GOOD PRACTICE GUIDE 283

Designing energy efficient refrigeration plant

GOOD PRACTICE GUIDE 283

ENERGY EFFICIENCY

BEST PRACTICE PROGRAMME

DESIGNING ENERGY EFFICIENT REFRIGERATION PLANT

This Guide is No. 283 in the Good Practice Guide Series. The Guide covers commercial and industrial refrigeration systems and offers guidance on the design of efficient plant – from the selection of simple components to the design of large and complex systems. It is not necessary to read all the Guide: use the Designer's Efficiency Toolkit in Section 3 to identify the parts required.

This Guide is aimed at designers of refrigeration systems, and at those procuring plant to help get the best deal from designers.

For an explanation of basic refrigeration technology, see Good Practice Guide 280 *Energy efficient refrigeration technology – the fundamentals*.

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FOREWORD

This Guide is part of a series produced by the Government under the Energy Efficiency Best Practice Programme. The aim of the programme is to advance and spread good practice in energy efficiency by providing independent, authoritative advice and information on good energy efficiency practices. Best Practice is a collaborative programme targeted towards energy users and decision makers in industry, the commercial and public sectors, and building sectors including housing. It comprises four inter-related elements identified by colour-coded strips for easy reference:

— *Energy Consumption Guides:* (blue) energy consumption data to enable users to establish their relative energy efficiency performance;

— *Good Practice Guides:* (red) and *Case Studies:* (mustard) independent information on proven energy-saving measures and techniques and what they are achieving;

— *New Practice projects:* (light green) independent monitoring of new energy efficiency measures which do not yet enjoy a wide market;

— *Future Practice R&D support:* (purple) help to develop tomorrow's energy efficiency good practice measures.

If you would like any further information on this document, or on the Energy Efficiency Best Practice Programme, please contact the Environment and Energy Helpline on 0800 585794. Alternatively, you may contact your local service deliverer – see contact details below.

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WHY IS EFFICIENCY IMPORTANT?

For business and environmental reasons

Refrigeration is very important on a national scale. It accounts for 14% of UK electricity consumption. Most systems use more energy than necessary because of the way they are designed, serviced and operated.

Efficiency can lead to big cost savings for users: systems typically cost six times as much to run over their lifetime as they do to buy, and yet simple measures can cut energy consumption by around 20%. Savings contribute directly to profits. An energy efficient system is usually a more reliable one which brings lifetime savings from less repair and downtime.

WHAT'S IN IT FOR ME AS A SYSTEM DESIGNER?

Better business

More end users are making lifetime cost an important part of their specifications, and this will continue to increase in importance.

Designing efficient systems should get you more business and help you to gain credibility with customers as an energy efficient system will reduce their lifetime costs.

Adopting these principles will enhance the environmental image of your company and the industry.

IS THIS RELEVANT IF I JUST USE COMPUTER SELECTION AND SIZING?

Yes!

A good computer software package will calculate the energy consumption of a system and should allow you to optimise, within limits, design operating conditions and to select different components which use less electricity.

WILL THIS TAKE MUCH LONGER?

Not necessarily

Designing different options will probably take more time to begin with, but use all the tools available to speed things up. In particular, use computer software to design and cost different system options. This will help to demonstrate the advantages of a more efficient system. Component suppliers should also be able to provide information which will make selection simpler and faster.

EFFICIENCY IS IMPORTANT AND ACHIEVABLE

WILL IT TAKE SPECIALIST KNOWLEDGE?

No

Efficiency is about applying current knowledge and existing technology. To understand more clearly how a system works and the factors that affect efficiency, read GPG 280 *Energy efficient refrigeration technology – the fundamentals*.

Energy efficient design is not rocket science: it is achieved by applying existing technology sensibly.

MY CUSTOMERS WON'T WANT TO PAY EXTRA!

They will if they know why

An energy efficient system does not always cost more to buy and the savings can be hard to ignore. The lifetime costs of an energy efficient plant are significantly lower than those for most standard designs. This Guide shows you how to make a persuasive case for any higher capital cost.

There are three Guides in this series that are aimed at plant owners/users – get free copies for customers as part of your enhanced service. They will help to convince them of the benefits of efficient systems. GPG 277 *Saving money with refrigerated appliances – a guide for smaller retailers, pubs, clubs, hotels and restaurants* is aimed at users of integral 'plug-in' refrigerated appliances. GPG 278 *Purchasing efficient refrigeration – the value for money option* is aimed at end users buying new plant or upgrading existing systems and GPG 279 *Running refrigeration plant efficiently – a cost-saving guide for owners* is aimed at end users with existing plant.

