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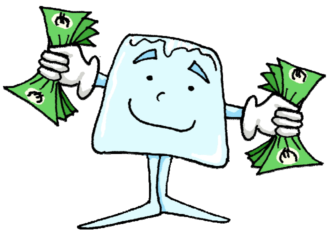

Guide 5
Site Guidance Topics

Reducing heat loads

Reducing head pressure

Improving part load performance

Reducing power consumed by pumps and fans

Other Project Sponsors

Improving Refrigeration System Efficiency

Guidance Based on Visits to 30 Food and Drink Factories

Chapters

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The Food & Drink Industry Refrigeration Efficiency Initiative

is a

Carbon Trust Networks Project

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Improving Refrigeration System Efficiency
Guidance Based on Visits to 30 Food and Drink Factories

This guide will help you improve the performance of refrigeration equipment. It is based on investigations carried out at 30 sites and covers 4 specific topics:

1. Make savings by reducing heat loads and by carrying out cooling at the appropriate temperature.
2. Making savings by improving the condenser system and, in particular, by reducing head pressure.
3. Making savings by improving part load performance.
4. Making savings by reducing power consumed by auxiliary pumps and fans.

1. Introduction

This Guide contains practical advice about how to improve the efficiency of industrial refrigeration systems used in the food and drink industry. It was produced under the Food & Drink Industry Refrigeration Efficiency Initiative, a project sponsored by the Carbon Trust and supported by the Food and Drink Federation, the British Beer and Pub Association, the Cold Storage and Distribution Federation, Dairy UK and the Institute of Refrigeration.

In the Refrigeration Efficiency Initiative, eight topics were investigated to provide guidance to food and drink manufacturing companies to help improve energy efficiency. Four of these topics are reported in this Guide. 30 manufacturing sites acted as “Host Sites”. Refrigeration specialists visited each of the sites and carried out an energy efficiency assessment related to the four topics selected. In this document we have brought together the “common themes” from the site investigations and explained numerous energy saving opportunities.

The structure of the Guide is as follows:

- **Section 2** gives general background to refrigeration efficiency and outlines the importance of the 4 topics selected for investigation.
- **Section 3** deals with opportunities related to heat loads, including heat load reduction and use of appropriate cooling temperature levels.
- **Section 4** explains how to keep the condensing temperature of a plant as low as possible, especially during periods of low ambient temperature.
- **Section 5** addresses the importance of part load operation and how to design and operate a plant efficiently at part load.
- **Section 6** discusses the use of auxiliary pumps and fans and how to avoid these becoming an excessive electricity consumer.

Four further “Generic Guides” have been produced during the Refrigeration Efficiency Initiative. These cover the energy efficiency issues related to: (a) Maintenance Contracts, (b) Purchase of New Plant, (c) Plant Operational Performance and (d) Refrigerant Regulations including phase out of R22 and the F-Gas Regulation. Details about how to obtain copies of these Generic Guides are given in Appendix 2.

2. Background to Topics Covered in This Guide

Refrigeration plant is one of the largest consumers of electricity at many food and drink manufacturing plants. Energy is used by the refrigeration compressors and also by associated auxiliaries such as pumps and fans. The importance of refrigeration in different parts of the industry is typically as shown in Table 1.

Table 1 Importance of Refrigeration Related Electricity Use

Industry Sector	% of electricity used for refrigeration
Liquid milk processing	25%
Breweries	35%
Confectionery	40%
Chilled ready meals	50%
Frozen food	60%
Cold storage	85%

Although refrigeration is of crucial importance to many types of food company, it is poorly understood and many plants are both designed and operated very inefficiently. Refrigeration systems are relatively complex and it is hard to know whether your plant is running efficiently. One of the problems faced by many owners and operators of plant is that they seem to get confusing and often conflicting advice from manufacturers and contractors.

This Guide, which addresses four vital areas of opportunity, together with four further Guides produced during the Refrigeration Efficiency Initiative, should help you understand some of the key issues and set up a good efficiency programme at your factory. There is no shortage of energy efficiency opportunities – the key is to spot the ones that are most applicable to your particular situation.

We have selected four topics as a basis for the site investigation work. These four topics were considered by the project's refrigeration experts to be those that address the most common causes of energy wastage in refrigeration system design and operation.

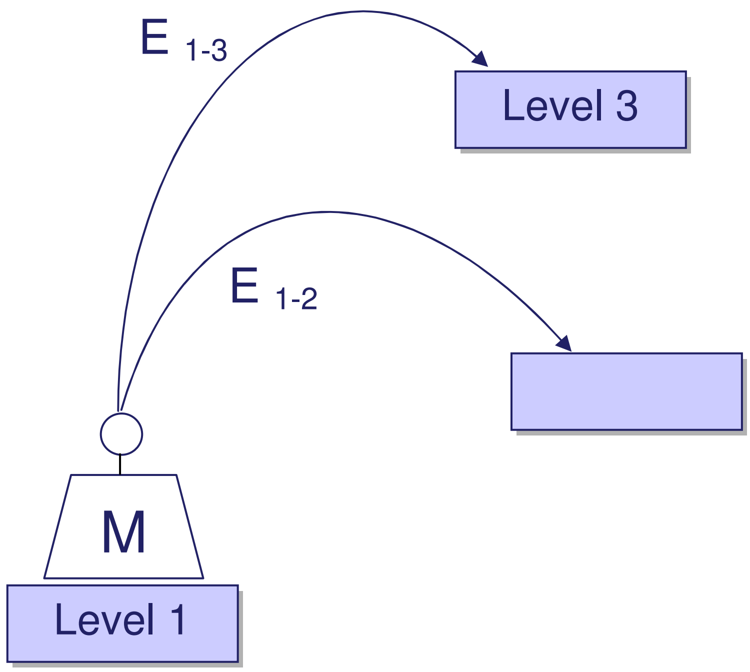
2.1 The Heat Pulley Analogy

Before explaining the four topics selected, it is useful to introduce an analogy that helps explain certain aspects of refrigeration energy efficiency.

A refrigeration plant can be thought of as a "heat pulley". In an ordinary pulley system, the objective is usually to lift a weight from a low level to a higher level. See figure 1. There are three fairly obvious ways in which the energy consumed by a pulley system is influenced:

1. **The size of the weight.** It takes more energy to lift a large weight than a small one.
2. **The height through which the weight is raised.** It takes more energy to lift the weight over a large difference in height than over a small one.
3. **The mechanical design and the operability of the pulley system.** The energy consumption could be affected by various design aspects e.g. the amount of friction on the pulley bearings.

Figure 1: The Heat Pulley Analogy



A refrigeration system has many similarities to a pulley system. The objective is to “capture” an amount of heat energy at a low temperature and to “lift” the heat to a higher temperature so it can be rejected into the environment. For example, you put some warm soup into a domestic refrigerator that is at 4°C. The heat energy in the soup is “loaded” on the Heat Pulley system at a temperature a bit below 4°C and then raised to a temperature where the heat can be rejected into the kitchen – which must be a temperature a bit above the temperature in the kitchen (e.g. >20°C).

In terms of energy efficiency, the same rules apply as described for the mechanical pulley system. The energy consumption depends on:

1. **The size of the heat load.** If you are “moving” more heat than necessary you will be wasting energy. For the domestic refrigerator described above it would be better to allow the soup to cool to, say, 30°C before putting it in the fridge than to put it in at 80 to 100°C straight from the saucepan. In this simple example you can save up to 75% of the heat load by “free cooling”. ***The “1st Heat Pulley Rule” is to minimise the heat load.*** As we see from the example, it is possible that your heat loads are substantially higher than necessary.

2. **The “temperature lift” of the plant.** The temperature difference between the “cold end” of a refrigeration plant (where heat is “loaded on to the pulley” and the “hot end” (where heat is “unloaded”, usually by rejecting the heat into the ambient) is equivalent to the height difference of a mechanical pulley system. ***The “2nd Heat Pulley Rule” is to minimise the temperature lift of the plant.*** This is really important as a refrigeration plant’s efficiency is very sensitive to temperature lift. A 1 deg C reduction in temperature lift can save 3 to 4%. It is very easy to have >10 deg C of excessive temperature lift, either through poor design, poor control or poor maintenance. Clearly this needs to be avoided.

   A common fault on many industrial refrigeration plants is that loads are grouped together on one system “for convenience”. If these loads are at different temperature levels then the “cold end” of the refrigeration plant will need to be cold enough to serve the coldest load. But this means the temperature lift for the other loads is unnecessarily high. For example a plant cooling both Product A to 5°C and Product B to -5°C will have a "lowest common denominator of -5°C and the temperature lift for Product B will be 10 degC higher than it would be if Product A was cooled on a different plant. This could waste 15% to 20% of power input. This situation leads to the ***"3rd Heat Pulley Rule" which is to ensure that each load is on a system at the highest possible temperature.***

3. **The mechanical design of the plant.** There are many ways in which an industrial refrigeration plant can be designed and built. Numerous details about the plant design (e.g. heat exchanger type and size, compressor type and controllability) can have a significant impact on energy efficiency. ***The "4th Heat Pulley Rule" is to ensure that the plant is designed properly and also that it is operated and maintained in such a way that the best level of efficiency can be achieved.***

2.2 The Four Topics in this Guide

As discussed in the introduction, there are numerous ways in which refrigeration plant efficiency can be improved. Indeed, one could list hundreds of individual opportunities. That is good – as it implies that in most situations there is lots of potential to save. It is typical to be able to cost-effectively save 15% to 20% on existing plants and in excess of 25% on new installations and major upgrades.

Unfortunately, to have a long and complicated list of opportunities is extremely unhelpful for non-experts as it is hard to know where to start looking and hard to judge which are the best opportunities. The four topics dealt with in Sections 3 to 6 of this Guide have been selected to help target your efforts towards the most beneficial areas.

Firstly, in Section 3 we deal with opportunities related to heat loads and cooling temperature levels. These address the 1st and 3rd Heat Pulley rules and provide various examples of ways that heat loads can be minimised and ways that cooling can be carried out at the most appropriate temperature level to minimise temperature lift.

Then in Section 4 we concentrate on the "hot end" of a refrigeration plant – the condenser system that rejects heat, usually into the atmosphere. Bad condenser system design, control or maintenance can break the 2nd Heat Pulley rule, leading to high condensing temperature and excessive temperature lift.

In Section 5 we address part load operation. Many refrigeration plants spend long hours operating at relatively low levels of part load – but unfortunately the 4th Heat Pulley Rule is often broken as the plant is not designed or operated to be efficient at these low loads.

Finally, in Section 6, we review the importance of auxiliary power used by pumps and fans. In some plants these can be a dominant part of the total energy use and yet they are often neglected from an efficiency perspective. "Cold end" auxiliaries such as evaporator fans or chilled water pumps are paid for twice! Firstly, you use energy to run the device e.g. 5 kW to run a fan. But what happens to that 5 kW? It becomes a heat load and you need to use refrigeration compressor power to remove the heat energy from the cooled space. Minimising auxiliary power input can produce very good energy savings.

3. Reducing Heat Loads and Cooling at the Appropriate Temperature

3.1 Background to the Energy Saving Opportunities

It is vital that any refrigeration energy efficiency initiative starts with a review of the heat loads on the plant. By understanding the nature of your loads better you can:

a) Reduce the amount of cooling that needs to be done – which will save energy and, if you are building a new plant, will also save capital cost.

b) Ensure that you are carrying out cooling at the appropriate temperature level – again saving energy and often saving capital cost as well.

c) Optimise the plant design and operation to deal with the whole "envelop" of operating conditions. Many refrigeration plants are designed to work well at the "design point" which is a single operating condition defined by maximum cooling load and maximum ambient temperature. In practice most plants spend very little time at their design point – most of the year the ambient temperature is well below the summer peak and the cooling load is often well below the design maximum. Only by reviewing the variations in cooling loads through the year can the plant be properly designed.

