-temperature refrigeration racks that provide cooling to refrigerated store rooms and display cases for frozen food. The result indicates that LRP rebuilt can achieve 17.26% total energy reduction with the payback of 1.6 years. [22]

3.2 VARIABLE SPEED DRIVE (VSD)

Variable Speed Drives (VSD) work by converting AC signals to DC using rectifiers and then inverting these DC signals back to AC. There are several types of VSD drive: frequency drives, flux vector divers, and servo drives. [24] The main propose of using VSD is to enable compressors work at full load efficiency neglecting the reduction of output capacity. Energy is saved from the increasing of compressor working efficiency. In refrigeration systems, compressors, condenser and evaporator fans can be equipped with VSD for energy saving considerations. A comparison had been made based on two different supermarket refrigeration systems that use on/off and VSD compressor controls for each. The simulation results indicate that VSD control system gain 27.8% energy savings over on/off control system. Besides, a higher COP is also achieved because VSD control provides more uniform and higher suction pressure, which leads to low compressor power consumption. [25] Only a practical application of VSD compressor for air conditioning system was found through the survey, but the principle can be adapted for supermarket refrigeration. The retrofit had been done on a building's air conditioning system which uses two York 615 kW centrifugal chillers. The monitoring results show that after adding VSD units on compressors and cooling tower fans, energy savings can reach 43% and 19% during the test period. [24]

Energy saving potential does exist in compressors and condenser fans in supermarket refrigeration systems in terms of variable speed drive retrofit. And one special point has to be mentioned is that much energy savings can also be achieved by using high efficiency evaporator fan motors in display cabinets, because the fans almost work 24 hours a day. Additionally, since evaporator fans contribute to refrigeration load, the use of high efficiency motors results in further savings through the reduction of compressor load.

The tested DX system uses indirect suction line pressure on/off control. The different point is that the temperature will control the suction pressure through an on/off valve on the suction line. While the SQD system employs a PLC (Programmable Logic Controller) to control the compressors on/off and the incoming signal is the secondary refrigerant outlet temperature after the evaporators. Such difference will not influence the installation and the operation of VSD drives in terms of system retrofit.

4 HEAT RECOVERY AND HVAC COMBINATION

The promotion of system efficiency is desired by all energy systems. In the system of supermarkets, around 50% of (electric) energy are consumed by refrigeration equipment. On the other hand, a lot of energy is rejected to atmosphere in the form of “waste heat” from refrigeration systems. Therefore, heat recovery is obviously necessary for reusing this part of energy. Normally, the recovered heat is used for space and water heating. Besides, high grade heating and cooling may be achieved from waste heat through the combination of heat pump technology.

There are two places for heat recovery processing in refrigeration systems. The first one is at refrigerant discharge line. A three-way valve is normally installed for directing hot refrigerant discharge to a heat exchanger in the central air handler where the heat is reclaimed. The refrigerant then passes to the condenser for rejecting the remaining heat. The flow rate can be adjusted according to the requirement. Heat recovery units can also be set on secondary condensing loops. This is usually used for centralized refrigeration systems with remote air-cooled or evaporative condensers.

4.1 BASIC UTILIZATION

Basic heat recovery is very easy to be realized. Simply by adding reclaim coils and fans on secondary condensing loops, the reclaimed heat can cover partial space heating needs. Such units were installed in the tested SQD system at HEMKÖP supermarket, Mörby Centrum, as auxiliary space heating components in the working area. The recovered heat can also be used for hot water supply. One example is that such typical system was installed in a poultry processing plant in Ontario, Canada. There are two stages in the system. The cold water is preheated to around 24.8 ˚C by using the heat from an ammonia refrigeration plant through a glycol & water circuit in the first stage. The second stage of the heat recovery occurs in a cascading condenser of a heat pump. While the heat source of the heat pump is the condenser for an ice machine system. The supply water has 40.6 ˚C average and 63 ˚C maximum temperatures. The system has operated under an average COP of 10.7 with the payback period of 2.9 years. It can be adapted for virtually all plants that require both refrigeration and hot water. [26]

4.2 ADVANCED UTILIZATION

High quality air conditioning is possible by the combination of heat recovery and HVAC systems. The outstanding point is that the efficiencies of both refrigeration and HVAC systems are highly improved. Following are some typical attempts.

