

Cold storage of food

Review of available information on energy consumption and energy savings options.

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Summary

FRPERC has carried out a detailed energy study of 3 cold store complexes. The data from those studies has been augmented with other published information on cold stores in different parts of the world in this report.

The first section of this report compares all the energy consumption data and the following sections look in detail at the three stores investigated.

Comparison data is presented for chilled, frozen and mixed (chilled and frozen) stores. It is clear from the data that a large range in efficiencies exists. In all cases the worst cold store consumed over 8 times the most efficient store.

In the ammonia store the survey showed how a plant that was originally designed for efficient operation has altered over the years and now consumes approximately 43% more energy than was originally intended.

It is expected that a two-stage ammonia plant with flash intercooler, pumped recirculation and evaporative condensers would be very efficient. However, poor part load isentropic efficiency of the compressors, a lack of balance between high and low stage compressor capacity and the need for maintenance has turned what should have been an efficient plant into an inefficient one.

To keep things in perspective, the estimated COSP of this plant was still better than an R22 direct expansion system with electric defrost serving a similar cold store complex.

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Introduction

There is very little published data on the energy consumption of cold storage systems for foods. A detailed study has been carried out of three UK cold stores and data obtained from other published sources.

The first section of this report compares all the energy consumption data and the following sections look in detail at the three stores investigated.

In the DX stores substantial savings could be achieved if operation of cold storage facilities were optimised in terms of heat loads on the rooms and the operation of the refrigeration system. Many improvements were low in cost (improved door protection, defrost optimisation and repairs) and most other improvements were cost effective and had short pay back times. Cold store 2 was the most efficient store in terms of heat removed by the refrigeration system and this was due to it being operated by a low-pressure receiver system. However, the plant used more energy per m³ and this was partly due to it being a small store and also to the way the store was operated with high transmission, infiltration and fixed heat loads. Improvements to its operation could be made by fitting better door protection and reducing fixed loads. Substantial improvements could be made to all cold stores 1 and 3 by fitting better door protection and pedestrian doors, installing liquid pressure amplification pumps and suction liquid heat exchangers and by optimising defrost settings.

Energy consumption

Data collected on energy usage in cold stores is shown in Figure 1, Figure 2 and Figure 3. Energy data was collected from a cold storage survey carried out by FRPERC and from published information from references 1, 2, 3 and 4. It is clear from the data that a large range in efficiencies exists. The data is shown plotted as kWh/m³/yr in Figure 4. In all cases the worst cold store consumed over 8 times the most efficient store.

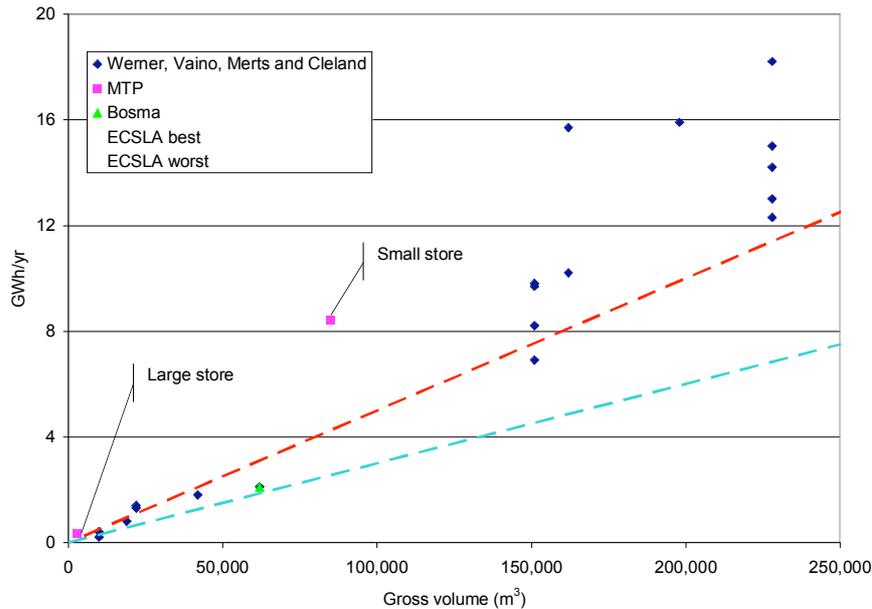


Figure 1. Energy consumption in chilled cold stores (Data labelled ECSLA from reference 4).