In this section of the report we deal with items (a) and (b) above. Item (c) relates to part load operation – this is discussed in Section 5.

Heat Load Reduction

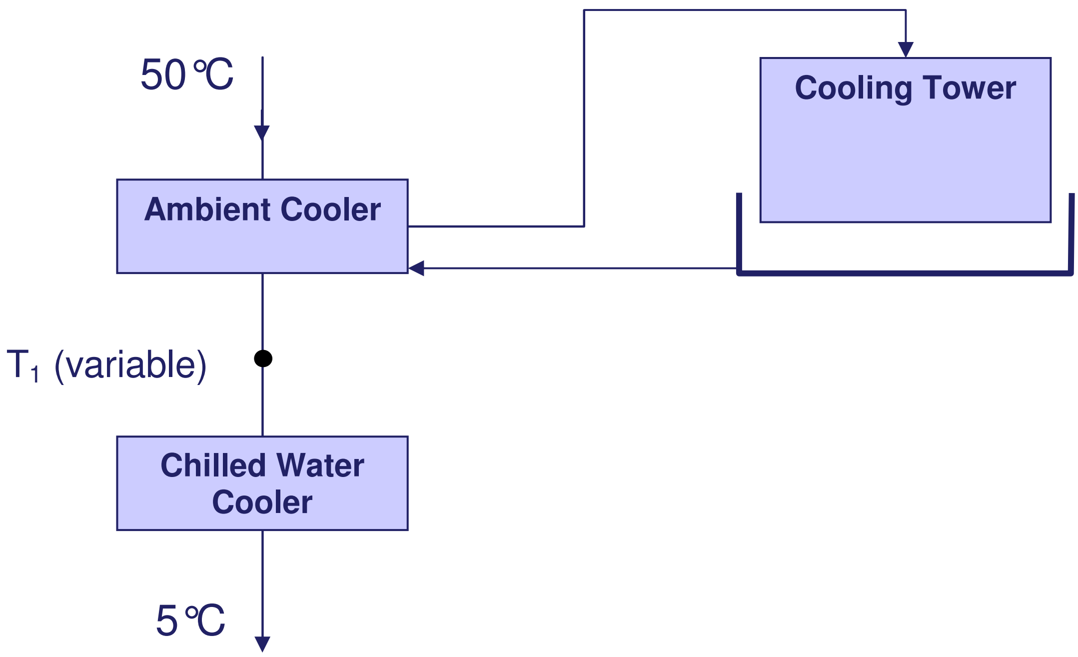
The "1st Heat Pulley Rule" is to minimise the heat load. During the 30 site audits carried out for this project we did not find any without opportunities to reduce their heat loads. We found that the heat load reduction opportunities fell into 4 general categories:

- **Category 1: Process Free Cooling.** In some processes there is a simultaneous need to heat something up and cool something else down. If you can use a heat exchanger to link these 2 streams and pass heat from the hot stream to the cold one you make a double saving – you save both heating energy and cooling energy. Case Study 1 in Section 3.3 gives an example of process free cooling related to pasteuriser operation.

- **Category 2: Ambient Free Cooling.** A more common opportunity is to be able to pre-cool a product <u>before</u> using a refrigeration system. For example a product that has been cooked at 100℃ should not go straight a chiller or blast freezer. It should be cooled as much as possible using an "ambient stream". This can either be air ducted from outside the factory or water that has been cooled in a cooling tower.

  The impact of ambient free cooling can be very dramatic. In figure 2 we see a hot cooked product that needs to be cooled from 65 to 5℃. Without free cooling the refrigeration plant needs to cool the product by 60 degC. If the product is passed through a heat exchanger cooled by water from a cooling tower it can be cooled to within, say, 5 degC of the water temperature. The water temperature coming off a cooling tower will vary from summer to winter, being about 25℃ on a hot summer day and 5℃ in cold winter conditions. Hence the temperature T1 on Figure 2 would vary between about 30℃ in summer and 10℃ in winter. This gives a heat load reduction of over 40% in summer and over 90% in winter!

Figure 2 Ambient Free Cooling



- **Category 3: Reduction / Removal of Unnecessary Process Heat Loads**. There are some situations where a process stream is being cooled unnecessarily. This sometimes happens when a single cooled stream (e.g. of process water) is used for 2 different purposes. Case Study 2 in Section 3.3 gives an example of this in a brewery.
- **Category 4: Reduction of Non-Process Heat Loads.** Many heat loads on a plant are not "core process" loads – they are often referred to as parasitic loads. Examples include heat generated by pumps, fans and lights in cooled spaces and heat ingress through insulation or through open doors. In many cases you can take simple steps to reduce these parasitic loads.

Appropriate Cooling Temperature Level

The "3rd Heat Pulley Rule" is to ensure that each heat load is on a system at the highest possible temperature. This rule is often broken with significant impact on energy consumption.

A common example is where a refrigeration system has a very cold load, e.g. a cold store held at -25°C, and an adjacent warmer load, e.g. a chill room at +5°C. It is very tempting to put the chill room on the same refrigeration system as the cold store, but the impact on energy efficiency is very large. A dedicated plant cooling a 5°C room could use half the amount of energy required if the load is added inappropriately to a plant serving a -25°C load.

Any situation where there is a large central plant used to serve several cooling loads there is a strong risk that the "lowest common denominator" load is much colder than some of the other loads. Case Study 3 in Section 3.3 gives an example of a glycol system that cools several loads that could be more cost effectively cooled on a dedicated warmer system.

In terms of the heat pulley analogy, if you use an inappropriate cooling temperature level you are "dropping the heat downhill" to a lower temperature level before "loading the heat onto the pulley system" to be lifted up to the condenser. Clearly this is not a very good idea!

A more subtle example of this opportunity is when a single load is cooled through a large temperature range. For example, process water is needed at 4°C. The water source is towns water which is at 20°C in summer. The usual way of cooling this stream is to use 1 evaporator that has refrigerant evaporating at, say, 0°C (it must be a few degrees colder than the target water temperature of 4°C). But, you could split the heat load into 2 steps:

a) Plant 1 could cool the water from 20 to 12°C

b) Plant 2 could cool the water from 12 to 4°C.

The temperature lift of Plant 2 is the same as that of the single step plant described above – so it will have the same efficiency. But the temperature lift of Plant 1 is 8 degC less than that of Plant 2 (because we only cool water to 12°C not to 4°C). The efficiency of Plant 1 could be about 30% better than Plant 2. As this is providing half the cooling, the overall saving of a 2 step plant would be half this i.e. 15%. If this was a small load the extra complexity is probably not justified, but if this is a large process load, then a 15% saving is worth having.

It is worth noting that the example above has an extremely variable heat load. In summer the towns water is cooled from 20 to 4°C. In winter, the towns water arrives at between 5 and 8°C. This means that the winter cooling load is between about 5% and 25% of the summer peak. It is important to recognise this before you start designing the plant if you want to get efficient operation in winter! This is dealt with in Section 5 that covers part load operation.

3.2 How to Identify Heat Load Opportunities

The starting point is to make a comprehensive list of all cooling loads and to identify certain key characteristics of the loads. It helps simplify things if you categorise the loads into process loads and other loads ("parasitic" loads). Table 1 illustrates the type of information that needs to be available, based on a fictitious frozen meals factory.

Table 1: Simplified List of Heat Loads at ABC Foods

Load	Peak Duty (kW)	Temperature range (°C)	Load variability	
			Seasonal	Daily
a) Process Loads				
Process water	100	20 to 4	5 to 4 in winter	8 hours/day
Sauce cooling	200	70 to 5	None	12 hours/day
Blast Freezer	500	50 to -20	None	20 hours/day
b) Finished Product Cold Store Loads (held at -20°C)				
Insulation	50	-20	Ambient related	24 hours/day
Evaporator Fans	50	-20	None	24 hours/day
Lights	20	-20	None	24 hours/day
Defrost	25	-20	None	4 hours/day
Door air infiltration	50	-20	Ambient related	24 hours/day
c) Raw Materials Chill Store Loads (held at 4°C)				
Insulation	20	4	Ambient related	24 hours/day
Evaporator Fans	15	4	None	24 hours/day
Lights	10	4	None	24 hours/day
Defrost	10	4	None	2 hours/day
Door air infiltration	15	4	Ambient related	24 hours/day

You need to examine each row in the table and identify which refrigeration plant each heat load is attached to and then assess the opportunities described above in Section 3.1. For example:

- The sauce cooling load is a prime candidate for ambient free cooling.
- Half the process water is only needed at ambient temperature – it should not be cooled first!

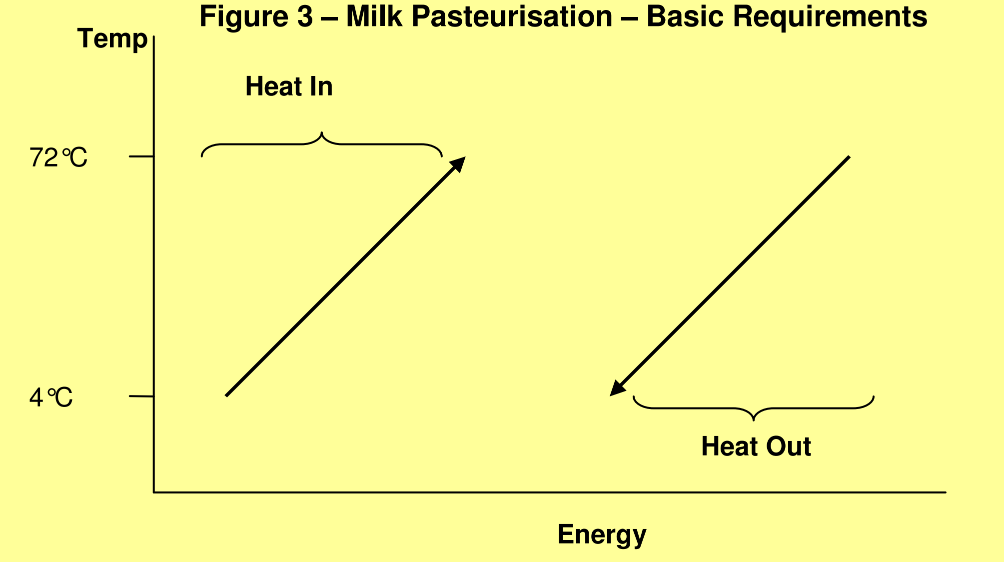
- The blast freezer might be reduced in 2 steps by using ambient air free cooling followed by a blast chilling stage before putting the product into the blast freezer.
- The evaporator fans in both the cold store and chill store run 24 hour/day. Use of variable speed drives and / or an on-off control strategy could remove a significant heat load.
- The door infiltration loads could be reduced by better door closing discipline – this would also have a useful knock-on effect of reducing the need for defrost.
- The defrost loads could be reduced by use of a "defrost-on-demand" control system.

3.3 Heat Load Case Studies

Heat Load Case Study 1: Process free cooling

Pasteurisers are commonly used in the drinks industry e.g. in the processing of milk and beer. The process requirements for milk processing are shown in Figure 3. Incoming milk must be heated from 4°C to 72°C. The milk is held at that temperature for a few seconds (to kill pathogens) and then is cooled back to 4°C.

Figure 3 – Milk Pasteurisation – Basic Requirements



Although it is possible to do the heating and cooling as separate processes this would clearly be very wasteful. All liquid milk pasteurisers use a process of heat recovery between the hot and cold milk streams to minimise both the heat input and cooling load. The layout is shown in Figure 4 and the way this saves energy is illustrated in Figure 5. The hot milk at 72°C is fed through a plate heat exchanger in counterflow with the cold milk at 4°C. The incoming cold milk is heated to temperature T1 and the outgoing milk is cooled to temperature T2.

A key opportunity for companies using pasteurisers is to ensure that the maximum amount of heat recovery is carried out. What are the optimum values for temperatures T1 and T2?