- **A Supermarket Refrigeration System based on the Water Loop Heat Pump Concept.** [27]

A mathematical model of a supermarket refrigeration plant had been constructed within this case. The whole system is connected by a common secondary loop with a single-phase fluid running in it. The loop acts as a heat sink for the high temperature (HT) and low temperature (LT) refrigeration units. A heat recovery unit on the loop connects a heat pump for store heating and air conditioning. Any surplus heat from the secondary fluid is rejected to the outdoor environment with a primary chiller. (See figure 27) Several system layouts had been simulated based on different secondary fluid temperature as well as different HVAC integrations. The practical design is based on the best option coming from the comparison, which is secondary temperature chosen to provide the air conditioning cooling needs directly, and a heat pump used to meet the store's heating needs.

The conclusion is that the heat recovery and HVAC integration does give the highest efficiencies for both refrigeration and HVAC systems. The energy saving potential can be maximized at the same time. Another point is such secondary systems can perform better than tradition direct expansion systems in terms of energy use and TEWI (Total Equivalent Warming Impact).

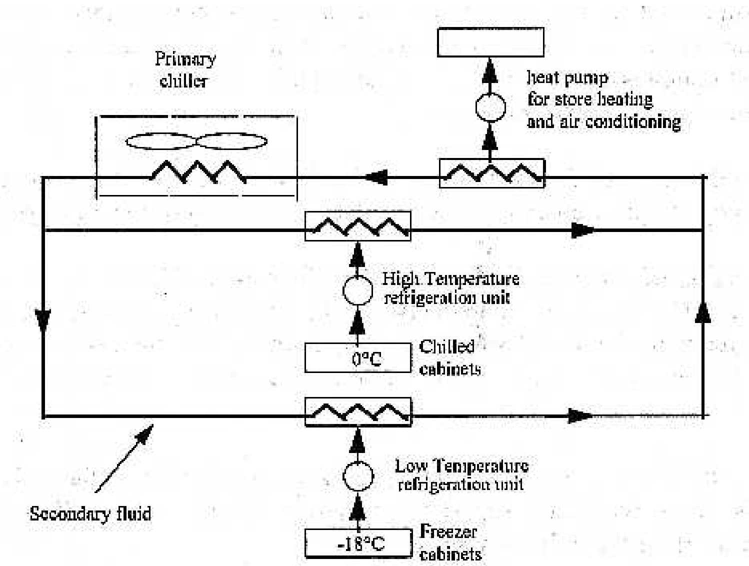


Figure 27. Schematic of Integrated Refrigeration and Store Heating System for a Supermarket

- **Dual-Path Heat Pump System Used in Supermarkets** [28]

A similar integrated system with dual-path heat pumps has been used in a variety of supermarkets and other buildings across the US. The common parts are heat recovery via a refrigeration desuperheater for water heating and heat recovery for space heating from the condensers via the water loop. The unique point is that the system's dehumidification (summer) and air heating (winter) are based on dual-path heat pumps linked to the water loop. A separate air flow path is used to dehumidify the ventilation air which is then mixed with the return air for delivery to the occupied space (see figure 28)