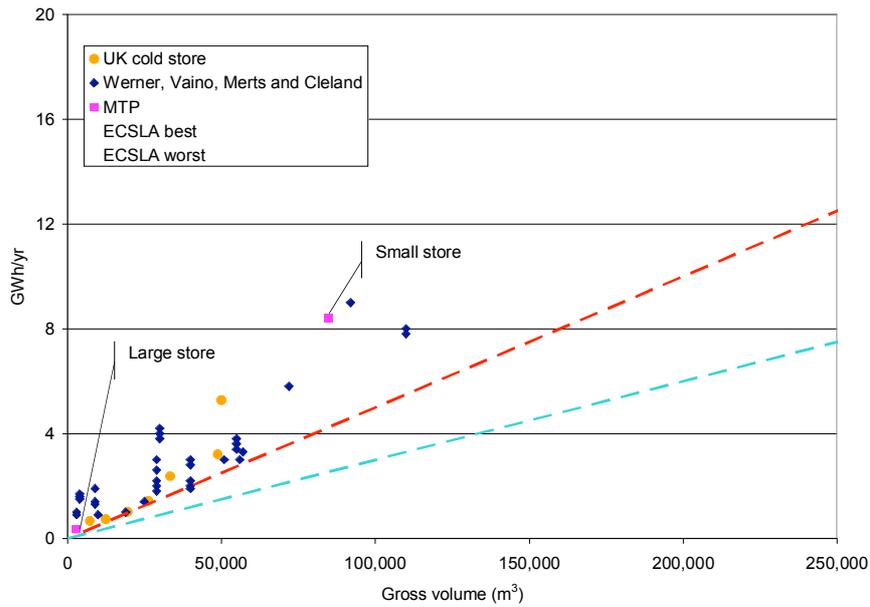


Figure 2, Energy consumption in frozen cold stores (Data labelled as ECSLA is from reference 4, UK cold store data is from FRPERC survey).

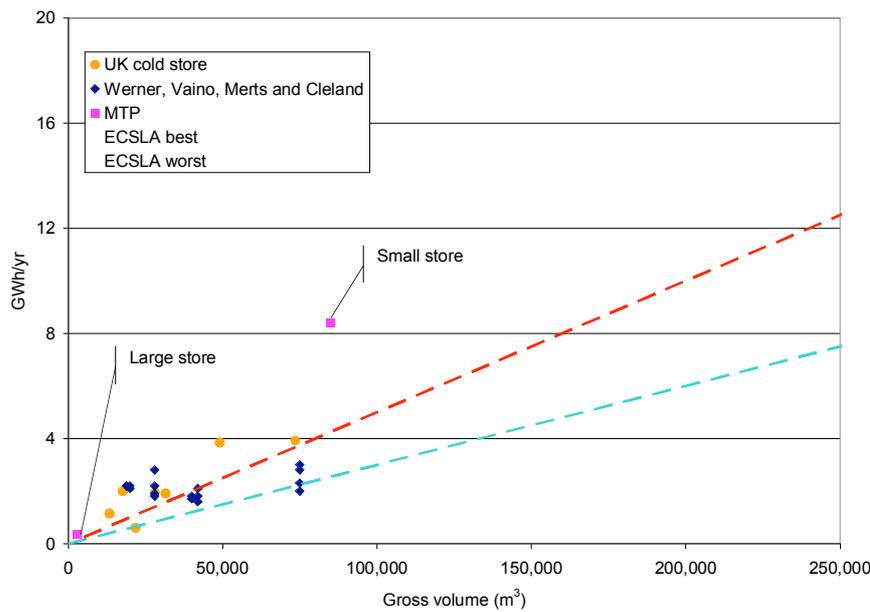


Figure 3, Energy consumption in mixed cold stores (Data labelled as ECSLA is from reference 4, UK cold store data is from FRPERC survey).

Energy saving options in R22 DX system

Three cold storage warehouse facilities in the UK were examined during a 3 week period in late summer 2006 with the aim of identifying procedures to reduce energy consumption. The storage facilities were divided as follows:

Cold store 1: 3 frozen chambers (1550 m²).

Cold store 2: 1 frozen chamber (910 m²).

Cold store 3: 3 chilled chambers, 1 frozen chamber (2458 m²).

Stores 1 and 3 were operated by a direct expansion refrigeration system with single stage reciprocating compressors and evaporative condensers. Store 2 was operated from a low pressure receiver system with a twin screw economised compressor and an air cooled condenser. All stores were operated on R22.

Data collection

Data were obtained from a variety of sources including the on site data loggers (that recorded temperatures in the cold rooms, energy consumed by each cold store and door openings), data loggers that were placed on the evaporators and plant of each cold store, manufacturers performance data and spot measurements in each cold store. Data logged from the refrigeration system were recorded at intervals of between 30 seconds and 2 minutes. Details of movement of food in and out of each room were recorded by the store operators.

Heat load across walls

The heat load on each room through the cold store fabric was calculated using the following equation:

$$Q = UA\Delta t \quad (\text{Equation 1})$$

Where; Q = heat (kW); U = overall heat transfer coefficient (W.m⁻².K); A = area of wall (m²); Δt = temperature difference between outer and inner wall (K)

Cold room ceilings were all 5” polystyrene, all walls were 5” polyurethane and the floor was 5” polyurethane plus 4” granolithic concrete. The temperature difference between the external and internal wall was calculated from internal chamber temperature obtained from logged data and dry bulb temperature obtained from the UK Meteorological Office. The temperature of the air on the outside of the cold store walls and in the roof space was recorded using data loggers placed in the roof space.