A dairy site had several pasteurisers that used a heat recovery heat exchanger that was considered too small. Temperature T1 was 58°C and T2 was 18°C. This meant that the heating section of the pasteuriser had to increase the milk temperature by 14 degC and the cooling section had to cool the milk by 14°C to reach the required process temperatures. For a plate heat exchanger this is well below the optimum size for the heat recovery section of the pasteuriser. By adding extra plates to the pasteuriser, T1 can be raised to around 65°C and T2 lowered to 11°C. This halves both the heat load and the cooling load. As with many process free cooling opportunities there is a double benefit i.e. a saving in cooling load and a saving in heating load. The payback period for the energy saving investment of fitting extra pasteuriser plates was around 1 year.

This site had a significant extra benefit. The site was short of refrigeration capacity for a new pasteuriser and was considering a £300k investment in a new water chiller. By investing only £30k in modifications to existing pasteurisers they reduced the cooling load by 50%. As well as the energy saving this also avoided the need for the expensive new chiller!

Figure 4 – Regenerative Pasteuriser, Configuration

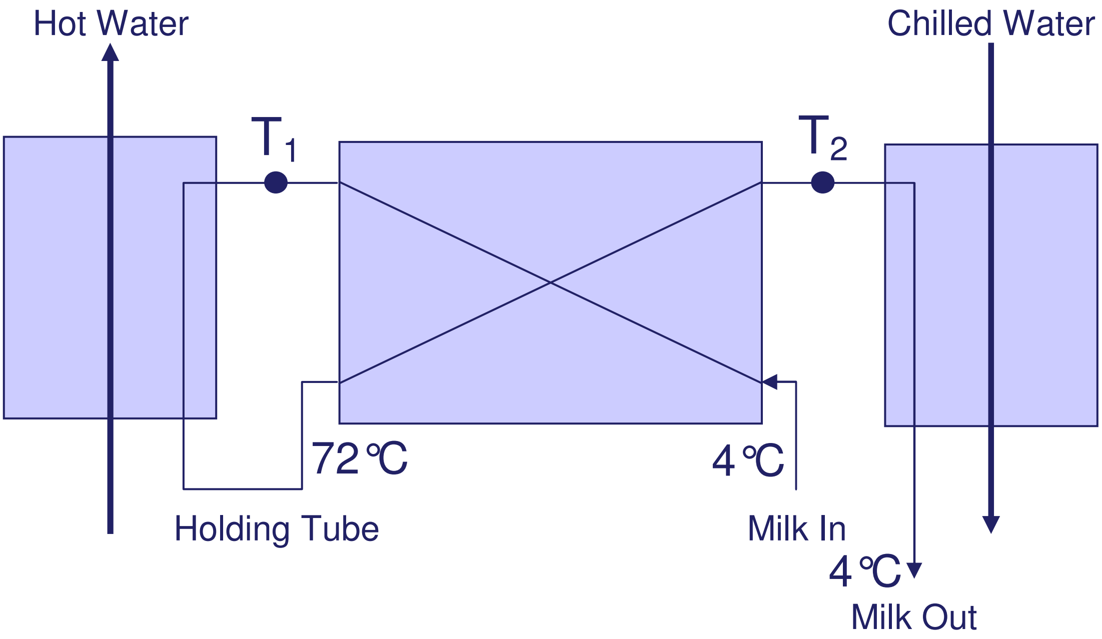
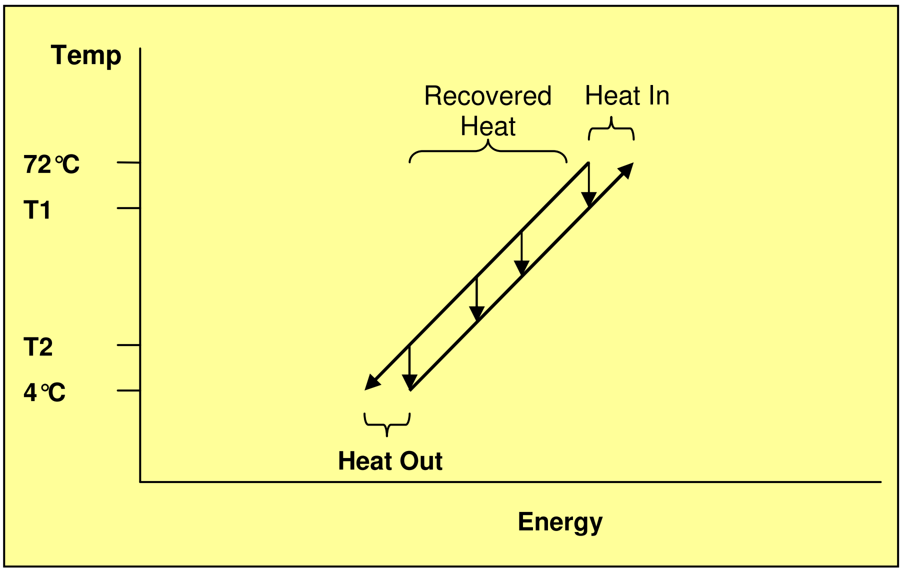


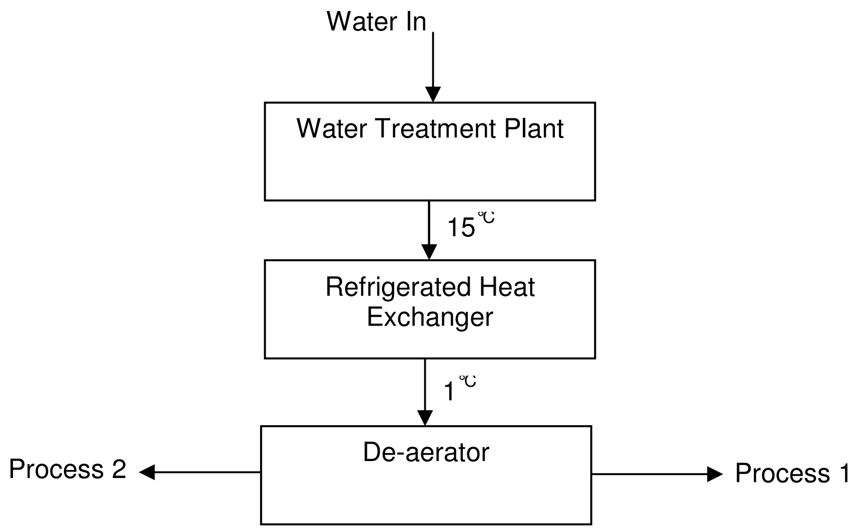
Figure 5 – Regenerative Pasteuriser, Energy Balance



Heat Load Case Study 2: Removing an unnecessary heat load

A site had a requirement for 2 large streams of de-aerated water. The layout of the plant was as shown in Figure 6. Towns water was treated, cooled to 1°C and then de-aerated.

Figure 6 Water Treatment and Cooling Layout



About half the de-aerated water went to each process, but only one of the processes needed cold water at 1°C! The other process worked fine with water at either 15°C or 1°C. By repositioning the de-aerator prior to the refrigerated heat exchanger it was possible to halve the cooling demand.

Heat Load Case Study 3: Avoiding "lowest common denominator" cooling

A common fault in many food and drink plants is to use a single large system for cooling loads at different temperatures. For example many breweries have a central chilled glycol plant, with glycol cooled to around -5°C. The glycol is pumped to all the cooling loads throughout he brewery. This type of system has practical advantages, but is it the most energy efficient and cost effective option?

The lowest temperature loads in a lager brewery are the beer chiller, a heat exchanger cooling lager to -1°C and the beer cellar, where tanks of beer are matured at that temperature. These loads define the need to have glycol at -5°C – they are the "lowest common denominator loads". Many of the other loads are much warmer. The biggest single load is the summer time requirement to cool incoming towns water from a peak of around 20°C to about 7°C. This warm load deserves it's own dedicated cooling system operating at a higher temperature.

If the load is added to the central glycol plant the evaporating temperature of the refrigeration system will be around -10°C. A far better design will be to have a dedicated 2 step chilling process (as described in Section 3.1). The steps would be:

Step 1: cooling the water from 20°C to 13°C, with an evaporating temperature of around +8°C.

Step 2, cooling the water from 13°C to 7°C, with an evaporating temperature of around +2°C.

The efficiency of this dedicated plant would be around 30% better than the glycol plant evaporating at -10°C. If a new plant is being built the efficient configuration would actually be cheaper than the glycol system as well!

4. Improving Condensers and Reducing Head Pressure

4.1 Background to the Energy Saving Opportunities

The efficiency of a refrigeration system is dependent on its temperature lift – the difference between the evaporating and condensing temperatures. The greater the difference the lower the efficiency – see section 2.1 for more information about this.

A refrigeration system moves heat from around the evaporator to around the condenser. High pressure superheated refrigerant de-superheats and liquefies in the condenser, losing the heat that it absorbed at the evaporator and compressor.

The condensing temperature depends on:

- **The condenser size (surface area)** – the larger the size, the lower the condensing temperature;
- **The condenser's condition** – a blocked or corroded condenser will reject heat less efficiently and therefore the condensing temperature will increase;
- **The air flow** – most condensers use air to remove the heat (with water in the case of an evaporative condenser). If the air flow is reduced or impeded the condenser will reject heat less efficiently;
- **The ambient temperature** – the higher the air temperature to which heat is rejected, the higher the condensing temperature. It is normal for the condensing temperature to vary with ambient temperature;
- **The presence of non condensable gases mixed with the refrigerant** – these include air and nitrogen, which can accumulate in the condenser, reducing the available heat transfer surface and increasing the condensing temperature.

If the condensing temperature is higher than necessary the capacity of the system is reduced because the compressor has to operate at a higher compression ratio. The compressor is less efficient and will have to run longer to provide the required cooling effect. The efficiency of the system therefore reduces. In extreme cases the capacity of the system is lower than the load, and the required process or storage temperature will not be reached.

Remember the rule of thumb – a 1deg.C increase in condensing temperature increases running costs by between 2 and 4%. What happens in the condenser is fundamental to system efficiency and running costs.

The temperature at which the refrigerant condenses (the condensing temperature) is directly related to the condensing (or head) pressure. The pressure measured anywhere on the high side of the system (between the compressor discharge and the entry to the expansion device) is the head pressure. The saturation temperature equivalent to this pressure is the temperature at which the refrigerant condenses.

Most pressure gauges show saturation temperature in addition to pressure. The one in the photo is measuring the head pressure on an ammonia system – the pressure is 8.1 bar gauge and the condensing temperature for ammonia is 22°C. On many systems this information is also displayed on the system controller.

The design information for your system should specify the condensing temperature, usually at maximum ambient. This will allow you to determine what to expect the condensing temperature to be at other ambient temperatures. For example, if the condensing temperature is specified as 45°C at a maximum ambient of 32°C, the condenser TD (temperature difference between the air and condensing temperatures) is 13deg.C. So if the ambient is 15°C, the condensing temperature should be 28°C. If it is higher the system is working less efficiently than it could. There is rarely a good reason for the condensing temperature to be higher.

If you do not have access to the design information you can use the following guide:

- For remote air cooled condensers the condensing temperature is typically 10 to 12degC above the dry bulb ambient temperature;
- For evaporative condensers the condenser TD is typically 10degC above the wet bulb ambient temperature;
- For air cooled condensing units the condenser TD is typically 15 to 20degC above the dry bulb ambient temperature.

Section 4.2 shows what can increase the head pressure on existing systems and how to avoid this. If you are specifying new systems you have the opportunity to minimise the head pressure:

- Use evaporative condensers if the heat rejection load is more than about 250 kW. The condensing temperature will be up to 5degC lower than with air cooled condensers (in line with the difference between the wet and dry bulb ambient temperatures);
- Avoid the use of air cooled condensing units where possible – a multi compressor central plant system plus a remote air cooled or evaporative condenser will have a running cost up to 40% lower than typical individual condensing units;
- Ensure condensers are sited so that the cooling air flow is not restricted or warmer than necessary;
- Specify a maintenance regime which ensures condensers are kept clean and in good condition.