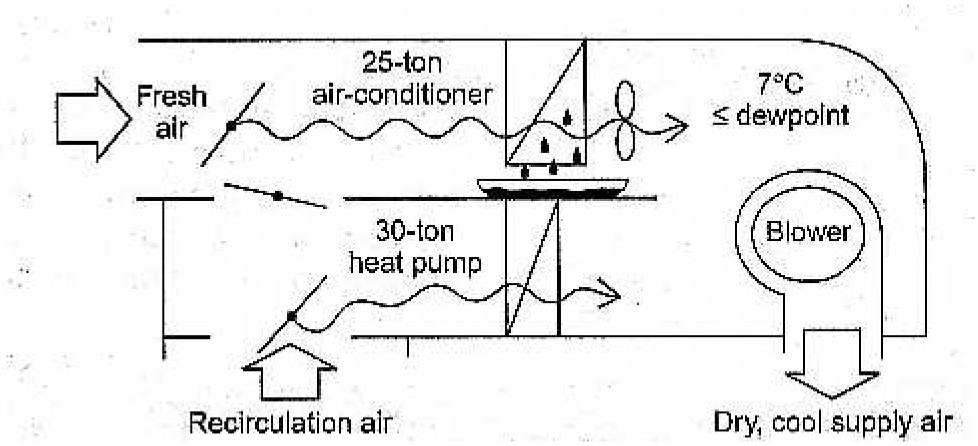


Figure 28. Configuration of the Dual-Path Heat Pump

Energy saving of 22% can be accomplished from the new HVAC system by separating and dehumidifying only the incoming humid ventilation airstream. After using dual-path heat pumps, the space humidity remain in consistently low levels, which leads to a further energy saving from supermarket refrigeration systems by the reduction of both ice formation and defrosting. Variable air volume is controlled by CO_2 and indoor air quality sensors to ensure adequate ventilation. This dual-path heat pump heat recovery system is almost the best solution for the supermarket refrigeration.

5 THERMAL ENERGY STORAGE (TES)

Thermal Energy Storage (TES) is a technology of spreading the day time load over 24 hours period in order to reduce refrigeration system capacity and even save energy. It is especially suitable to the place where has different on-peak and off-peak electricity price, because the operating cost can be reduced greatly by shifting on-peak running to off-peak storage. There are two different methods of TES. Full storage and partial storage. The first method enable a chiller operate consistently during the night to shift the entire load into off-peak hours, and the chiller does not run at all during the day. This is mainly used in air conditioning systems. Partial storage system is more attractive for supermarket refrigeration. Smaller compressors are selected to run 24 hours a day. It charges the thermal storage component and covers the relatively lower cooling load at night, while provide the higher cooling capacity during the day with the help from stored cooling. In this case, the chillers are always working under full load condition; therefore the compressor efficiency can be maximized. On-peak demand charges (if there exists) are reduced due to the help of stored cooling. One subsidiary benefit is that total energy usage is lower because air-cooled condensers perform most efficiently when the outdoor temperatures are relatively low during the night.

The most commonly used thermal storage material is water (for usage higher than 0 ˚C) and some phase changing material, such as ethylene glycol water solution (for usage lower than 0 ˚C). Due to limitation of applicable temperature range, currently, thermal storage can not directly be used for supermarket refrigeration which normally has – 8 ˚C and – 24 ˚C supply air temperatures. However, a literature survey find that a company named CALMAC already used salt eutectics for low temperature thermal storage, and the energy can be stored at –11 ˚C with discharge temperature varying from –9 ˚C to –4 ˚C. [29] That means medium temperature refrigeration systems can also use TES directly. But there is no such supermarket application yet from company's information. Other indirect application are using ice storage for condenser cooling or subcooling, which are very attractive for low temperature systems, since low condensing temperature, such as 10 ˚C, can be maintained during the storage energy discharge cycle.

In general, thermal energy storage is a useful tool to reduce system installation capacity as well as save energy and operating cost for supermarket refrigeration.

6 ENERGY LABELING OF SUPERMARKET CABINETS

An EU project to evaluate energy labeling of supermarket refrigerated cabinets has recently been completed by FRPERC in collaboration with TNO in the Netherlands and the DTI in Denmark. The aim of the work is to provide an energy indicator that would give users information on the energy efficiency of retail cabinets in a comparable manner. [30] All kinds of cabinets from different manufacturers are divided in different categories based on the type of use intended for the cabinet. Energy consumption and functionality are determined and compared based on different manufacturer representative models within each category. All measurements are obtained from EN441, a recently introduced European Test Standard. Finally, the Energy Efficiency Index, which ranges from A (the most efficient) to G (the least efficient), can be calculated from the actual energy consumption divided by the standard energy consumption. The results do show that in most categories there are cabinets with the same functionality that use three times more energy than others within the same category. The introducing of energy labeling of supermarket refrigeration cabinets will absolutely direct to two effects: Users are allowed easily to choose the most efficient cabinets and manufacturers strive to produce more efficient cabinets. And all of these are done to realize one aim – energy saving.

7 DISCUSSION AND CONCLUSION

Considering all different advanced technologies in previous discussions, some combinations can be made in one refrigeration system to achieve the maximum efficiency. Such combinations could be Liquid Refrigerant Pumping (LRP) and heat recovery, or Variable Speed Drive (VSD) and heat recovery. The utilization of high efficiency motors on fans and pumps will always be beneficial to total system energy savings.