Energy is the single largest component of refrigeration lifetime costs.

Improve efficiency and both users and suppliers will gain.

THROUGHOUT THIS GUIDE:

A triangle is used to show information that is very important to energy efficient design.

A signpost is used to show you where you can find more detailed information about a topic, elsewhere in this Guide or in another publication.

1

PUTTING THE CASE FOR DESIGN OF ENERGY EFFICIENT PLANT

The case for energy efficient design can easily be presented in terms of higher profits. Accountants generally set the budget for the plant engineer, and will always demand financial proof to justify any extra costs for energy efficient plant. The following points will help you, the design engineer, to persuade your customers of the benefits.

An energy efficient plant:

- is generally **more reliable** – this will cut the cost of emergency call-outs, plant downtime and associated lost production;
- has relatively **low maintenance costs** due to its comfortable operating conditions and appropriate maintenance will keep it efficient;
- will tend to **last longer**;
- has relatively **low operating costs** – 20% to 50% less to do the same job;
- enhances a company's 'green' credentials from lower global warming impact, and can contribute towards the requirements of ISO 14001.

The move to more efficient refrigeration plant is going to happen because of increased pressure from your customers, the general public, Government and other groups for more environmentally friendly refrigeration. Don't get left behind, especially as many of the efficiency measures in this Guide cost little or nothing.

Work to build a good relationship with your customer. Presenting efficient designs in a professional manner will gain your organisation a marketing advantage. Indeed, failure to present efficiency measures will soon put you at a disadvantage. Another Guide, GPG 236 *Refrigeration efficiency investment: putting together a persuasive case*, will help your customer or the plant engineer to persuade budget-holders to invest.

See GPG 236 *Refrigeration efficiency investment: putting together a persuasive case*

We all have a responsibility to protect our environment. This Guide shows you how to contribute in a positive and effective way, and still provide the best service and refrigeration plant.

2

CALCULATING AND COMPARING RUNNING COSTS

A refrigeration plant can cost six times as much to run during its lifetime, as it did to buy. It is, therefore, crucial to explain running costs to customers to save them money in the longer term.

It is important to calculate the running costs of systems as accurately as possible, especially where different options are compared. To do this you need:

- the load profile – the load is unlikely to be constant and will vary with ambient temperature or product throughput;
- the ambient temperature profile – remember that for over half the year the ambient temperature in the UK is less than 10°C.

The load profile can be calculated from information supplied by your customer. For storage applications this should include:

- ambient temperature outside the store (this may be outside ambient, or a warehouse temperature depending on the store location);
- usage of the store, including frequency of door opening and number of operatives and trucks inside the store;
- loading and unloading of product, and the temperature and quantity of product loaded into the store;
- operation of auxiliary loads within the store, such as defrosts and fan motors.

There is a blank load profile form in Appendix A that can be used as a basis for calculating lifetime costs.

Example 1

The total heat load changes with ambient temperature for a 10 m x 7.5 m x 2.5 m high, heavily used freezer store at –20°C are shown in Figure 1. The product enters at the storage temperature.

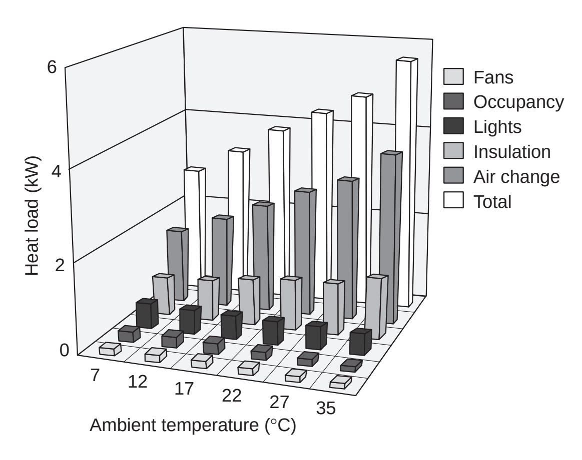

Figure 1 Breakdown of total heat load for various ambient temperatures

Figure 1 shows how the size and proportions of the load components vary. The most important, and very weather-dependent, contributors are the air change load and the heat gains through walls, ceiling and floor. The lighting, fan and occupancy loads are constant and less significant across the temperature zones.