4.2 How to Identify Condenser and Head Pressure Opportunities

The following sub sections show what increases the head pressure unnecessarily, how to identify the problem and recommended solutions.

4.2.1 Damaged or dirty condenser

Problem – The surface of the condenser can become blocked with air borne contamination which is drawn by the fans onto the condenser. In addition the condenser is prone to damage by air borne pollutants and by salt at coastal locations. It can also be damaged by impact, e.g. from fork lift trucks.

Identification – A visual inspection of the condenser will show whether this is a problem.

Solution – Condenser cleaning is one of the most important routine maintenance activities. For some locations annual cleaning is sufficient, but quarterly cleaning may be necessary in others. If a condenser is badly damaged then it should be replaced. This is an opportunity to improve the efficiency of the plant by installing a larger condenser.

4.2.2 Location

Problem – An air cooled or evaporative condenser needs to be sited where the air flow is not restricted or warmer than necessary. The following situations cause problems:

- Where air flow is restricted due to the proximity of a wall etc;
- Where multiple condensers are sited where air off one condenser can be drawn onto another. This is often a problem where multiple air cooled condensing units are sited together.

Identification – Measure the temperature of the air onto a condenser and check it is not higher than the ambient temperature.

Solution – Relocating condensers is usually the only solution, although in some cases baffles can be used to re direct air.

4.2.3 Housings

Problem – Most air cooled condensing units are located outside in weatherproof housings. The housing should be designed to provide good air flow through the condenser. However, in many cases the air can re-circulate around the condenser because the housing does not fit closely over the unit.

Identification – Housings usually have an open area for the air on face of the condenser. The air on face should not be obstructed by the housing, nor should there be gaps between it and the housing. In addition, there should be sufficient louvers to allow the warm air to exit the housing. Measure the temperature of the air onto a condenser and check it is not higher than the ambient temperature.

Solution – If the air onto the condenser is restricted then the housing should be modified to remove the obstruction. If there are gaps which allow the air to re-circulate, panels should be fitted all around the condenser (top, sides and bottom) to block off the re circulating air. It is important that a plenum is not produced as this will increase the air side pressure drop, reducing the air flow.

4.2.4 Non condensable gas in the system

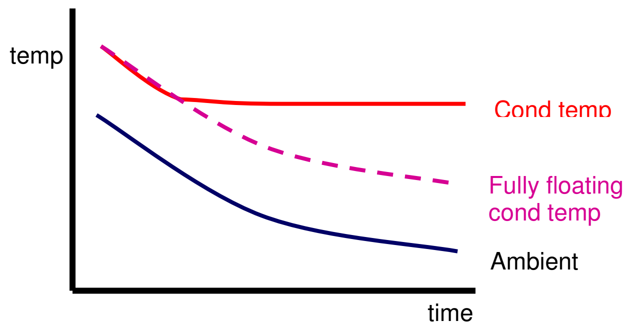
Problem – (1) Air can be drawn into the system if the evaporating pressure is below atmospheric pressure and there is a leak in the low side of the system (this is common in ammonia frozen food systems). (2) Nitrogen or air can remain in systems which have not been evacuated correctly during installation or service.

Identification – The air or nitrogen will accumulate in the condenser and increase the head pressure. The pressure of the refrigerant in the condenser at standstill should be equivalent to the temperature of the condenser at standstill. If the pressure is higher, there are non condensable gases in the condenser. The temperature should be measured with a probe thermometer.

Solution – (1) A non condensable gas purger should be fitted to the system. Ideally this should be an automatic refrigerated type to minimise the emission of refrigerant during purging. The operation of a system on a vacuum should also be reduced or eliminated by increasing the evaporating pressure if possible. (2) The system should be correctly evacuated during service. To remove the non condensable gas as a one off exercise the air can usually be carefully purged from the condenser if there is a suitable connection. If this is not possible the refrigerant should be recovered and the system evacuated and re charged. A qualified refrigeration engineer should do this.

4.2.5 Head pressure control

Problem – The condensing temperature (and therefore head pressure) should fall as the ambient temperature reduces. However, in many cases it is not allowed to float fully, but is controlled so that it does not fall below a pre set level, regardless of the ambient temperature. The chart shows this graphically for a time period between midday and midnight. The lower (blue) line is the ambient temperature. If the condensing temperature is allowed to fully float it follows the dotted line. If it is controlled then it will only float down to the set point and then remain at this condition (the red line).



Identification – The measured condensing temperature can be assessed by checking the head pressure and converting this to the saturation temperature (e.g. by using the temperature scale on the gauge for the refrigerant in the system). Section 4.1 gives an indication of how much higher the condensing temperature should be compared to ambient. The head pressure setting should be part of the commissioning documentation, or it may be accessible from the system controller.

Solutions – There are often good reasons why the head pressure setting is high, but in most cases it can be safely reduced:

- Expansion devices require a minimum pressure drop to work correctly. Electronic expansion valves need a minimum of 4 bar pressure drop and thermostatic expansion valves need a minimum of 6 bar. In most cases the head pressure can be reduced significantly without going below these minimum pressure drops. The actual setting will depend on the evaporating pressure – the head pressure can float lower for a frozen food application than for chilled food.
- Some screw compressors require a high head pressure for the oil system to operate reliably, typically ?? bar g. In this case the setting is determined by recommendations from the compressor supplier and it should not be reduced below this.
- Hot, warm or saturated gas defrost requires a higher head pressure when the system is on defrost. The head pressure must be at the level required when the system is defrosting. However, the control strategy can often be adjusted so the head pressure is floated lower when the system is not defrosting (i.e. for most of the time).
- Reducing the head pressure too low can cause liquid refrigerant to evaporate in the liquid line if it is routed through an area where the surrounding temperature is significantly higher than the outside ambient. The set point should not be lower than the temperature of the area through which the liquid line runs. For example, if the liquid line passes through an area maintained at 20℃, the control strategy should not allow the head pressure to drop below the pressure equivalent to this temperature. This would be 7.5 bar g for ammonia, 8.2 bar g for R22 and 10 bar g for R404A.

The head pressure is usually controlled by switching condenser fans or controlling their speed. There is an optimum point below which the power used by the fan motors is greater than the compressor motor saving. In this case it is better to switch the fans off. The optimum depends on the application and the type and number of fan motors, but it is likely to be a pressure equivalent to between 20℃ and 25℃ condensing temperature.

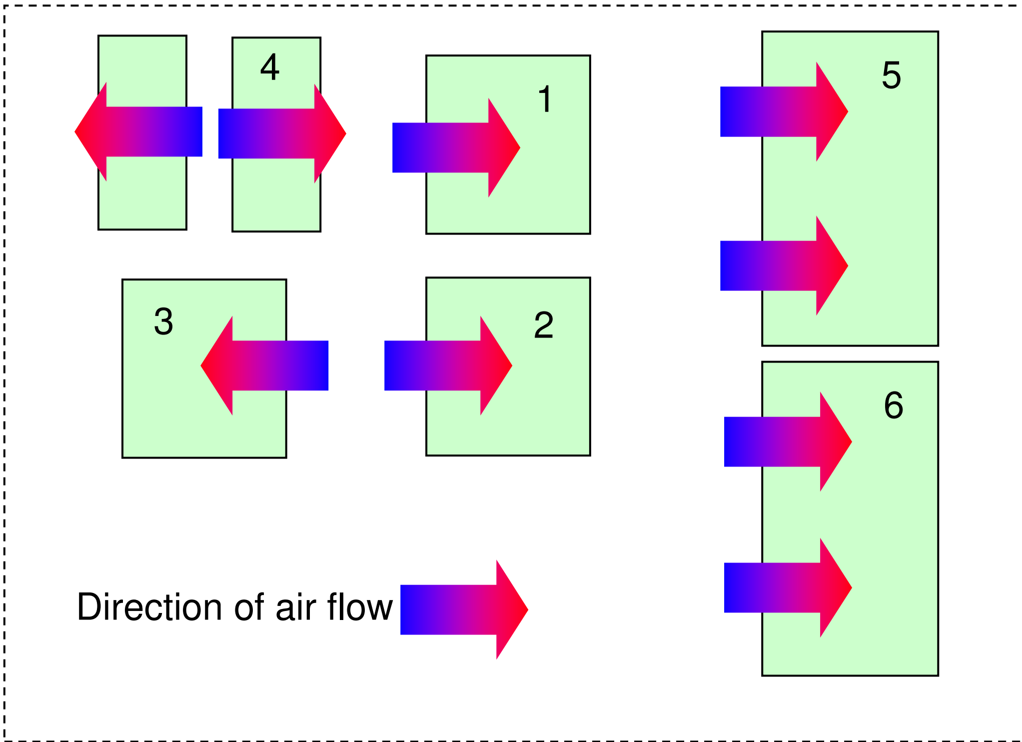
4.3 Condenser and Head Pressure Case Studies

The case studies below are from the surveys carried out during this Food and Drink Industry Refrigeration Efficiency project. The savings have been calculated using life cycle costing analysis, with actual kWh measurements where available, and based on a price of £0.07 / kWh.

Head Pressure Case Study 1 – location

In this installation seven condensing units are located on a roof area. The air off some of the units flows onto the condensers of other units, as shown in the photo and the plan view. The photo shows that some effort has been made to overcome the problem for unit 1 (not visible – it is behind the plywood panel), but the air off unit 4 now flows onto units 2 and 3 (indicated by the red arrows).

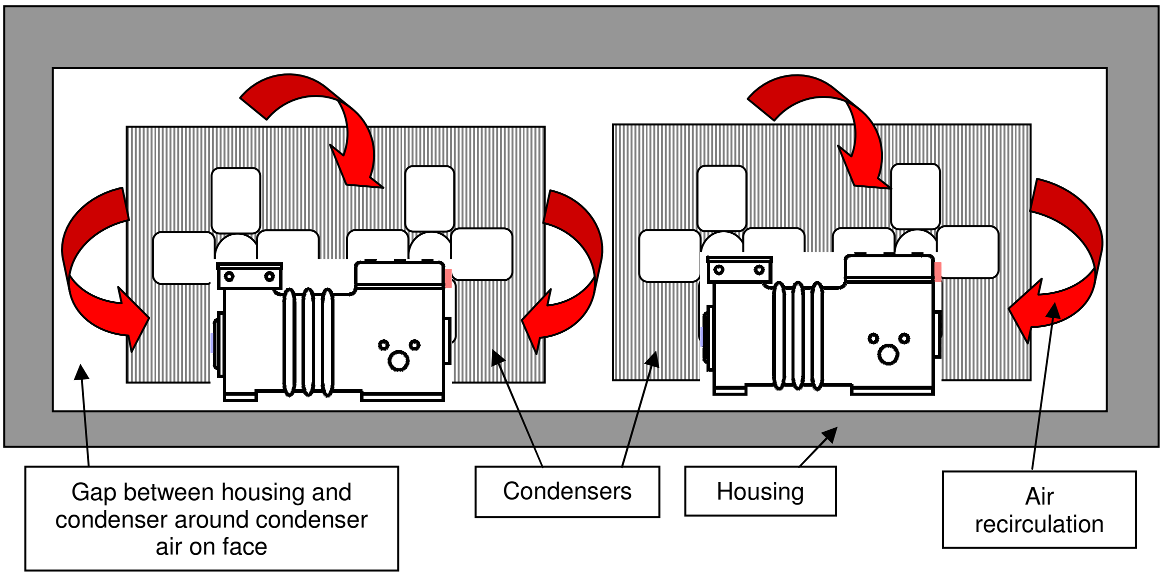
The distance between units 2 and 3 is approximately 0.75m. Units 1, 2 and 3 are the worst effected – the temperature of the air onto their condensers is 7degC higher than the ambient temperature. Life cycle cost analysis shows that this increases the power consumption of these units by 23%. In addition, the air off of units 1 and 2 increases the temperature of the air onto units 5 and 6 to a similar degree.