An advanced supermarket refrigeration system could be described as follows. The whole system is connected by a common secondary loop circulated with single-phase fluid, such as water. The loop acts as a heat sink for both High Temperature (HT) and Low Temperature (LT) refrigeration units. Dual path heat pumps are connected with the loop to recover the waste heat and provide supermarket air conditioning. The waste heat generated from secondary condensing cycle is further used by warm liquid defrosts in both high and low temperature circuits. Liquid Refrigerant Pumps (LRP) are used in both HT and LT circuits. Compressors can use HCs or ammonia as refrigerant. Thermal Energy Storage (TES) is used in HT circuit to provide cooling to HT cabinets during the daytime, and subcooling to LT circuit. Ethylene Glycol or other Phase Changing Materials (PCMs) can be used in the TES circuit. A potassium formate secondary loop is used on LT evaporating cycle. All the fans and motors are driven by high efficiency motors. Compressors can be served by variable speed drive. Such system's configuration can be seen from figure 29.

Theoretically, a highest system performance and a maximum total efficiency can be achieved by such design, but the practical conditions will decide the degree of combinations. More advanced combinations will lead to more complex system design and higher system cost. The optimum point does exist between the degree of combination and the performance of refrigeration systems.

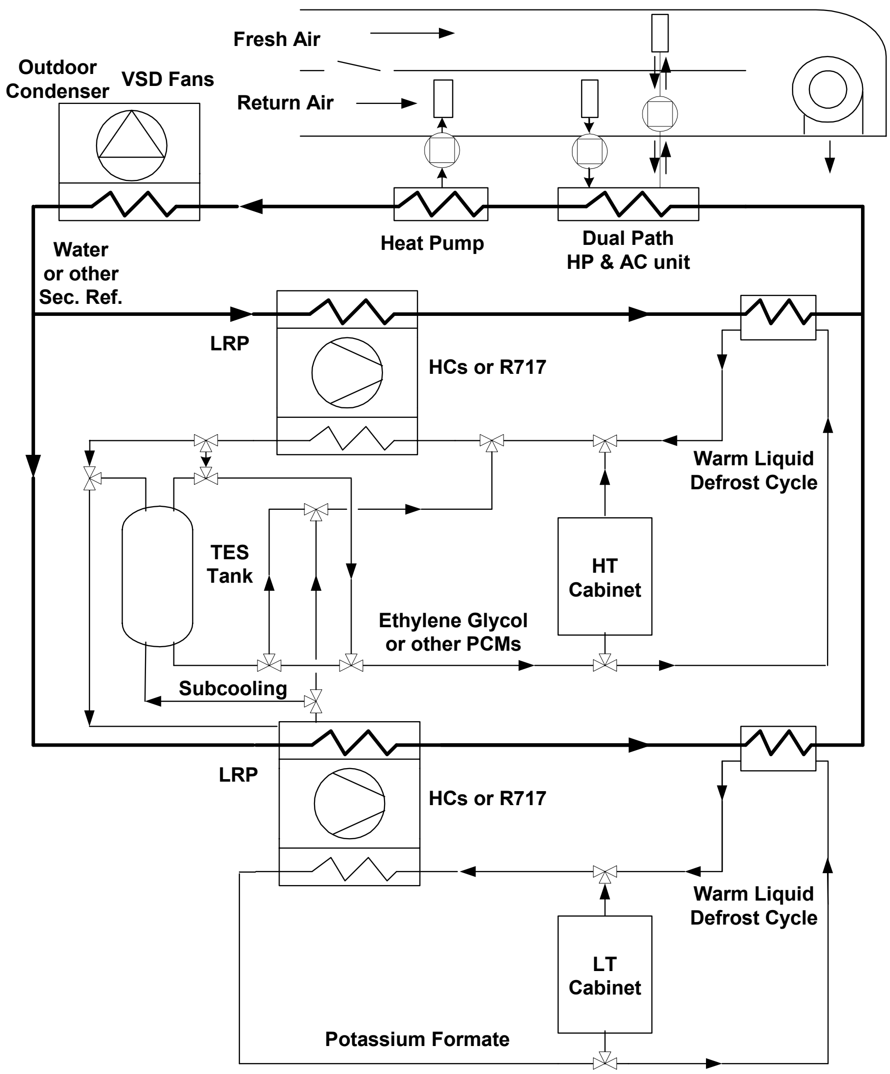


Figure 28. Configuration of an Ideal Supermarket Refrigeration System

GENERAL CONCLUSION

The first part of intensive project study shows that a representative completely indirect system is better than a representative partially indirect system, which has a secondary condensing circuit, in terms of system performance, energy consumption, total investment and environmental impact. Moreover, such results are applicable to other similar systems.

Many types of promising refrigerants, alternative systems as well as refrigeration system optimization methods were extensively explored in the second part of the investigation. The results indicate that, on the one hand, a lot of retrofitting and update work an be done based on current existing traditional refrigeration system, on the other hand, many latest technologies can be used when building up new systems. The possibility of building up an ideal supermarket refrigeration system by the combination of all possible methods was also discussed. A serious of tests and experiments should be further arranged to confirm the theoretical study results for completing the project.