Table 1 below shows:

- The **ambient temperature** variation through the year. For a significant part of the year, the temperature is below 10°C, so it is important to know what the system does at low ambient temperatures.
- The **heat load** at each ambient temperature.
- The **compressor capacity, power input and COP**[1] for each ambient temperature. The compressor is sized for the maximum load at the maximum ambient temperature. Capacity and COP rise as the ambient reduces, so the compressor is significantly over-sized for most of the year. The condensing temperature has been assumed to float with ambient, and is 13K above the ambient temperature.
- The amount of time the compressor will run, both in terms of hours per day and percentage.
- The power input for the year at each ambient temperature, i.e.:

Number of days per year x Compressor power x Running hours per day

Table 1 Calculating annual compressor power consumption according to hours operating at/near various ambient temperatures

Ambient temperature (°C)	**7**	**12**	**17**	**22**	**27**	**35**
Hours per year	2,320	2,570	2,570	1,016	246	38
Heat load (kW)	2.82	3.33	3.89	4.39	4.84	5.68
Comp capacity (kW)	12.00	10.87	9.76	8.68	7.65	6.16
Comp power (kW)	4.32	4.51	4.66	4.79	4.90	5.05
Comp COP	2.78	2.41	2.09	1.81	1.56	1.22
Comp running hours per day	5.6	7.4	9.6	12.1	15.2	22.1
Comp running time %	23.5	30.6	39.8	50.6	63.3	92.2
Power input per year (kWh)	2,355	3,574	4,767	2,462	763	177

Therefore, total compressor power input per year = 14,098 kWh

The condenser fan motor (400 W) runs when the compressor does, i.e. for about 3,042 hours per year, and uses 1,217 kWh of electricity. The evaporator fans run constantly, i.e. 8,760 hours per year, with the two 90 W motors consuming 1,577 kWh per year.

The running costs can be calculated from the electricity price. At 6p/kWh, **running costs amount to £845 per year for the compressor power input alone, and £73 per year for the condenser fan and £94 per year for the evaporator fans. Total running costs amount to over £1,000 per year.**

[1] Coefficient of Performance – for a full definition see GPG 280 *Energy efficient refrigeration technology – the fundamentals*.

For a processing application, the load profile will be dependent on the process. The process requirements will need to be carefully analysed in relation to quantity of heat to be removed, duration of the load and, significantly, the process temperature required.

Where heat loads can be handled at different times, a system need only be sized to match the highest simultaneous combination of loads. Thermal storage could be used to allow a short, high peak load to be handled by a system of smaller capacity. In addition to thermal storage as ice, alternative eutectic storage media are now available over a range of temperatures.

Where process cooling is required to different temperatures, ensure that the heat loads are handled at the highest compressor suction pressure available, and hence at reduced temperature lift and increased efficiency. In larger systems it is often better to install separate systems when load temperatures are different by more than about 5°C.

Example 2

In an ice cream factory, initial ice cream freezing is done in scraped surface heat exchangers with a suction pressure equivalent to –35°C. Final hardening is done in blast freezing tunnels on a –45°C suction. **Using separate compressors for each duty avoids all of the load being handled less efficiently on the –45°C system**. Chilled water, for use in the pasteurising process, goes directly to the high stage compressors of the two-stage system and hence is handled at a suction of around –10°C or higher.

2.1 Ensuring comparisons are accurate

It is important to ensure that comparisons are fair and accurate. The following variables make a significant difference and can point you in the wrong direction if not taken into consideration in your calculations.

- **Load profile.** This usually changes throughout the year with ambient temperature and process conditions. When the load varies significantly, the choice of compressor type, number and configuration is wider and there is more probability of a poor or inappropriate choice being made.
- **Ambient temperature.** Lower ambient temperature often reduces the load, and will always increase system capacity and efficiency where floating head pressure is used.
- **Compressor rating conditions.** These vary for different manufacturers, and are always different than for actual systems.
- **Compressor performance at part load.** This will vary depending on the compressor type and its capacity.
- **Suction return temperature and liquid sub-cooling.** This will vary with ambient temperature and should not be assumed to be the same at all load/ambient conditions.

All **power users,** such as fan motors, pumps and open compressor drives, should be included in the calculations, and bear in mind they may be different under different equipment options.