The solution is to either relocate these condensing units, or replace them with a multi compressor pack coupled to a remote condenser. The advantage of the second option, although more expensive, is that the remote condenser can be selected so that it operates with a lower condensing temperature than a condensing unit would.

Head Pressure Case Study 2 - Housings

This installation comprised two twin fan condensing units, each with a 4 cylinder 20 hp compressor. The housing is a purpose built enclosure, but it does not fit closely around the unit. The diagram below shows the front view. Air re circulates round the gaps between the air on face of the condenser and the housing, rather than being expelled through the louvers in the front of the housing.



The impact of the air re circulation is to increase the temperature of the air onto the condenser by up to 27degC. The situation is worse when one of each twin fan is off on the head pressure control.

Life cycle costing has shown that eliminating the air recirculation will reduce power consumption by 37% and running costs by £2690 / year. This can be achieved by simply blocking the gap between the housing and the air on face with plywood.

This system had another problem which increased its running costs. The receivers are located above the level of the condenser outlet. In this case the original receivers on the units had been replaced by new ones fitted above the compressors at the top of the condensers. The liquid formed in the condenser will therefore not drain into the receiver, and will instead back up in the condenser. This reduces the available condenser space for condensing refrigerant, and the condensing temperature and pressure increases to compensate. In addition the expansion valves and evaporators are starved of refrigerant. The solution is simply to relocate the receivers at or below the condenser outlet. It is not uncommon for receivers on systems with remote condensers and receivers to be located above the level of the condenser outlet, with consequent operational problems and a decrease in efficiency.

Head Pressure Case Study 3 – head pressure set point reduction

This site comprises 2 spiral freezers with screw compressors running on ammonia coupled to evaporative condensers, and four air cooled condensing units running individual cold room evaporators. In all cases the head pressure setting was higher than necessary.

	Spiral 1	**Spiral 2**	**Condensing units (4 off)**
Head pressure set point during survey	29 to 36°C	34 to 40°C	34°C
Recommended head pressure set point	25°C	25°C	25°C
Saving, %	10.5	22	10
Saving, £/year	4,800	4,800	2470

To achieve these savings of £12070 / year a simple change needs to be made to the controller. For the spiral freezers the electronic system controllers need adjustment. For the condensing units high pressure switches control the head pressure and need adjustment.

In both cases the head pressure is controlled by switching off condenser fans. The differential is such that the head pressure and hence condensing temperature varies widely (see the values for the spiral freezers above). Controlling the head pressure by the use of variable speed fans would be better as it would allow more stable, and hence more efficient, operation of the compressors. Section 6 provides more detailed information about savings associated with variable speed drive on condenser fan motors.

5. Improving Part Load Performance

5.1 Background to the Energy Saving Opportunities

Many plants are particularly inefficient when operating under part load conditions. When operating at part load it is common to find:

- Excessive compressor power because of large compressors being used at low load.
- High head pressure (and hence excess temperature lift) because the condenser design and control system prevents the full benefit of reduced heat rejection load and/or reduced ambient temperature.
- Low suction pressure (and hence excess temperature lift) because the evaporator design and control system prevents the full benefit of reduced cooling load.
- Excessive auxiliary power consumption (e.g. evaporator and condenser pumps and fans) because the auxiliaries run at 100% load even though the cooling demand is well below the peak.

Energy wastage of well over 20% is possible through poor part load operation. Part Load Case Study 1 in Section 5.3 below shows a saving of 50% achievable through simple modifications to the control system.

5.1.1 Understanding the Design Point and the Operating Envelope

The reason that so many plants operate badly at part load conditions is often that the system is only designed for a single set of operating conditions instead of the full range of conditions likely to be encountered. Ironically, the single set of "design point" conditions that is usually used for plant design and efficiency optimisation only occur for a very small percentage of the hours of the year.

The "design point" of a plant is usually defined as the peak cooling demand coinciding with the peak ambient weather conditions. These are the most arduous conditions, and it is important that the plant can deliver sufficient cooling at the design point. But, how often does the plant operate at the design point? In most cases for well under 5% of the year. This is because:

- The peak ambient temperature occurs for less than 5% of the year.
- In most industrial plants the cooling load is highly variable and, in many cases, only reaches a peak for a few hours per day. In some situations the plant never needs to run at full cooling load either because the original design was overestimated or because of changed circumstances since the plant was built.

To be able to design a plant to operate well at part load conditions it is essential to define the "operating envelope". This represents a set of conditions that define the way the plant is likely to operate throughout the year. Table 5.1 shows a simple example of the operating envelope of a glycol chiller. In this example:

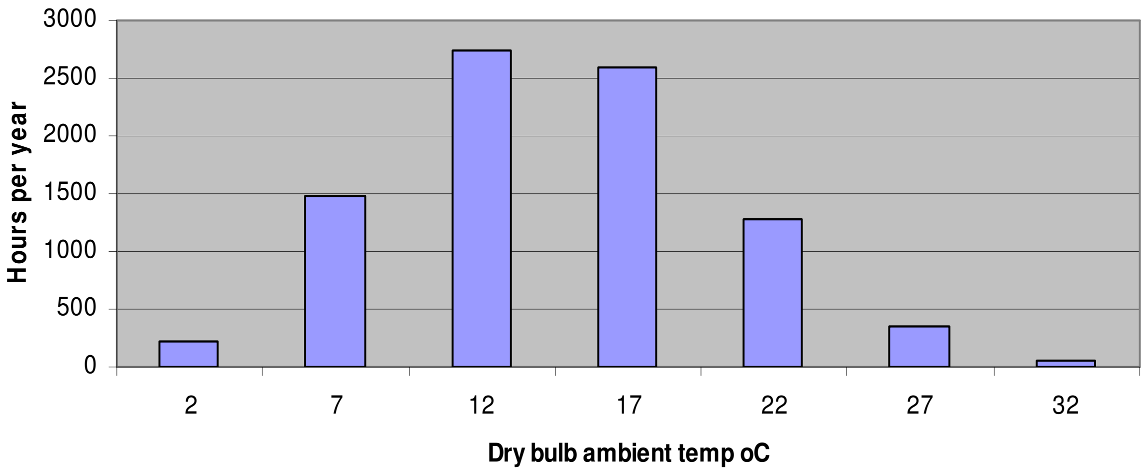
- The "design point" only occurs for 1% of the year.
- The most average level of cooling demand is only around 50% of full load.
- The most common ambient temperature is around 20 degC below the design point.

Figure 5.1 shows a typical range of ambient temperatures through the year. From the data in Table 5.1 and Figure 5.1 it is clear that a plant that is optimised for the "design point" of 1000kW at an ambient of 30°C is likely to run at poor efficiency for most of the year!

Table 5.1   Example Operating Envelope

Ambient Temperature	Cooling Load kW	Hours per Year	% of hours
25 to 30°C	1000	60	1%
	700	100	1%
	500	200	2%
15 to 25°C	950	500	6%
	600	700	8%
	400	1200	14%
5 to 15°C	900	800	9%
	550	1600	18%
	350	2500	29%
0 to 5°C	850	200	2%
	500	400	5%
	300	500	6%
	Total	8760	100%

Figure 5.1   Ambient Temperature Profile for London (Dry bulb)



5.2 How to Identify Part Load Opportunities

The starting point is to establish a reasonably accurate picture of the operating envelope. This is best done using a spreadsheet and should ideally have a bit more detail than shown in Table 5.1 with smaller steps of ambient temperature and cooling load.

If you are buying a new plant the table of data should be used by tenderers to provide energy consumption for each set of operating conditions. In that way you can compare the tenders on the basis of total annual energy consumption.

For existing plants it is still useful to establish the operating envelope as this can be used to identify those load conditions where plant performance is poor.

The data for the operating envelope need not be very accurate – it would require an unnecessary amount of work to try and establish accurate figures and, in most cases, it varies slightly from year to year anyway. A reasonable set of cooling load "guestimates" combined with published data for weather conditions should prove accurate enough – and a far better means of comparison than just using the design point!

Once the operating envelope has been established it should be reviewed with the following four opportunities in mind:

5.2.1 Efficient Compressor Operation at Part Load Conditions

All refrigeration systems are fitted with capacity control systems that enable them to provide a reduced level of cooling when a plant needs to run at part load. A key part of the control system is to reduce the capacity of the compressor(s). Many of these compressor capacity control systems lead to relatively poor efficiency at part load. The types of capacity control are typically as shown in Table 5.2.

Figure 5.2 gives typical figures for the effectiveness of these capacity control systems.

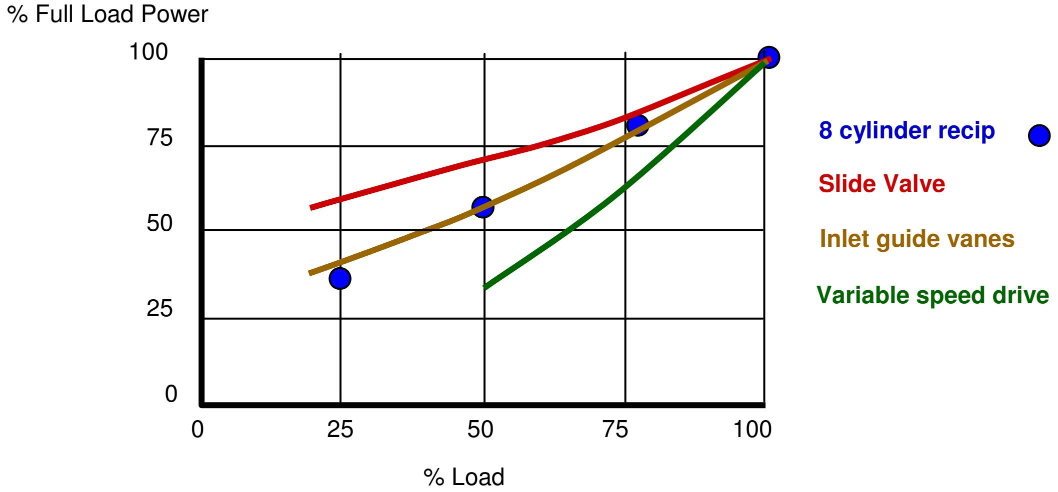
Table 5.2 Compressor Capacity Control

Compressor Type	Capacity Control System	Comments
Small compressors (e.g. <10 kW)	No capacity control	Part loads accommodated by on/off control of the compressor
Reciprocating compressors	Cylinder unloading	Giving 3 or 4 load settings, e.g. : • 6 cylinder compressor 100%, 66% and 33%; • 8 cylinder compressor 100%, 75%, 50% and 25%
Screw compressors	Slide valve unloading	Giving continuous control between 100% and 10% capacity
Centrifugal compressors	Inlet guide vane control and/or hot gas bypass	Giving continuous control between 100% and 10% capacity
Any compressor	Suction throttling	A particularly crude and inefficient way of controlling capacity.
Any compressor	VSD (variable speed drive).	In principle the most efficient form of capacity control but usually limited in application and often can only control down to around 50% capacity.

In terms of energy efficiency each type of capacity control system has different characteristics, as illustrated in Figure 5.2. This shows that most systems are reasonable at loads down to 70% but that the relative level of efficiency gets much worse at loads below 50%.