Infiltration

In the cold stores examined entry to the room was exclusively through the main cold store doors. Data on cold store usage obtained from sensors placed on the cold store doors was used to calculate the heat load on the room during door openings. All cold store doors were automatically opened for 15 seconds per opening. All storage rooms were fitted with strip curtains and the integrity of the strip curtains were assessed to provide a level of effectiveness. The effectiveness of the air curtains was adjusted to provide an effectiveness that included the time prior to entry of a person or fork lift when the strip curtain was intact, a period when the curtain was broken by physical interference and a period when the strip curtain returned to its original effectiveness. The areas of all cold room doors were measured and were between 6.3 and 9.3 m².

The model developed by Gosney and Olama (1975) [5] has been shown by Foster et al (2003) [6] to provide the most accurate prediction of infiltration through cold room doors. The Gosney and Olama equation (Equation 2) assumes that the air temperature within the cold room remains stable during door openings (this is a reasonable assumption in a large room that is not left open for extended periods).

$$q = 0.221A(hi + lr - hr)\rho_r \left(1 - \frac{\rho_i}{\rho_r}\right)^{0.5} (gH)^{0.5} F_m \quad (\text{Equation 2})$$

Where

q = heat through infiltration (W)

A = Area of cold store door (m²)

hi = Enthalpy of ambient air (kJ.kg⁻¹)

lr = Latent heat of refrigerated air (kJ.kg⁻¹)

hr = Enthalpy of refrigerated air (kJ.kg⁻¹)

ρ_i = Density of ambient air (kg.m⁻³)

ρ_r = Density of refrigerated air (kg.m⁻³) calculated from $\rho = p / R T$ (where p = pressure in Pa (assumed to be 100,000), T = temperature in K and R = universal gas constant (287))

g = Acceleration due to gravity (9.81 m.s⁻²)

H = Height of cold store door (m)

$F_m = (2/(1+(\rho_r/\rho_i)0.333))1.5$

In all calculations the RH in the cold store was assumed to be 90% (at low temperatures the enthalpy of the water content of the cold store air does not vary much and therefore the RH value used was not critical). In one case in store 1 entry to another ambient store was

through one of the frozen chambers and therefore these additional door openings were added to the load on the frozen chamber.

Heat load from food

Although ideally food should not be frozen or chilled in a cold store occasionally food was cooled after entry into the chambers. Data provided by the cold store operator on temperature, type of food and quantity of food entering the cold stores was used to calculate heat load on each store.

Fixed heat loads

The heat loads added to the room from pedestrian access and fork lift trucks were derived from the door opening data and food entry data. It was assumed that food entry and exit to the rooms was always achieved using a fork lift and heat loads were obtained from fork lift manufacturers' data. In all cases large fork lifts remained in the store rooms during daily operation whereas smaller fork lifts were used to transport food in and out of the stores. The difference between door openings for entry of food and the total number of door openings was assumed to be due to pedestrian access to the room. The heat load due to pedestrians was calculated from the following equation (from ASHRAE):

$$Q = 273 - 6t \quad (\text{Equation 3})$$

Where: Q=heat load (W) t=temperature of room (°C)

The fixed heat loads on the rooms from lights, defrost heaters and evaporator fans were measured.

Efficiency of refrigeration plant

Data recorded from the data loggers placed on the refrigeration system was used to calculate heat extracted by the evaporators. Energy consumed by the compressors, evaporator fans and condenser fans was measured and energy required to extract 1 kW of heat from the room calculated. The actual energy consumed by each store was compared to the calculated heat loads divided by the COSP (Coefficient Of System Performance) that was calculated for each room under stable operating conditions and included the energy consumption of compressors, evaporator and condenser fans. The efficiency of each cold store was compared and options to improve efficiency identified and the savings in energy calculated.

Heat loads

The heat loads from transmission, infiltration, fixed loads, food and machinery/people are shown in Figure 5. In all cases heat loads through transmission were the greatest heat loads on the room. This was partly because the thickness of insulation was less than would be usual in modern cold stores. Although difficult to rectify, increasing the level of insulation would have a significant effect of heat loads (Figure 6). By increasing the insulation to 8" (203 mm) heat loads would be reduced by 35%, if increased to 10" (245 mm) heat loads would be reduced by 47% and if increased to 13" (330 mm) the reduction would be 64%.

The second highest heat load was from fixed loads. Little could be done to reduce heat loads from fans in the rooms apart from pulsing the fans or by increasing the time when the cold store was switched off. Options to reduce the defrost energy were identified in the chilled stores where evaporators could be switched from electric to 'off cycle' defrosting. This was estimated to save approximately 40,000 kWh per year in store 3 (the only facility with chilled storage). Energy reductions in the frozen stores could be achieved by optimising the defrost settings and ensuring complete defrosting of the evaporators.

The other major heat load on the rooms was from infiltration. A major source of heat in the rooms that were heavily used was from pedestrians opening the doors (rather than door openings for entry or exit of food). If pedestrians used smaller doors for entry into the cold stores (assumed to be 2.5 m²) the energy savings were considerable. Figure 7 shows the effect of usage of pedestrian doors combined with the current main doors with strip curtain for entry of food. If the door protection on the main doors used for food entry were further improved (by usage of rapid roll doors or a dehumidified air lock) additional energy savings could be made (Figure 7). Depending on the usage of each room, savings of between 47 and 59% could be made by fitting pedestrian doors and improving door protection.