In general, two predetermined sub-purpose were achieved in the two separated parts of a thorough Deep-Freeze Refrigeration System Investigation.

APPENDIX

APPENDIX A. SPECIFIED TEST SYSTEM INSTALLATION DESCRIPTION

- **A1.** SPECIFIED DX SYSTEM INSTALLATION DESCRIPTION

Compressor
1 Semi-Hermetic Motor Compressor # BITZER® 4N-12.2Y model
Max power consumption: 14.1 kW
Performance (charged with R404A):
Condensing temperature: 30 °C
Evaporating temperature: -35 °C
Refrigerating capacity: Q=11.61 kW
Power consumption: P=6.42 kW

Lubricant
Volume: 3.0 dm^3 Polyester oil # BITZER® (factory charged)

Refrigerant
Primary circuit R404A Amount: 18 kg
Dry cooler circuit Propylene Glycol

Liquid Receiver
1 Tank Liquid Receiver Volume: 24 L # BITZER® YF-240-S model

Heat Exchanger
TAU® Brazed Plate Heat Exchangers
1 Condenser (Dry cooler side) # H25-50LL model
1 Condenser (Municipal water backup) #H25-24LL model

Display Cabinet
WICA COLD® Storage Display Cabinets

Table 19a. Cabinet Parameters at Sundbyberg

Model	Amount	Gross Volume	Evaporating Temp.	Supply Temp.	Rated Capacity 100% (24hr)
		L	°C	°C	W
FLG-25	2	1 210	-33	-30	1 510
FLG-37	1	1 815	-33	-30	2 265
DLG-11	1	428	-33	-31	780
				Total	6 065

Dry Cooler
Several dry cooler units put in one group serves all the refrigerating systems for the supermarket.
ASARUMS INDUSTRI® KKVB-13 model

- **A2.** SPECIFIED SQD SYSTEM INSTALLATION DESCRIPTION

Compressor

2 Semi-Hermetic Motor Compressors # BITZER® 4J-13.2Y model
Max power consumption: 15.7 kW
Performance (charged with R404A):
Condensing temperature: 30 °C
Evaporating temperature: -35 °C
Refrigerating capacity: Q=13.53 kW
Power consumption: P=7.26 kW

Lubricant

Volume: 4.0 dm^3 Polyester oil # BITZER® (factory charged)

Refrigerant

Table 20. Refrigerant in SQD

Location	Type	Amount
Primary circuit	R404A	7 kg
Sec. Evaporator circuit	Temper® 40	732 kg *
Dry cooler circuit	Propylene Glycol	--
Sub-cooler circuit	Temper 20	--

* ρ=1.22kg/liter, $M = \rho \times V$ =1.22 kg/L * 600 L=732 kg [6]

Heat Exchanger

TAU® Brazed Plate Heat Exchangers
2 Condensers # H55-30LL model
2 Evaporators # M55-40LL model
2 Sub-coolers # M18-40-LG model
1 SQD defrosting plate heat exchanger # ALFA LAVAL® CB26-36 model