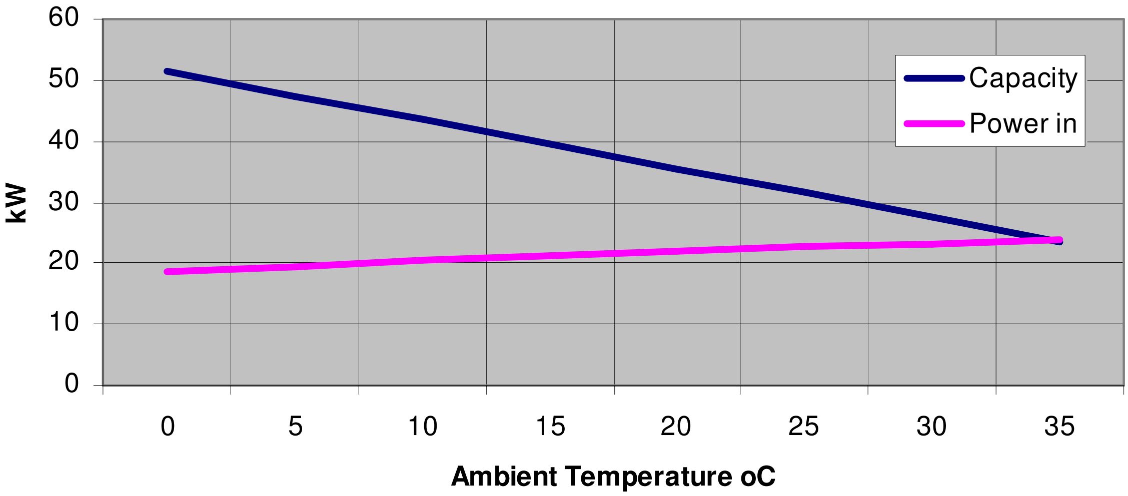
It is not uncommon to find large compressors running at very low loads for many hours at a time. For example in a factory with a large process load that only occurs during a single shift 5 days a week the plant could be at, say, 25% load for 120 hours per week. If the compressor used was a screw compressor, Figure 5.2 shows that the compressor is absorbing over twice as much power per unit of cooling than the design value at 100% load. This very wasteful situation can be avoided by ensuring that the correct sized compressors are installed in a system. By understanding the operating envelope you can assess what sizes of compressors are needed to avoid long periods of operation at a low setting of a compressor capacity control system.

Figure 5.2 Typical Characteristics of Compressor Capacity Control Systems



It is worth remembering that compressor capacity goes up as the head pressure falls. Assuming that the plant has fully floating head pressure (see (B) below) this will exacerbate the part load problem. The compressor becomes "oversized" at low ambient temperature. Figure 5.3 shows that a typical compressor has about 50% extra capacity when running at the average ambient of 10℃ compared to a design point at 30℃. So even if the cooling demand is at 100% of the design point value (which is unlikely in cool weather) the compressor is still well oversized for most of the year.

Figure 5.3 Relationship between Compressor Capacity and Ambient Temperature (assuming fully floating head pressure)



It is vital that a large refrigeration system has several compressors, so that at part load the cooling demand can be met by a compressor that is running close to full load. Ideally, a system should have compressors of different size available – that makes it easier to sequence the compressors to meet all load levels at a high level of efficiency.

For example, a system with a design point load of 1000 kW could be served by 4 compressors, 2 sized at 400 kW and 2 smaller ones, sized at 200kW. Assuming the small compressor can work reasonably efficiently at 50% load then the compressors can be sequenced to efficiently supply any load between 100kW and 1000 kW. Use of a VSD controller on one of the small compressors would provide the opportunity for really good part load efficiency.

It is obviously best if you get compressor sizing right when you are buying a new plant. However, the energy wastage can be so large that it is often cost effective to add an extra small compressor to an existing system.

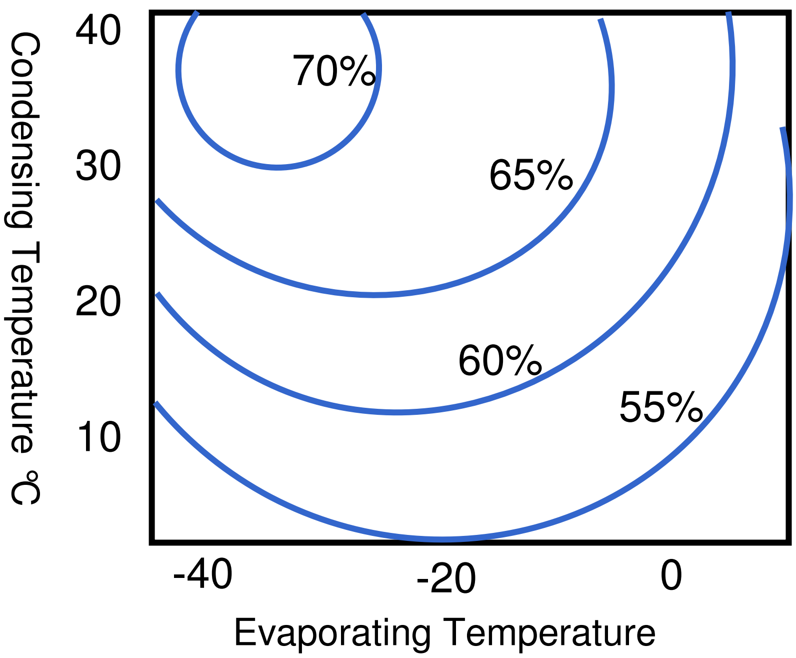
A common situation where part load losses occur is when there is one large compressor serving a variable cooling load. However, there can also be problems on multi-compressor systems that are not controlled and sequenced correctly. A frequently encountered situation is to find a 3 compressor plant running at about 30% load, with all 3 compressors running at a low part load setting. It is vital to modify the controls so that, in this situation, 2 compressors are turned off and one is running at full load.

5.2.2 Efficient Compressor Operation at High Load and Varying Ambient

It is important to recognise that the efficiency of a refrigeration compressor varies at different evaporating and condensing temperatures. If a compressor is selected for maximum efficiency at the design point it might be 10 to 20% less efficient at the more common operating conditions occurring at medium and low ambient temperature.

Figure 5.4 shows a "contour map" of compressor efficiency across a range of operating conditions. This example is based on a screw compressor and it has a peak efficiency at a low evaporating temperature (around -35℃) and a high condensing temperature (around 38℃). But for most of the year this compressor will run with a condensing temperature between 20 and 20℃ – with an efficiency around 10% below the peak. This is not a good selection across the whole operating envelope.

Figure 5.4 Example of Compressor Energy Efficiency Contours



5.2.3 Operation at Lowest Possible Head Pressure

A vital aspect of part load operation is to ensure that the system is running at the lowest possible condensing temperature. There are 2 interacting opportunities:

- As the heat load falls, the condenser becomes “oversized” and the temperature difference between the condensing refrigerant and the ambient can fall, which will allow the head pressure to fall.
- As the ambient temperature falls the head pressure should fall with it.

This topic has been dealt with in detail in Section 4 of this document – you should refer to this for more detail. Every effort should be made to reduce the head pressure at part load conditions. Very small investments can maximise this opportunity, especially for new plants.

It should be noted that under some conditions the extra auxiliary power needed to achieve minimum head pressure may outweigh the savings in compressor power. There is a need to identify an optimum balance between compressor power and auxiliary fans and pumps.

5.2.4 Operation at Highest Possible Suction Pressure

If cooling demand is well below the design point value it may be possible to raise the suction pressure of the plant, which will reduce the temperature lift and improve efficiency. At low load the evaporator is “oversized” and can do sufficient cooling at a higher evaporating temperature (which is equivalent to the suction pressure.

A common situation is that several loads are on the same system and that the suction pressure can be raised significantly when the “lowest common denominator” load is not operating. This is illustrated in Part Load Case Study 3, below.

5.2.5 Efficient Control of Auxiliaries at Part Load

A further very important aspect of part load operation is the use of auxiliary pumps and fans to circulate fluids through the evaporator and condenser.

At full cooling load it is typical for the total auxiliary power to be about 15% to 20% of the compressor power. For example, a system running at full load with 100 kW of compressor power might have 20kW of evaporator and condenser pumps and fans.

For many systems the auxiliary power input is fixed, so at part load it becomes a dominant part of the energy consumption. For the above example at 25% load, the compressor might absorb, say, 30kW. The auxiliaries are still absorbing 20kW, which is now over 60% of the compressor power!

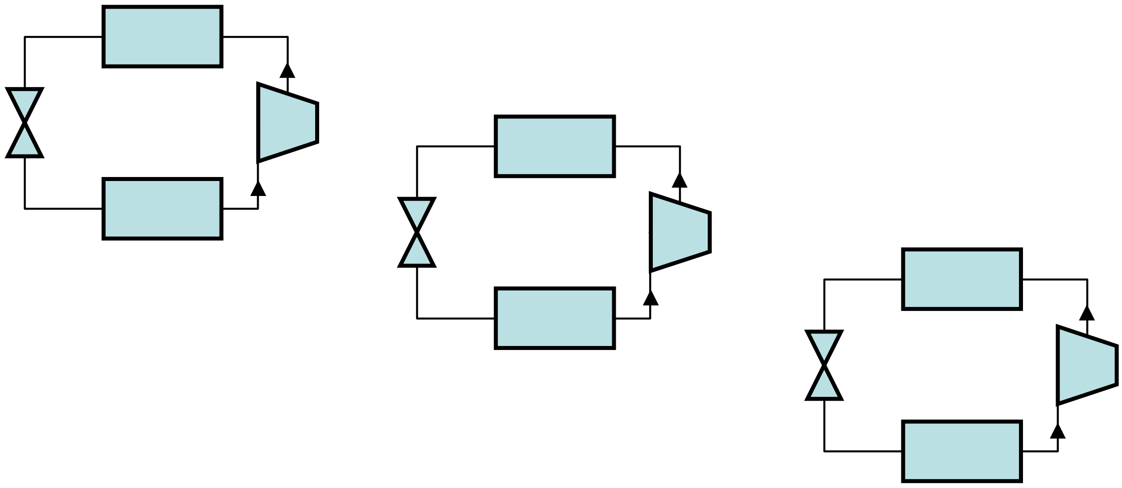
It is essential to reduce the input of auxiliary power at low part load conditions. Ways of doing this are dealt with in Section 6 of this document which covers pumps and fans.

5.3 Part Load Case Studies

Part Load Case Study 1: Poor Sequencing of Chillers

A confectionary plant used numerous water chillers for cooling of product. One of their systems included 3 “modular” systems, each with a completely separate refrigerant circuit as illustrated in Figure 5.5.

Figure 5.5 Modular Water Chiller System, 3 Chillers



The control system for these plants was very badly configured. A very common operating condition was at about 30% of design point capacity. The system was configured in such a way that all 3 modules were running at 30% load to satisfy this condition. The absorbed power was as shown in Table 5.3.

Table 5.3 Chiller System Power (Poor Part Load Control)

Item	System Number	Load	Power
Compressors	1	30%	90
	2	30%	90
	3	30%	90
Evaporator chilled water pumps	1	100%	20
	2	100%	20
	3	100%	20
Condenser cooling water pumps	1	100%	25
	2	100%	25
	3	100%	25
		TOTAL POWER	**405**

A simple change to the control system ensured that only one chiller operated when the load was 30%. The savings were dramatic – more than 50% was saved as shown in Table 5.4. There are 2 key aspects of this improvement:

- The auxiliary pump load was reduced by 66%, because 2 sets of pumps were switched off.

- The compressor power absorbed was significantly less, because the part load performance of each compressor was poor – at 30% load each compressor absorbed 60% of full load power.

Table 5.4 Chiller System Power (Improved Part Load Control)

Item	System Number	Load	Power
Compressors	1	100%	150
	2	0%	0
	3	0%	0
Evaporator chilled water pumps	1	100%	20
	2	0%	0
	3	0%	0
Condenser cooling water pumps	1	100%	25
	2	0%	0
	3	0%	0
		TOTAL POWER	**195**

Part Load Case Study 2: Poor Suction Pressure Control

A food freezing plant had 2 large loads on a low temperature ammonia plant. These were:

- A blast freezer, which operated 10 hours per day, 5 days per week.
- A large cold store, which operated 24 hours per day, 7 days per week.

The loads were on a single pumped circulation plant which was controlled by suction pressure control. The pressure setting was equivalent to an evaporating temperature of -42°C. This very low evaporating temperature was required because of the way the blast freezer was designed and operated.