The heat load from food placed in the cold rooms was not especially high in any of the rooms except in one room in cold store 1 where food was regularly accepted for freezing. In most rooms the temperature of the food on entry to the room was similar to the room temperature. However, in several instances food entered the room at below the room temperature and therefore the food acted as a cooling load on the room (as can be seen in Figure 5 for plant 3).

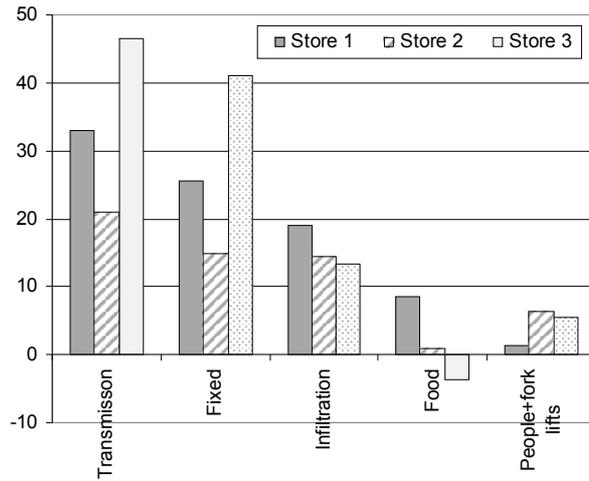


Figure 5. Heat loads (kW) on each cold store.

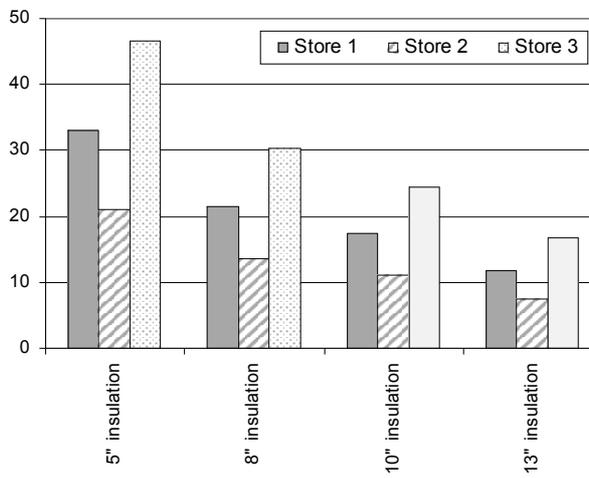


Figure 6. Reduction in heat loads (kW) by increasing insulation thickness.

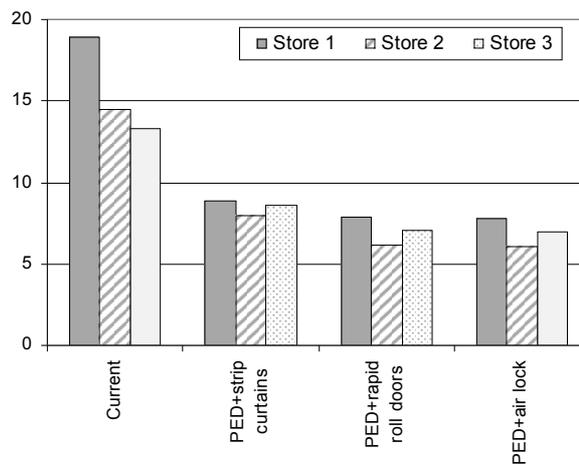


Figure 7. Reduction in heat loads (kW) by altering door protection and usage.

Efficiency of refrigeration plant

The amount of energy that must be used to remove 1kW of heat from the evaporators in each of the refrigeration plants is shown in Table 1.

Refrigeration plant	Energy used to remove 1kW of heat from the evaporator (kW)
Cold store 1	0.66-1.07
Cold store 2	0.79
Cold store 3	1.09 (frozen), 1.02-1.10 (chilled)

Table 1. The amounts of energy that must be used to remove 1kW of heat from the evaporators in each of the refrigeration plants.

One store room in cold store 1 was the most efficient plant. This was because it operated at a warmer temperature (-14°C) than the other cold stores (that were set for -22°C) and the refrigeration plant had been optimised for the warmer conditions. Cold store 2 was the next most efficient refrigeration plant and operated to maintain a set temperature of -22°C in the chamber. The chamber was cooled by a low-pressure receiver system which was more efficient than a conventional direct expansion of refrigerant as the evaporator was fully wetted with liquid refrigerant and therefore used to its maximum effectiveness.

The efficiencies of the other plants were broadly similar to each other, the differences reflecting small differences in condensing and evaporating temperature. Store 3 was designed to operate chambers at -18°C and was being used to operate 2 chambers at 1°C . Though the plant had been modified, it had not been optimised to run with such warm temperatures and, although the loads on the room were much smaller than on frozen chambers, the refrigeration plant ran very inefficiently, no advantage being taken of the warmer evaporator temperatures possible.