Display Cabinet

System contains three cabinet groups named Group A, B, and C.
WICA COLD® Storage Display Cabinets

Table 19b. Cabinet Parameters for Group A at Mörby

Model	Amount	Gross Volume	Evaporating Temp.	Supply Temp.	Rated capacity 100% (24hr)
		L	°C	°C	W
FLG-37	2	1 815	-33	-30	2 265
DLG-11	2	428	-33	-31	780
			Subtotal		6 090

Table 21. Cabinet Parameters for Group B and C in SQD System

Model	Amount	Rated capacity 100% (24hr) W
FHG-37	2	2 540
DHG-11	2	880
	Subtotal	6 840

Dry Cooler
Several dry cooler units put in one group serves all the refrigerating systems for the supermarket.
ASARUMS INDUSTRI® RCHB-21 model

APPENDIX B. EES® PROGRAM FOR SYSTEM COP CALCULATION

- **B1.** DX SYSTEM COP CALCULATION PROGRAM

```
"Calaulation for The Test Direct Refrigeration System"
p1=16.081*100
p2=1.409*100

T1k=89.156
T1s=32.514

T2k=-12.511

h1k=ENTHALPY(R404A,T=T1k,P=p1)
h1s=ENTHALPY(R404A,T=T1s,P=p1)
h2k=ENTHALPY(R404A,T=T2k,P=p2)
h2s=h1s
COP2=(h2k-h1s)/(h1k-h2k)

h1=ENTHALPY(R404A,x=0,P=p1)
T2s=TEMPERATURE(R404A,h=h1s,P=P2)


T2kr=-25.598
h2kr=ENTHALPY(R404A,T=T2kr,P=P2)

T1sr=28.824
h1sr=ENTHALPY(R404A,T=T1sr,P=P1)

COP2r=(h2kr-h1sr)/(h1k-h2k)

Pr[1]=P1;H[1]=H1k
Pr[2]=P1;H[2]=H1
Pr[3]=P1;H[3]=H1s
Pr[4]=P2;H[4]=H1s
Pr[5]=P2;H[5]=H2kr
Pr[6]=P2;H[6]=H2k
Pr[7]=P1;H[7]=H1k

Pr[8]=P1;H[8]=H1sr
Pr[9]=P2;H[9]=H1sr
Pr[10]=P2;H[10]=H1s
```

- **B2.** SQD SYSTEM COP CALCULATION PROGRAM

```
"Calaulation for The Test Indirect Refrigeration System"
p1=16.339*100
p2=1.591*100

T1k=75.146
T1s=-5.554

T2k=-28.965

h1k=ENTHALPY(R404A,T=T1k,P=p1)
h1s=ENTHALPY(R404A,T=T1s,P=p1)
h2k=ENTHALPY(R404A,T=T2k,P=p2)
h2s=h1s
COP2=(h2k-h1s)/(h1k-h2k)

h1=ENTHALPY(R404A,x=0,P=p1)
T2s=TEMPERATURE(R404A,h=h1s,P=P2)

Pr[1]=P1;H[1]=H1k
Pr[2]=P1;H[2]=H1
Pr[3]=P1;H[3]=H1s
Pr[4]=P2;H[4]=H1s
Pr[5]=P2;H[5]=H2k
Pr[6]=P1;H[6]=H1k
```

APPENDIX C. LIFE CYCLE COST DISCOUNT FACTORS

- **C1.** TABLE OF SPV DISCOUNT FACTOR

SPV Factor Table from Annual Supplement to Handbook 135

Table A-1. SPV factors for finding the present value of future single amounts (non-fuel, 1995)

Single Present Value (SPV) Factors

Year of Occurrence (t)	DOE Discount Rate 3.0%	OMB Discount Rates[a] Short Term[b] 2.5%	OMB Discount Rates[a] Long Term[c] 2.8%
1	0.971	0.976	0.973
2	0.943	0.952	0.946
3	0.915	0.929	0.920
4	0.888	0.906	0.895
5	0.863	0.884	0.871
6	0.837	0.862	0.847
7	0.813	0.841	0.824
8	0.789	0.821	0.802
9	0.766	0.801	0.780
10	0.744	0.781	0.759
11	0.722		0.738
12	0.701		0.718
13	0.681		0.698
14	0.661		0.679
15	0.642		0.661
16	0.623		0.643
17	0.605		0.625
18	0.587		0.608
19	0.570		0.592
20	0.554		0.576
21	0.538		0.560
22	0.522		0.545
23	0.507		0.530
24	0.492		0.515
25	0.478		0.501
26	0.464		0.488
27	0.450		0.474
28	0.437		0.462
29	0.424		0.449
30	0.412		0.437