The cold store was holding product at -23°C. The design of the cold store evaporators required an evaporating temperature of -32°C at peak load conditions. For much of the time the load on the cold store was less than the peak design point figure and the store could be cooled adequately with an evaporating temperature of up to -28°C.

Ideally, these loads should not be on the same refrigeration plant because the blast freezer requires an evaporating temperature that is always 10 degC below that of the cold store. Unfortunately the plant design and layout made modification to separate systems too costly to justify. However, a simple change to the control strategy provided a significant proportion of the savings. Two improvements could be made:

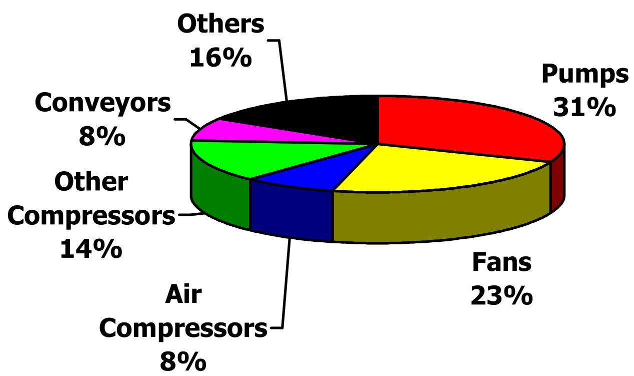
- A simple 2-level control strategy could be implemented. For 50 hours per week, when the blast freezer was running, the control set point for the evaporating temperature was set to -42°C. For the rest of the week (about 70% of the time), the set point was raised to -32°C, providing reduced temperature lift and a saving of 20%.
- An adaptive pressure control system could be used to optimise the evaporating temperature to the highest possible level. During periods of low activity and low ambient temperature the evaporating temperature could rise up to around -28°C, providing a further 10% saving.

6. Reducing Power Consumed by Auxiliary Pumps and Fans

6.1 Background to the Energy Saving Opportunities

Fans and pumps represent the largest users of electrical motive power across the whole range of industrial and commercial applications in the UK.

Figure 6.1

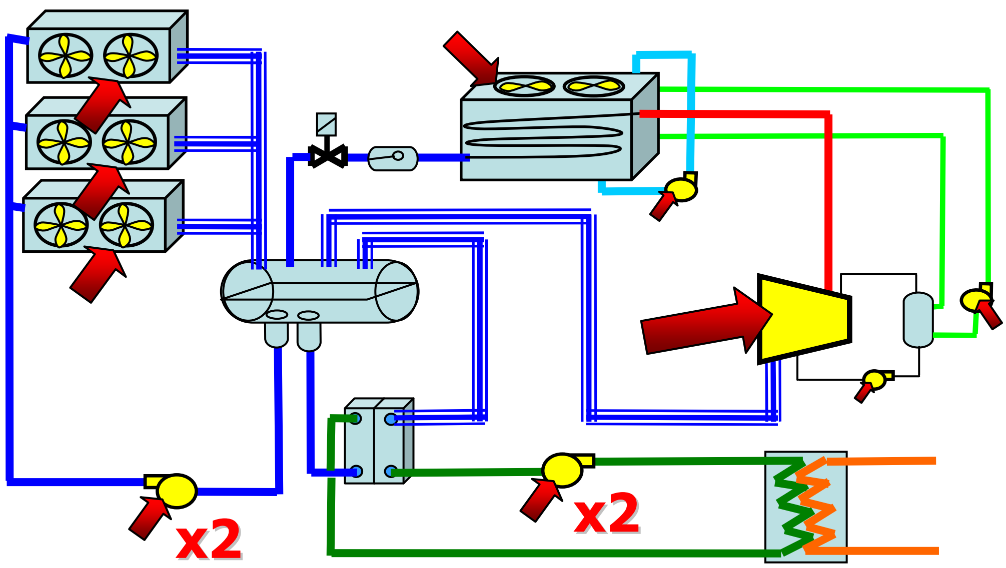


It is estimated that in the UK pumps and fans account for the consumption of 35 TWh / annum of electricity and the emission of 3.7 million tonnes of carbon. It is also the case that in many instances pumps and fans are operated inefficiently. The reasons for this will vary from process to process and application to application, but the outcome is always the same, with a cost to industry through wasted energy and a cost to the environment through the emissions caused by the generation of this wasted energy.

6.1.1 Pumps & Fans in Refrigeration Systems

In terms of refrigeration systems the focus on efficient operation is often on the compressors and their associated drive motors, as they are invariably the largest drives associated with the systems. However, the operation of these refrigeration systems is also crucially dependent on a myriad of auxiliary drives, required to move liquids and air that are being passed through various heat exchangers, ie evaporators, condensers and tertiary heat exchangers.

Figure 6.2



The efficient operation of these drives is also important for a number of reasons, most crucially because:

a) At certain operational conditions, i.e. periods of low cooling demand, the electrical load demand from these auxiliary drives can become predominant.

b) Some of the auxiliary drives, i.e. those in the cooling stream, be it cold air or a chilled liquid, need to be paid for more than once. That is, there is a direct cost associated with the electricity consumption required to run the auxiliary drives and an indirect cost associated with the electricity consumption required to run the compressor to extract the heat they have put into the cooling stream.

   While the total electrical cost is not normally twice as much, for low temperature applications, e.g. cold storage or freezing applications, it can easily be 1.5 times the rating of the auxiliary drives in the cooling stream.

6.1.2 Power Considerations for Rotodynamic Pumps & Fans

The power consumed by rotodynamic pumps and fans, eg centrifugal pumps and axial fans, follow certain proportionality rules:

$$\textit{Power} \propto \rho,\ D^5\ \&\ N^3$$

Where ρ = fluid density – normally effectively fixed for a given application

D = impeller / fan diameter

N = impeller / fan speed

The implication of these relationships is important for the energy efficient operation of pumps and fans as they indicate that:

1. Minimising impeller diameter will deliver large benefits in reduced power consumption, e.g. a 10% reduction in diameter reduces power consumption by 41%
2. Operating speeds should be reduced as far as practicable to reduce power consumption, e.g. a 10% speed reduction reduces power consumption by 27%

While 1 is a decision that is most easily achieved at the design and procurement phase of the supply of new equipment, the ability to achieve 2 can be considered at either the design and procurement phase or during the operational phase of a pump or fans life. In either case the most appropriate method for reducing operating speeds is by the provision of Variable Speed Drive (VSD) systems.

6.1.3 Rotodynamic Pump Design for Efficiency

Designing a pump for energy efficient operation starts with defining the requirements of the system it is to serve. The objective of a pumping system is to transfer a liquid from one point to another, e.g. from the evaporator in a chiller to a number of process heat exchangers and back again. To make the liquid flow, a pressure must be generated that is large enough to overcome the head “losses” in the system. These losses are of two types, static and friction.

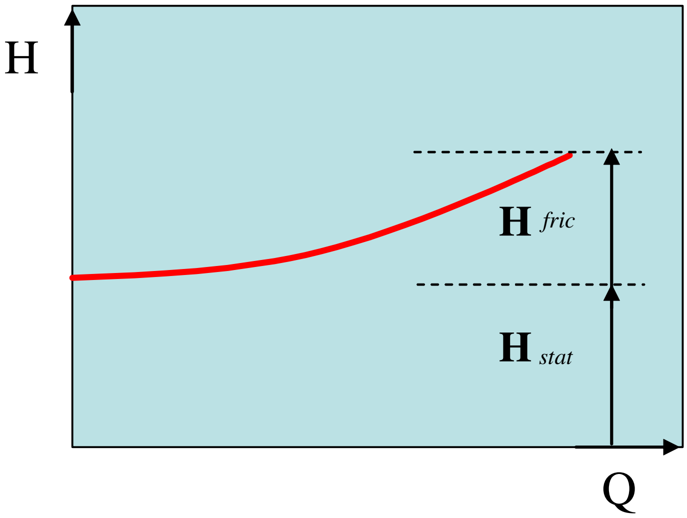
Static head, H^{stat} in Figure 3, is simply the difference in height between the source point, e.g. the chiller, and the destination point, e.g. the process heat exchangers. For a given application this is a fixed or constant requirement.

Friction head, H^{fric} in Figure 3, sometimes called the dynamic head, is the friction loss caused by the liquid being moved through pipes, valves and heat exchangers in the system.

$H^{fric} \propto Q^2$, where Q = the flow rate.

So for a given system design; in terms of length and run of pipework, pipe size and number and sizes of valves and heat exchangers; increasing the flow rate increases the friction losses exponentially.

Figure 6.3



Most fluid systems have a combination of static and friction head. The ratio of static to friction head over the operating range influences the benefits achievable from variable speed drives. Essentially, if the static head dominates then the opportunities to benefit from VSD systems are more limited then for systems where static and friction head are more balanced or where friction head dominates.

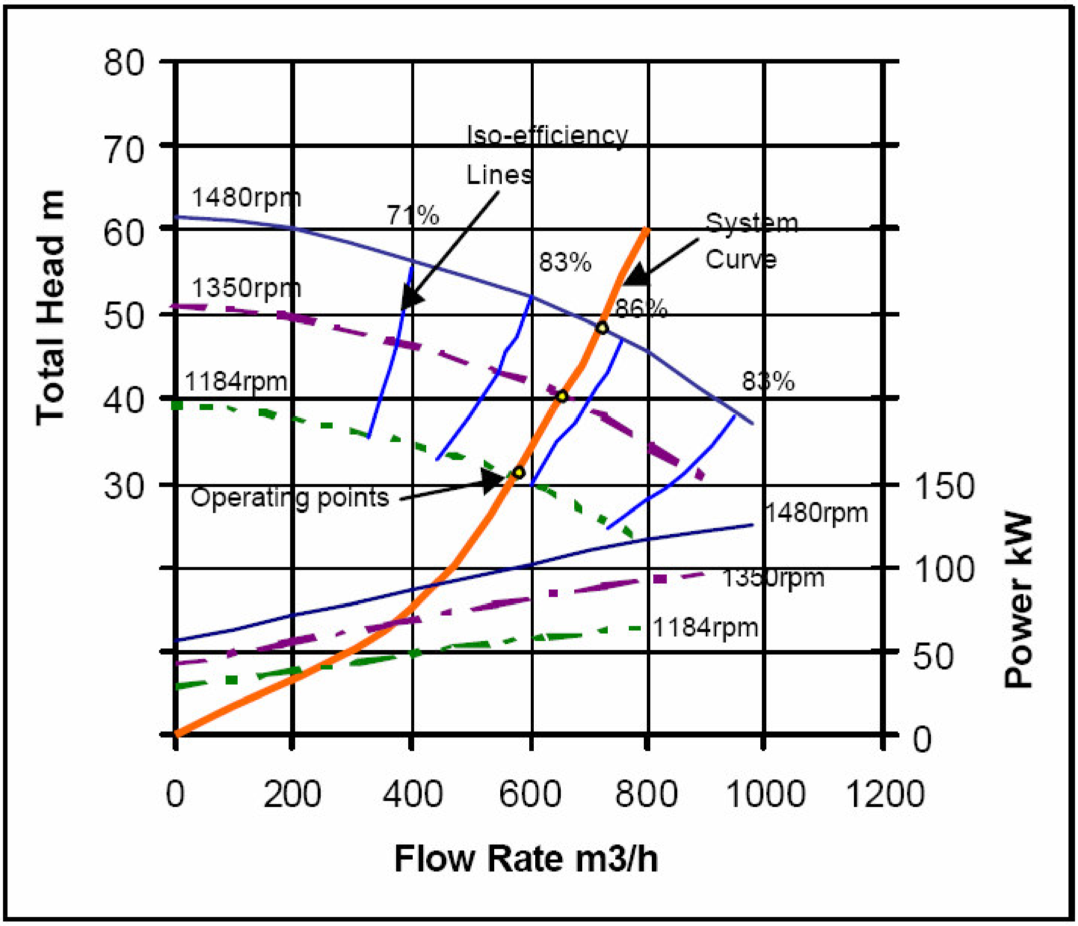
The general rules for maximising the fluid system performance and consequently reducing pumping power requirements are:

- Minimise the static head
- Minimise friction losses
- Minimise flow requirements

Once the fluid system has been designed, and the consequent static and friction head characteristics fixed, appropriate pump selection becomes the next important requirement. Whether the intent is to run a pump at a fixed speed or to fit a VSD system, the key issue is to match the high efficiency region of the pump curve to the operating envelope of the system curve. For fixed speed operation this will be simple, as there is only one proposed operating point. For a VSD system this is more complex, but, as shown in Figure 4, by superimposing the system curve over the pump curve a clear picture will be provided of the suitability of a particular pump selection for the proposed application. In the case of the pump represented by Figure 4, it meets the criteria for a good selection, since as the pump speed is reduced across its proposed range, from 1480 rpm to 1184 rpm, the system performance curve parallels the lines of constant ("iso") efficiency, indicating no loss of pump performance.