The refrigeration plants set for -22°C were not designed for such cold operating temperatures and the effect was that in summer, when the survey was carried out, the plant was too small to extract the amount of heat needed to maintain the chambers at the set temperature. Therefore the chambers operated at -20°C and would only be able to achieve the set point temperature in the winter months. As the set temperatures were not being achieved and products were still being adequately stored it was though possible that the set point temperatures could be raised to reduce energy consumed in the winter months.

Apart from in cold store 2 where the evaporator ran fully flooded significant amounts of the surface areas of the evaporators on all chambers were used to superheat the refrigerant before leaving the evaporator. Suction liquid heat exchangers could be used to eliminate the need

for evaporator surface for superheating and this extra surface used, either to reduce the temperature difference between the chamber air and the evaporating temperature, so increasing the evaporator temperature, or to increase the duty of the evaporator if the chamber is not maintaining its set temperature.

Reductions in the energy used by the plants were also identified if condensing temperatures could be reduced. In the survey the condensing temperature was controlled to a high pressure in cold stores 1 and 3 to ensure that there was always adequate refrigerant flowing into the evaporators. In cold store 3 evaporators furthest from the plant room were found to have insufficient pressure of refrigerant to prevent local boiling in the liquid. By fitting liquid line pumps to increase the liquid pressure from the receiver to the expansion valves the condensing pressure can be allowed to decrease as the ambient wet and dry bulb temperatures decreases.

Actual performance of the cold stores

The actual performances of each cold store per cubic and square metre are shown in Table 2. Predicted savings in energy assuming that door protection was improved, pedestrian doors fitted, liquid pressure amplification pumps fitted, defrosts optimised, suction liquid heat exchangers fitted and other minor issues corrected would result in reductions in energy of 23% in cold store 1, 5% in cold store 2 and 39% in cold store 3. If cold store 2 were fitted with an evaporative condenser the savings would increase to 38%.

Figures published on cold store energy consumption by Werner et al [1], Famarazi, Coburn and Saradian [3] and the Market Transformation Programme [2] report energy consumption of between 370-560 kWh per square meter per year and 8-120 kWh per cubic metre per year for cold stores of similar sizes.

Refrigeration plant	kWh/yr	kWh/yr/m ³	kWh/yr/m ²
Cold store 1	710,335	57.3	458.3
Cold store 2	652,573	71.1	710.6
Cold store 3	1,138,178	57.9	463.1

Table 2. Energy consumed by each cold store.

Conclusions

Substantial savings could be achieved if operation of cold storage facilities were optimised in terms of heat loads on the rooms and the operation of the refrigeration system. Many improvements were low in cost (improved door protection, defrost optimisation and repairs)

and most other improvements were cost effective and had short pay back times. Cold store 2 was the most efficient store in terms of heat removed by the refrigeration system and this was due to it being operated by a low-pressure receiver system. However, the plant used more energy per m³ and this was partly due to it being a small store and also to the way the store was operated with high transmission, infiltration and fixed heat loads. Improvements to its operation could be made by fitting better door protection and reducing fixed loads. Substantial improvements could be made to all cold stores 1 and 3 by fitting better door protection and pedestrian doors, installing liquid pressure amplification pumps and suction liquid heat exchangers and by optimising defrost settings.

Efficiency of large ammonia plant

Plant configuration.

The original installation comprised fluidised bed, blast and plate freezers and a cold store with 3 chambers. The temperature at which the cold store was designed to operate is uncertain. The process plants and cold store were served from a complex interlinked ammonia system. There were 3 surge drums, 5 low stage screw compressors of varying capacity, each with an unloading slide, 2 open flash intercoolers, 4 high stage screw compressors, each with unloading and variable volume ratio slides, 2 evaporative condensers and 2 receivers. The number of compressors that could be linked to each surge drum or intercooler could be varied to suit the evaporating or intercooler temperature.

At some time 2 more cold stores were built on the site and a new plant built to serve them. This plant comprised a single surge drum, a low stage screw compressor, a closed flash intercooler, a single high stage compressor and an evaporative condenser. Both compressors had unloading and variable volume ratio slides. A second compressor, equal in capacity to the low stage compressor, could be used to standby for either the low or high stage machine. At some time after it was installed the plant was linked to the header system of the original plant.

The history of the modifications to the plant is not known. No one currently working at the site knew the detailed history and there were few records. It is believed that the changes were made in stages to meet short-term objectives.

Current configuration of the plant

Currently the site has 3 cold stores, one with a single chamber, a second cold store divided into 3 interconnecting chambers and a third, the original cold store, divided into 3 separate chambers, each operating at a different temperature. There are also 2 blast freezers (not original).

These units are served by 2 ammonia systems using components of the original plants. The first 2 cold stores, both operating nominally at -18°C , were served by an ammonia system with a single surge drum, single low and high stage compressors both with slider unloading, a closed flash intercooler, evaporative condenser and receiver (Plant 1). There are no open connections between this and the second ammonia plant. The other cold store and the 2 blast freezers are operated off a second system. There is a single surge drum, single low and high

stage compressors both with unloading slides, an open flash intercooler, an evaporative condenser and receiver (Plant 2). Defrost on both plants was by hot gas (Figure 8).