[a] OMB discount rates as of March 1994. OMB rates are expected to be revised in February 1995.
[b] Short-term discount rate based on OMB discount rate for 7-year study period.
[c] Long-term discount rate based on OMB discount rate for 30-year study period.

- **C2.** TABLE OF UPV DISCOUNT FACTOR

UPV Factor Table from Annual Supplement to Handbook 135

Table A-2. UPV factors for finding the present value of future single amounts (non-fuel, 1995)

Uniform Present Value (UPV) Factors

Year of Occurrence (t)	FEMP Discount Rate 3.0%	OMB Discount Rates[a] Short Term[b] 2.5%	OMB Discount Rates[a] Long Term[c] 2.8%
1	0.97	0.98	0.97
2	1.91	1.93	1.92
3	2.83	2.86	2.84
4	3.72	3.76	3.73
5	4.58	4.65	4.61
6	5.42	5.51	5.45
7	6.23	6.35	6.28
8	7.02	7.17	7.08
9	7.79	7.97	7.86
10	8.53	8.75	8.62
11	9.25		9.36
12	9.95		10.07
13	10.63		10.77
14	11.30		11.45
15	11.94		12.11
16	12.56		12.76
17	13.17		13.38
18	13.75		13.99
19	14.32		14.58
20	14.88		15.16
21	15.42		15.72
22	15.94		16.26
23	16.44		16.79
24	16.94		17.31
25	17.41		17.81
26	17.88		18.30
27	18.33		18.77
28	18.76		19.23
29	19.19		19.68
30	19.60		20.12

[a] OMB discount rates as of March 1994. OMB rates are expected to be revised in February 1995.

[b] Short-term discount rate based on OMB discount rate for 7-year study period.

[c] Long-term discount rate based on OMB discount rate for 30-year study period.

- **C3.** TABLE OF UPV (ENERGY) DISCOUNT FACTOR

Exhibit 3-3
UPV* Discount Factor Table from Annual Supplement to Handbook 135

Table Ba-1. UPV* Discount Factors adjusted for fuel price escalation, by end-use sector and fuel type, FY 1995
Discount rate = 3.0 percent (FEMP)

Census Region 1 (Connecticut, Maine, Massachusetts, New Hampshire, New Jersey, New York, Pennsylvania, Rhode Island, Vermont)