What is important to note, from an energy efficiency perspective, is that the 20% reduction in pump speed and flow rate, translates to a nearly 50% reduction in power consumed by the pump. This is the potential prize that fitting VSD systems to pumps offers, if the fluid system characteristics, i.e. variations in cooling demand and friction head, are assessed to provide the opportunity.

Figure 6.4 – Ref: "VSD Pumps, Best Practice Guide, BPMA"



6.2 Identifying Auxiliary Pump and Fan Opportunities

Identifying opportunities requires an understanding of the primary refrigeration plant and the processes they serve, both in terms of cooling loads and load profiles, and of component type and numbers. As with Section 3 the starting point is to make a comprehensive list of all cooling loads and to identify certain key characteristics of the loads. Section 3.2 provides guidance on the approach that should be adopted in preparing the list. As well as the loads, information on the pumps and fans serving the systems should be added, including their individual motor rating, in kW, and their existing method of operation and control. For example a process pump might be described as continuous running, fixed speed operation, with manual operator initiated control, and condenser fans as variable running, fixed speed operation, with automatic step control based on discharge pressure.

With this data it is possible to review where opportunities may lie.

Pumps

Typical in food factories, diaries and breweries are large secondary chilled fluid circulation systems, operating with one or more large pumps. In many cases the pumps are set to run continuously. These systems can often benefit from the fitting of some means of automatically switching them ON & OFF and / or VSD to the pumps. Typical system operation, which may provide opportunities for energy saving include:

- A fixed speed pump serving a single process heat exchanger with varying cooling load, utilising a 3-port spill / by-pass valve or an in-line pressure control valve to regulate flow. Control of the pump operation is often manual, which typically means it is never switched off, even if there is no load on the process heat exchanger.

- A fixed speed pump or pumps on a common main, serving multiple process heat exchangers with varying loads, utilising 3-port spill / by-pass valves and / or an in-line pressure control valve to regulate flow. Control of pump(s) operation may be manual or semi-automatic via a pressure sensor.
- A VSD pump is fitted on a common main, serving multiple process heat exchangers with varying loads. Flow control on to some of the process heat exchangers may be by 3-port spill / by-pass valve. Speed control is via a differential pressure sensor fitted across the flow and return lines. This sensor is fitted either local to the pump or out at the end of system. Symptomatic of incorrect operation of this system arrangement would be:
    - The pump speed is constant, often at full speed, under all cooling load conditions.
    - The flow and return temperature differential is smaller than design, i.e. the system is designed to send glycol out at –5℃ and return for chilling at 0℃, but instead it is coming back at –2.5℃.

These symptoms are often caused by the imbalance between the hydraulic, i.e. flow, and thermal, i.e. cooling, loads, caused by the 3-port spill / by-pass valves not allowing the former to match the latter.

This symptom has a particularly detrimental effect on cooling systems with multiple, modular packaged chillers, as along with not allowing the pump speed to vary, and hence save energy, it can also lead to multiple chillers operating at part load conditions. This is an inefficient way to operate chillers, particularly when screw compressors are fitted.

Replacement of the 3-port spill / by-pass valves with 2-port modulating valves or blanking of the by-pass port can alleviate this problem.

Fans

Fans that may offer energy saving opportunities are found on the evaporators and condensers on refrigeration systems. In cold and chill stores, the fans on coolers account for a significant part of the cooling load, between1/4 and 1/3 in some large cold stores. On condensers, they are found on both air-cooled and evaporative types and are a significant electrical load.

- Evaporator Fans

  Typical issues that should be investigated are continuous running of evaporator fans, even when the cold / chill store is duty satisfied. “Smart” control strategies should be considered, for example the possibility of fitting VSD systems or the switching / cycling of fans as the store temperature approaches the duty satisfied condition.

  Keeping evaporator fans running when a store is duty satisfied is often justified on the grounds of wanting air circulation at all times. If this is the case, consideration should be given to fitting auxiliary booster fans, which will have lower rated motors then the store evaporator fans, and switching them on during duty satisfied periods. However, it should be noted that many stores are operated effectively with no fans operating during periods when they are duty satisfied.

- Condenser Fans

  Most fans on condensers are operated as ON or OFF devices, control of them being associated with the control of compressor discharge pressure. There are benefits to be derived from fitting VSD systems to condenser fans, whether they are operating on air-cooled or evaporative units.

Proper assessment of pumps, fans and the systems they serve are important before making the commitment to invest in VSD technology. While there is significant case history supporting the benefits of the application, care must be taken to understand, for example, the range of speed control operation, so as not to forfeit the larger related savings that come from efficient operation of refrigeration systems. Particularly, reducing flow too far in heat exchangers operating at part load could have a detrimental effect on their performance, as the heat transfer coefficient might collapse. This could lead to higher discharge pressures then necessary or poor chiller and cooler performance.

6.3 Auxiliary Pump and Fan – Site Visit Observations

Observations from the various site visits conducted as part of the refrigeration efficiency initiative highlighted that the suggested opportunities detailed in Section 6.2 were implemented at some of the sites. However, the majority of sites visited had not reviewed or implemented the opportunities detailed, and the recommendations in the various site specific reports encourage this be done.

In terms of general observations the following were noted across many of the sites:

- Condenser fan operation was often not optimised, many not running when they should have been. Step control on multiple condenser fan systems meant that full condensing capacity was not utilised until the discharge pressure was much higher than it should have been. Very few condenser fans were fitted with VSD systems, although encouragingly some are now in the process of doing so.
- Cooler fans in a number of temperature controlled stores run excessively. Not switched off or sequenced for cycled running when the stores were duty satisfied.
- Fixed speed pumps run in preference to ones with VSD systems fitted. This seems to have been the case in some instances due to failures out at process heat exchangers, where use of 3-port spill / by-pass valves prevented reduction in flow, even where cooling demand had reduced. This has led to severe imbalance between hydraulic and thermal loads, compounded by excessive part load operation of multiple chillers.
- Pumped circuits with high pressure loss components and redundant branches where flow was allowed to continue.
- Systems where little or no instrumentation is fitted and those systems where instrumentation is fitted but where poor calibration means the data provided is suspect. It is much more difficult to make assessments on potential opportunities for energy efficiency improvements without substantiating data.

Appendix 1 Glossary

Auxiliary energy	Energy used by devices in a refrigeration system other than the main compressors – usually this refers to pumps and fans.
Coefficient of Performance (COP)	A way of expressing the efficiency of a refrigeration plant. Defined as Cooling carried out divided by Energy input.
Compressor suction and discharge pressures	The pressure at the inlet and outlet of a refrigeration compressors.
Cooling duty	The total cooling load on a system.
Cooling load	The total amount of cooling carried out by a refrigeration plant – usually made up of several individual heat loads.
Dry bulb temperature	The "normal" temperature of ambient air that does not depend on the relative humidity of the air. (See wet bulb temperature)
Heat rejection temperatures	The temperature that a refrigeration plant rejects heat from the condenser, usually into ambient air or cooling water.
HP liquid receiver	A High Pressure Receiver, a vessel located beneath a condenser, used a reservoir of liquid refrigerant.
L%	Load setting of compressor in percent of full load.
P_d	Discharge pressure at compressor exit, which is approximately equal to the condensing pressure.
P_s	Suction pressure at compressor inlet, which is approximately equal to the evaporating pressure.
Pow_A	Power being absorbed by auxiliaries.
Pow_C	Power being absorbed by compressor.
Reciprocating compressor	A type of refrigeration compressor using a piston to compress vapour trapped in a cylinder..
Screw compressor	A type of refrigeration compressor using a rotating screw to trap a volume of vapour and compress it.
Suction strainer	A strainer at the inlet of a compressor designed to prevent damage caused by small objects entering the compressor.
T_{amb}	Ambient temperature
TC_{in}	Temperature of cooling medium (e.g. ambient air or cooling water) as it enters a condenser.
TC_{out}	Temperature of cooling medium (e.g. ambient air or cooling water) as it leaves a condenser.
TE_{in}	Temperature of fluid being cooled as it enters an evaporator.

TE_{out}	Temperature of fluid being cooled as it leaves an evaporator.
Temperature Lift	The temperature difference between the “cold end” of a refrigeration plant (the evaporator) and the “hot end” (the condenser).
TEV	Thermostatic expansion valve, a commonly used expansion valve used on small and medium sized refrigeration systems.
T_{liq}	Refrigerant liquid temperature at condenser exit.
T_s and T_d	Refrigerant vapour temperatures at compressor suction and discharge.
VSD	Variable Speed Drive – a method of motor speed control using a variable frequency drive.
Wet bulb temperature	The temperature of ambient air that accounts for the relative humidity of the air. It can be measured by wetting the bulb of a mercury in glass thermometer – in periods of low humidity the evaporation that takes place cools the thermometer bulb. (See dry bulb temperature)

Appendix 2 Sources of Further Information

Food and Drink Federation	Trade association for food and drink manufacturers	www.fdf.org.uk
Institute of Refrigeration	Professional body for refrigeration and air conditioning engineers	www.ior.org.uk
British Beer and Pub Association	Trade association for brewing and pub sector	www.beerandpub.com
Dairy UK	Trade association for dairy sector	www.dairyuk.org
Cold Storage and Distribution Federation	Trade association for the temperature controlled supply chain	www.csdf.org.uk
British Refrigeration Association	Trade organisation for companies in the refrigeration and air conditioning industry	www.feta.co.uk
Carbon Trust	Information and support regarding climate change issues.	www.carbontrust.co.uk
Guide 1	Appointing and managing refrigeration contractors	www.ior.org.uk
Guide 2	Procurement of new plant	www.ior.org.uk
Guide 3	Checklist for operational improvements	www.ior.org.uk
Guide 4	HCFC phase out and F gas regulations	www.ior.org.uk
Guide 5	Reducing heat loads	www.ior.org.uk
Guide 6	Avoiding high head pressures	www.ior.org.uk
Guide 7	Improving part load performance	www.ior.org.uk
Guide 8	Reducing auxiliary fan and pump power	www.ior.org.uk
EN378	Refrigerating systems and heat pumps. Safety and environmental requirements	www.bsi-global.com
Refrigeration and Air Conditioning	Comprehensive text book covering all aspect of refrigeration and air conditioning	ISBN 0-13-323775-3
GPG 278	Purchasing efficient refrigeration – the value for money option	www.carbontrust.co.uk
www.ior.org.uk		
GPG 279	Running refrigeration plant efficiently – a cost saving guide for owners	www.carbontrust.co.uk www.ior.org.uk
GPG 280	Energy efficient refrigeration technology – the fundamentals	www.carbontrust.co.uk www.ior.org.uk
GPG 347	Installing and commissioning of refrigeration systems	www.carbontrust.co.uk www.ior.org.uk
GPG 364	Service and maintenance technicians guide	www.carbontrust.co.uk www.ior.org.uk
RAC	Monthly subscription trade journal and year book	www.emap.com