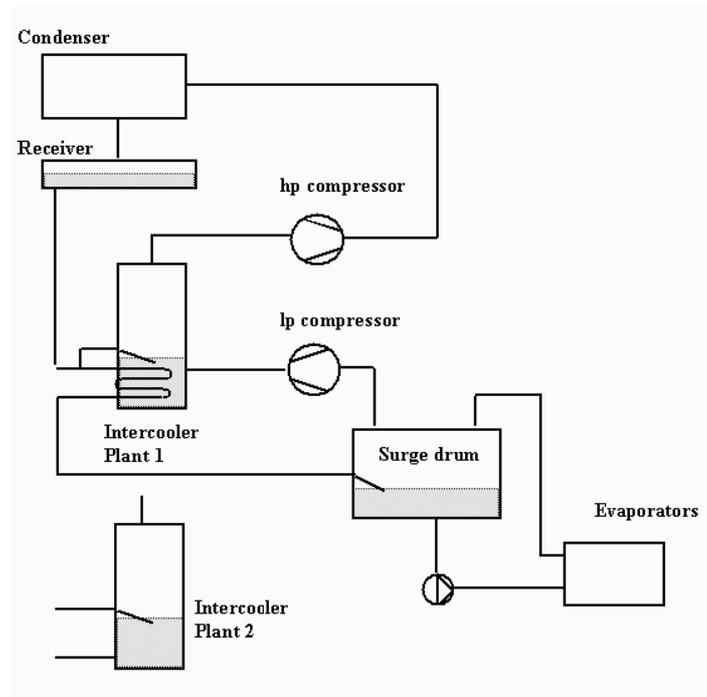


Figure 8. Schematic layout of the 2 refrigeration plants.

Measurements

A survey of the site was carried out during one week in winter. No intrusive measurements were made so the mass flow rate of refrigerant was not measured and the only pressure readings taken were from the gauges fitted to the plant. Data was collected from:

1. Site diagrams and blue prints where these existed.
2. Temperatures of selected positions on the refrigeration systems recorded on data loggers placed there temporarily for the survey.
3. Spot readings of temperature and pressure at selected positions on the refrigeration system.
4. Manufacturers' data.
5. The staff at the site.
6. The meteorological data were obtained from the nearest weather recording station to the site.

Between 3 and 8 temperature sensors (single point 'T' type thermocouples) were attached to each data logger. Temperature sensors were placed on the outside of the refrigerant pipework

or vessel under the insulation at the following locations on each refrigeration plant and recorded at 1 min intervals. The positions were:

1. First turn of evaporator coil, wet return, hot gas line and return air from the cold store at selected evaporators.
2. Suction and discharge of low and high stage compressors.
3. Entries and exits of refrigerant at intercoolers.
4. Surge drum.

Manual readings were taken to back up the above readings and of the entry and exit temperatures of the oil to the compressors. Pressures readings were taken from the gauges fitted to the plant. During the week a calibrated pressure gauge was substituted for each plant gauge to check its accuracy at that pressure. The electricity energy consumption of the plants was taken from daily meter readings and spot measurements were made of the power consumption of the compressors, fans and pumps.

Results

The aim was to make estimates of the COSP (Coefficient of System Performance = the total amount of energy used to remove one kW of heat from the load) of the refrigeration systems. The data collected was examined for consistency. The refrigerant pressures were compared to the temperature readings in the intercoolers, the surge drums and the evaporators. Before checking the calibration of the pressure gauges there was very poor agreement between the suction pressures of the low or high stage compressors and the surge drums and intercoolers. The calibration of the gauges was therefore checked against a calibrated gauge during periods of steady running by substituting the plant gauge with the calibrated gauge. This showed that the pressure gauges at some locations were inaccurate. This included the pressure transducers for the control of pressure in the surge drum and intercooler of one of the plants.

Data needed to calculate the COSP included the isentropic efficiency of compression. The efficiency of screw compressors is good when operating at full load but decreases significantly as the load on the compressor is reduced. It was not possible to measure the degree of unloading of the compressors during the survey and therefore it had to be estimated. As the isentropic efficiency of screw compressors decreases the extra energy used in compression increases the temperature of the compressed gas and the oil leaving the

compressor. The discharge and oil leaving temperatures can therefore be used as a guide to the compression efficiency.

Compressor efficiency

Figure 9 shows the discharge temperature from the low and high stage compressors and the saturated temperature of the surge drum and intercooler for Plant 1. If the compressors had been in good condition and operating at their maximum capacity the manufactures data showed that the discharge temperatures would have been 47°C for the low pressure (lp) stage and 56°C for the high pressure (hp) stage under the prevailing conditions and the corresponding isentropic efficiencies 68% and 77% respectively.