	RESIDENTIAL				COMMERCIAL					INDUSTRIAL					TRANSPORT	
N	ELEC	DIST	LPG	NTGAS	ELEC	DIST	RESID	NTGAS	COAL	ELEC	DIST	RESID	NTGAS	COAL	GASLN	N
-	-----	-----	-----	-----	-----	-----	-----	-----	-----	-----	-----	-----	-----	-----	-----	-
1	0.98	0.98	0.98	0.99	0.96	0.98	1.01	1.00	1.00	0.96	0.98	1.01	1.00	0.95	1.01	1
2	1.93	1.94	1.95	1.96	1.90	1.94	2.03	1.98	1.97	1.89	1.95	2.04	1.99	1.88	2.02	2
3	2.86	2.88	2.91	2.92	2.80	2.90	3.07	2.94	2.92	2.79	2.91	3.08	2.96	2.78	3.02	3
4	3.77	3.82	3.85	3.86	3.69	3.86	4.10	3.89	3.84	3.66	3.86	4.12	3.93	3.74	4.00	4
5	4.65	4.75	4.78	4.77	4.55	4.81	5.13	4.82	4.75	4.52	4.82	5.15	4.88	4.69	4.97	5
6	5.53	5.67	5.71	5.67	5.38	5.76	6.16	5.73	5.63	5.35	5.78	6.19	5.81	5.56	5.93	6
7	6.38	6.58	6.62	6.56	6.20	6.71	7.20	6.63	6.52	6.16	6.73	7.25	6.75	6.41	6.88	7
8	7.22	7.49	7.53	7.43	7.00	7.67	8.26	7.52	7.38	6.96	7.69	8.31	7.67	7.23	7.81	8
9	8.05	8.39	8.42	8.27	7.77	8.62	9.33	8.39	8.22	7.72	8.65	9.39	8.58	8.03	8.73	9
10	8.86	9.28	9.30	9.12	8.53	9.56	10.39	9.26	9.04	8.47	9.60	10.46	9.49	8.81	9.64	10
11	9.65	10.15	10.17	9.95	9.26	10.50	11.44	10.12	9.82	9.19	10.54	11.53	10.39	9.57	10.53	11
12	10.43	11.01	11.02	10.77	9.97	11.42	12.50	10.96	10.57	9.90	11.47	12.60	11.28	10.30	11.40	12
13	11.20	11.86	11.85	11.58	10.67	12.33	13.56	11.81	11.31	10.59	12.38	13.68	12.18	11.02	12.26	13
14	11.95	12.69	12.68	12.40	11.35	13.23	14.61	12.66	12.02	11.27	13.29	14.75	13.07	11.73	13.11	14
15	12.71	13.51	13.48	13.20	12.02	14.12	15.63	13.50	12.71	11.93	14.19	15.78	13.96	12.43	13.94	15
16	13.44	14.32	14.28	13.98	12.66	14.99	16.64	14.32	13.37	12.58	15.08	16.81	14.84	13.10	14.75	16
17	14.17	15.11	15.06	14.76	13.28	15.86	17.65	15.14	14.02	13.20	15.95	17.84	15.71	13.77	15.55	17
18	14.88	15.89	15.84	15.52	13.88	16.71	18.65	15.94	14.66	13.82	16.81	18.86	16.57	14.42	16.34	18
19	15.57	16.65	16.60	16.27	14.47	17.56	19.65	16.73	15.28	14.41	17.66	19.88	17.42	15.05	17.12	19
20	16.26	17.40	17.35	17.01	15.04	18.39	20.64	17.51	15.88	14.99	18.50	20.90	18.27	15.68	17.89	20
21	16.92	18.14	18.08	17.74	15.60	19.21	21.63	18.28	16.47	15.56	19.33	21.91	19.11	16.29	18.65	21
22	17.58	18.86	18.81	18.45	16.13	20.01	22.61	19.04	17.05	16.12	20.15	22.92	19.93	16.90	19.39	22
23	18.22	19.57	19.52	19.15	16.66	20.81	23.59	19.79	17.62	16.65	20.96	23.93	20.75	17.49	20.13	23
24	18.85	20.27	20.23	19.85	17.17	21.60	24.56	20.53	18.17	17.18	21.76	24.94	21.56	18.07	20.86	24
25	19.46	20.96	20.92	20.53	17.66	22.38	25.52	21.26	18.71	17.69	22.55	25.94	22.35	18.63	21.57	25
26/a	20.06	21.63	21.60	21.20	18.14	23.14	26.49	21.98	19.24	18.19	23.33	26.94	23.14	19.19	22.28	26
27/a	20.65	22.30	22.27	21.86	18.60	23.90	27.44	22.68	19.76	18.68	24.10	27.94	23.91	19.74	22.98	27
28/a	21.23	22.95	22.94	22.51	19.06	24.64	28.40	23.38	20.26	19.15	24.86	28.94	24.68	20.28	23.66	28
29/a	21.80	23.59	23.59	23.14	19.50	25.38	29.34	24.07	20.75	19.62	25.61	29.93	25.44	20.80	24.34	29
30/a	22.35	24.22	24.23	23.77	19.92	26.11	30.29	24.75	21.24	20.07	26.35	30.92	26.19	21.32	25.01	30

APPENDIX D. *PHOTO ALBUM*

Figure 29a. DX System Display Cabinet during Plastic Chicken Test Period

Figure 29b. DX System Machine Room

Figure 30a. SQD System Display Cabinet during Plastic Chicken Test Period

Figure 30b. SQD System Machine Room

Figure 31a. DX System Display Cabinet during Tinytalk Test Period (1)

Figure 31b. DX System Display Cabinet during Tinytalk Test Period (2)

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