It is especially important that comparisons are made at the most common operating conditions for the particular system, as well as at the design point.

2.2 Use of computer programs

Computer programs can speed up the selection of equipment and the calculation of running costs, and result in a more comprehensive and accurate selection process. More options can be compared, and each calculation should be more accurate because it is easier to ensure capacity and power are calculated at the correct operating levels.

Beware when using computers. You need to feed in sensible data, otherwise the solutions provided are meaningless.

2.3 Calculating payback

See GPG 236 *Refrigeration efficiency investment: putting together a persuasive case*

Payback is often needed to persuade a customer of the benefits of using a technology to reduce running costs. A simple payback is calculated as follows:

$$\text{Payback in years} = \frac{\text{Additional investment}}{\text{Savings per year}}$$

Don't forget that the investment will continue to make savings over all its working life, so life cycle costs may be well worthwhile presenting as well as, or instead of, the payback period.

Some companies require a more complex payback to be calculated, for example, taking into account bank interest charges. More information on this is given in GPG 236 *Refrigeration efficiency investment: putting together a persuasive case*.

2.4 Total Equivalent Warming Impact (TEWI)

Refrigeration systems contribute twice to global warming:

- through leakage of refrigerants which are greenhouse gases – the direct effect;
- through use of electricity which is generated by burning fossil fuels – the indirect effect.

The combination of the direct and indirect effect is called the Total Equivalent Warming Impact. This is calculated taking into account:

- expected leakage over the lifetime of the system;
- global warming potential of the refrigerant;
- energy consumption of the system over its lifetime.

See Appendix C for BRA contact details

The factors above will vary depending on the system type and design and, therefore, the TEWI for each system must be calculated individually. An example calculation is given in Appendix B. Full guidelines on calculating TEWI are published by the British Refrigeration Association.

TEWI is reduced by:

- selecting a refrigerant which has a low global warming potential (i.e. ammonia or hydrocarbons);
- designing a system which contains the refrigerant as well as possible;
- designing an energy efficient system.

2.5 Reliability and maintenance

Energy efficient systems operate with lower temperature lifts (or compression ratios) which bring significant benefits to plant reliability. The lower the compression ratio across the compressor, the lower its discharge temperature.

For example, if an R404A system frozen food store has a dirty condenser which increases its condensing temperature by 10°C (not unusual), the compressor discharge temperature rises by about 12°C. The higher the discharge temperature the more likely the lubricant is to break down, causing bearing wear and possible failure. In addition, the compressor will have to run nearly 20% longer to produce the required cooling effect.

Conditions which produce the greatest energy efficiency are always easier on the compressor, such as:

- low temperature lift;
- low suction superheat;
- good ventilation and, therefore, the coolest air possible around the compressor.

See Section 13 for more details on maintenance.

3

THE DESIGNER’S EFFICIENCY TOOLKIT

The plant designer controls several major factors that affect the efficiency of a system. You may not need to consider them all, so refer to Figure 2 (page 9) to see which are the most beneficial for the type of system(s) you design. Each of the factors in the Designer’s Efficiency Toolkit is summarised below and discussed in more detail in the Section indicated.

For more information on basic refrigeration technology, see GPG 280 *Energy efficient refrigeration technology – the fundamentals*

Section 4 Load level

This can often be reduced, e.g. by specifying thicker insulation for the structure of cold stores and introducing a door management system to cut open-time.

Section 5 Plant sizing

Many systems are over-sized due to uncertainty of load and components’ capacity data. Over-sizing leads to higher than necessary capital and running costs. It pays to find and use accurate information.

Section 6 Overall system design

Large systems can be single or two-stage, maybe with economisers. Medium-to-large systems with varying loads allow several options for compressor number, sizing and control. Matching the system to the load brings significant efficiency gains.

Section 7 Temperature lift, evaporators and condensers

The ‘lift’ is the difference between the evaporating and condensing temperature and is probably the single most important factor for a system designer. Each 1°C reduction in lift gives a 2% to 4% cut in energy consumption, and increases capacity.

Section 8 Defrost

This is an important balancing act to get right. A well designed defrost system can use approximately 5% of the total energy consumption. However, a poorly designed defrost system will use significantly more energy both directly (to defrost evaporators) and indirectly. Too much defrost dumps heat into the cooled space/fluid, too little means ice build-up and reduced efficiency.