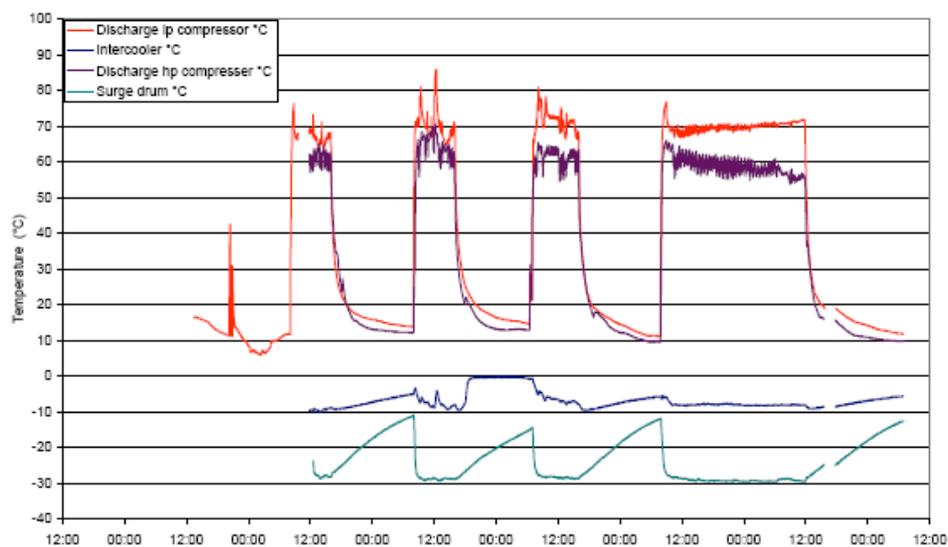


Figure 9. Surge drum and intercooler saturated temperature and lp and hp compressor discharge temperature for Plant 1. Evaporating pressure 1.26bara, condensing pressure 8.6bara.

The plant was run intermittently. At each start the lp compressor reduced the pressure of the surge drum quickly, thereafter operating at a reduced capacity to control the drum pressure to its controlled value (-29°C). The discharge temperature was 70°C; according to the manufactures data this was warmer than would be expected even if operating at minimum capacity if the compressor was in good condition. This indicated that the compressor was worn and or that the oil flow rate was too small irrespective of its loading.

During the eight-hour period that the plant was run each day the pressure in the intercooler was continuously reduced from between -4 and 0°C at start-up to -9°C when the plant was switched off. This indicated that the hp stage was running at 100% loading over this period. The discharge temperature was 64°C, 8°C warmer than if the compressor had been in good

condition. For the discharge temperature to increase by this amount at full load and with the design oil flow rate the isentropic efficiency would have had to decrease from 77 to 60%.

At steady operation the mass flow through the lp stage must be equal to the hp stage less the amount of flash gas in the intercooler. This indicated that the lp stage was operating at 60% capacity. At this capacity the discharge temperature of a compressor in good condition and with the design oil flow rate would have been 47°C. For the discharge temperature to have increased to 70°C the isentropic efficiency would have had to reduce from 60% to 29%. This would have consumed more power than was actually measured and therefore the oil flow rate may also have been reduced. It was estimated that the compressor efficiency was 46% a figure that agreed closely with the measured power.

These estimated efficiencies agreed well with the measured power consumption for the compressors and were therefore used to calculate the COSP for the system. A similar exercise was carried out for Plant 2. In addition to the power for the compressor motors the COSP included the auxiliary loads on the system for the condenser fans and pumps, evaporator fans (in cold stores), refrigerant pumps, drain heaters and oil cooler fans and pumps. These were all taken from direct measurements. The evaporator fans in the blast freezers, served by Plant 2, were not included in the COSP as they consumed a lot more power than the cold store fans and the figures for the 2 plants would not have been comparable.

	As measured during survey	At measured conditions with compressors in good condition	Compressors in good condition and with lp and hp stages working at 100% capacity
Plant 1	0.66	0.55	0.53
Plant 2	1.03	0.71	0.64

Table 3. COSP for the 2 systems as measured, if the compressors had been in good condition and if the compressors had also been fully loaded. The figures for Plant 2 exclude the evaporator fans in the blast freezers but including the evaporator fans in the cold stores.

COSPs for the 2 plants during steady running are shown in Table 3. Also shown is the COSP if the compressor had been in good condition but working at part load and if in addition the size of the low and high stage compressors had been matched and running at full load. Conventionally, if there is no load on the system the intermediate pressure is chosen for maximum efficiency and the capacity of the low stage compressor varied to maintain it. Figure 10 shows the effect of altering the isentropic efficiency of the lp and hp compressors between 0.65 and 0.8 and the resulting effect on COP (Coefficient of Performance). It can be seen from Figure 10 that when the isentropic efficiency of the hp and lp compressors are the

same, the COP is not a strong function of the intercooler pressure (l_p 0.8 h_p 0.8). When operating at an evaporating pressure of 1.26 bara and a condensing pressure of 8.6 bara the optimum intercooler temperature was -6°C saturated temperature. However, for this theoretical case the intercooler pressure can vary over a wide range with little loss of efficiency.