Section 9 Avoiding head pressure control

This is the key to perhaps the most significant area of running cost savings. Condensing temperatures should ideally be allowed to float downwards in cool ambient conditions, which means lower temperature lift for much of the year.

Section 10 Compressor selection and control

There are different types of compressor available and selecting the wrong one has an adverse effect on efficiency. It is important you understand manufacturer’s selection data and select the correct type and size(s).

Section 11 Refrigerants

Here options can be bewildering. For every type and size of system there are several refrigerants that would be appropriate, but their efficiencies in a particular system often vary.

Section 12 Pipework

Pipework design can play an important role in overall efficiency, especially on large and remote systems. It is important to get sizing and routing right while meeting other requirements such as good oil return.

Section 13 Design for monitoring and maintenance

All the good work carried out in achieving an energy efficient design can be wasted if the plant is subsequently poorly maintained, and good plant maintenance is helped by monitoring system operation.

Section 14 Other components

These also have an impact on energy efficiency, so it is important to understand their effect and when to use them. For example, electronic expansion valves can significantly reduce energy consumption through floating head pressures.

Section 15 Installation, commissioning and maintenance

This should be specified by the design engineer to ensure the system works as well as it should throughout its life.

Figure 2 may help to prioritise which issues to look at, especially if time is limited. The order is not random – those at the top usually give the best savings, so start there. Each system is different, however, so address as many as possible to improve efficiency. The other issues should be considered as time allows, but can also be important to optimising efficiency.

Energy efficiency design priorities

Cold storage – small systems (under 100 kW installed motor size)	**Cold storage – large systems** (over 100 kW installed motor size)	**Process cooling**
Reduce temperature lift e.g. larger condenser and evaporators and good control strategy See Section 7 (page 20) and Examples 9, 11, 13, 14, 16	**Reduce temperature lift** e.g. larger condenser and evaporators and good control strategy See Section 7 (page 20) and Examples 9, 11, 14, 16	**Reduce temperature lift** e.g. larger condenser and evaporators and good control strategy See Section 7 (page 20) and Examples 14, 16
Load reduction e.g. reducing auxiliary use such as fans and pumps, door management, strip curtains See Section 4 (page 10) and Examples 1, 2, 3, 4, 5, 6, 10, 21	**Load reduction** e.g. reducing auxiliary use such as fans and pumps, door management, strip curtains See Section 4 (page 10) and Examples 1, 2, 3, 5, 10, 21	**Load reduction** e.g. free cooling for part of load See Section 4 (page 10) and Example 7
Compressor size/type e.g. reciprocating vs scroll, hermetic vs semi-hermetic See Section 10 (page 28) and Examples 17, 19	**Compressor size/type** e.g. reciprocating vs screw, avoidance of operation at part load, good control See Section 10 (page 28) and Examples 17, 18, 19, 20	**Compressor size/type** e.g. reciprocating vs screw, avoidance of operation at part load, good control See Section 10 (page 28) and Examples 17, 18, 19, 20
Refrigerant e.g. HFC and HC alternatives, good refrigerant containment See Section 11 (page 36) and Example 18	**Overall system design** e.g. one-stage vs two-stage See Section 6 (page 16) and Examples 2, 8, 9, 15	**Overall system design** e.g. one-stage vs two-stage See Section 6 (page 16) and Examples 2, 7, 8
Overall system design e.g. maximise sub-cooling, minimise non-useful superheat, good defrosting See Section 6 (page 16) and Examples 2, 9, 12, 15	**Pipework** e.g. avoid long runs, optimise pipe diameter for oil return and low pressure drop See Section 12 (page 39)	**Pipework** e.g. avoid long runs, optimise pipe diameter for oil return and low pressure drop See Section 12 (page 39)
Pipework e.g. avoid long runs, optimise pipe diameter for oil return and low pressure drop See Section 12 (page 39)	**Refrigerant** e.g. HFC alternatives, good refrigerant containment See Section 11 (page 36) and Example 18	**Refrigerant** e.g. ammonia vs R22, good refrigerant containment See Section 11 (page 36) and Example 18

kWh and £££ savings

Figure 2 The top priorities for design of energy efficient plant

4

LOAD LEVEL

See also the load profile and energy running cost form in Appendix A

This Section is split into two parts, in line with the usual design procedure:

- firstly, assessing the load correctly initially;
- secondly, minimising the load on the system to reduce the required capacity and running costs, for cold storage applications and then process plant.