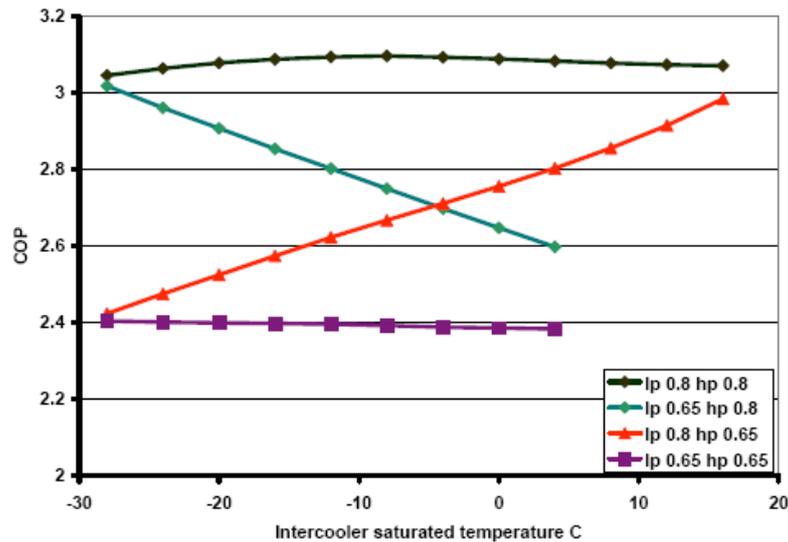


Figure 10. The COP of the system when operating at an evaporating pressure of 1.26 bara and a condensing pressure of 8.6 bara and the isentropic efficiency of both l_p and h_p compressors is either 0.8 or 0.65.

However, the COP is strongly dependent on the isentropic efficiency of compression. If the isentropic efficiency is the same for both l_p and h_p compressors the optimum intercooler pressure hardly changes but the COP will decrease with decreasing efficiency. Figure 10 shows the effect on the COP of changing the isentropic efficiency from 0.65 to 0.8 for both compressors.

If one compressor is more efficient than the other then the optimum intercooler pressure will be the same as that for a single stage system using only the efficient compressor. In practice this may not be possible as a single stage system operating at these pressures may have an excessive discharge temperature that would lead to a rapid deterioration of oil and compressor.

The above shows that if compressors have dissimilar efficiencies the optimum intercooler pressure is that which maximises the use of the efficient compressor without causing excessive discharge temperatures.

Defrost operation

Defrosting of the evaporators had originally been automatic. However, over time faults had developed in the controls and the system was defrosted manually. As a result the defrosts were not optimal and frost built up on some of the evaporators. A second fault with the defrost was that some of the hot gas valves at valve stations had failed and continually passed hot gas into the evaporators. This was effectively an increased heat load on the evaporators and is not shown in the COSP figures.

Insulation

The insulation of the liquid feed and wet return lines and the surge drums had deteriorated over time. The insulation was icebound with a corresponding increase in heat ingress. This effect was also treated as an increase in the load and was not included in the COSP figures. The heat gain in the lines of Plant 1 was calculated to be 23 to 59 kW and for Plant 2 between 30 and 70 kW (depending on the wind speed, wet and dry bulb temperature and solar radiation).

Control system

It was found that the surge drum and intermediate pressure controls were inaccurate and that the plant operators were unaware of this. On this site the energy consumption of the plant was reduced as a result of this error as the pressure transducers for the surge drums were indicating a lower pressure than the actual pressure. The operators therefore thought that the system was operating at -38°C whereas it was actually operating at -29°C , a temperature that was adequate for the loads prevailing at the time. However the reverse could easily have been the case and the plant could have been operating at colder evaporation pressures than thought. This illustrates the need for accurate instruments on plant.

Over time the flow control valves at the valve stations had also been altered and the recirculation rate in the evaporators had altered. This effect was not measured during the survey but on two evaporators it had been reduced to less than 1 and superheated vapour was leaving the evaporator.

Discussion

The plants as originally designed would have had COSPs of 0.53 and 0.64 when operating at the measured conditions, the difference reflecting different evaporating temperatures, Plant 2 operating a blast freezer. The systems as currently configured were operating with COSPs of

0.66 and 1.03, which represent a 13 and 61% increase in energy consumption. The reason for this inefficiency was that the compressors were worn and in need of maintenance and that there was a mismatch between the sizes of the low and high stage compressors.

The figures show that lack of maintenance is a false economy. The estimates made on this plant show that the extra cost due to wear in the compressors alone was £59,000 per annum. In addition to this, the cost of operating one of the compressors at part load, and hence lower efficiency, was £12,300 per annum (the cost of energy was £0.075 per kWh).

The increased heat load through the ice bound insulation on the liquid and wet return pipes increased the cost of running the plants by £44,000 per annum. It was not possible to estimate the cost of the extra heat load due to leaking hot gas valves but this was expected to be more than the annual cost of maintenance.

Conclusion

The survey has illustrated how a plant that was originally designed for efficient operation has altered over the years and now consumes approximately 43% more energy than was originally intended.

It is expected that a two-stage ammonia plant with flash intercooler, pumped recirculation and evaporative condensers would be very efficient. However, poor part load isentropic efficiency of the compressors, a lack of balance between high and low stage compressor capacity and the need for maintenance has turned what should have been an efficient plant into an inefficient one.

To keep things in perspective, the estimated COSP of this plant was still better than an R22 direct expansion system with electric defrost serving a similar cold store complex.

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