4.1 Assessing the load

In order to assess the load correctly, accurate information is required from your customer regarding the amount of product to be cooled in a given time, and the temperature of the product before and after cooling. Future requirements also need to be determined, for example, the usage of a cold store often increases.

Vague specifications from a customer can lead to serious over- or under-estimates of the cooling load, resulting in a system design that will either not do the job required or perform inefficiently.

Often, part of the load can be met by ‘free’ cooling. For example, when cooling food on a production line, ambient air can be used initially to reduce the temperature to, or close to, the required temperature. Water from a bore hole, river or cooling tower can also be used as the first stage of process cooling. Do work out water treatment costs, as these may alter the cost-effectiveness of this ‘free’ cooling.

It is vital to build a good relationship with your customers. By understanding their business as well as their refrigeration needs, you will be able to design a plant which matches their requirements more accurately.

4.2 Minimising the load on the system: cold storage

The actual load on the system will usually be more than the load associated with the product. In the case of cold storage, product load is often minimal. Typical heat loads are shown in Figure 3:

- heat gain through walls (20%);
- air change load (30%);
- evaporator fans (15%);
- lighting (10%);
- defrost (15%).

All the contributing heat loads can be minimised.

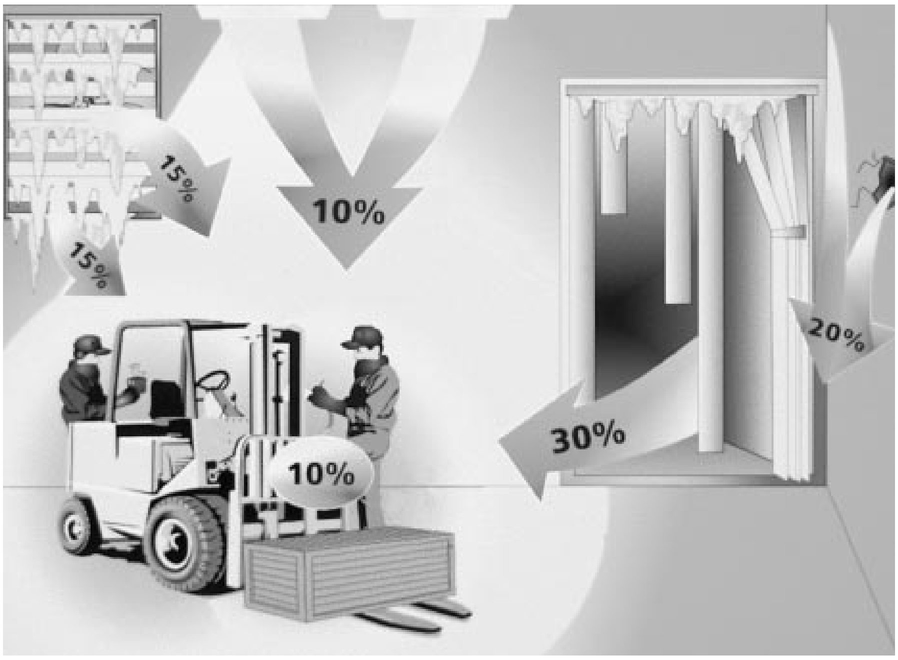

Figure 3 Heat loads on a cold store as a percentage of the total heat load

Heat gain through walls, ceiling and floor

Approximately 20% of the heat load on a cold store is that gained through the walls, ceiling and floor. Thicker, good quality insulation reduces this heat gain and is especially important in frozen food stores (Example 3, page 12).

Air change load

The air change load accounts for about 30% of the total heat load. It can be significantly reduced (Example 4, page 12) by:

- installing strip curtains or an air lock (good design, installation and maintenance of these is essential to ensure ice does not build up, causing a hazard and increasing heat gains);
- using door management, e.g. a self-closing door.

See GPCS 350 *Strip curtains on chilled display cabinets*

Evaporator fans

Evaporator fans represent about 15% of the total heat load, but they contribute to the electricity cost twice. They directly use power, of which a high percentage is converted to heat that must be removed by the refrigeration system. Lower power fans are now available for most applications, often at no extra cost (Example 5, page 12).

Larger diameter fans operate at lower speeds and, therefore, use less power. They are also quieter. For example, a 1,000 mm diameter fan with the same air flow as a 750 mm one, uses up to 60% less power.

An additional saving can be achieved by switching fans off when they are not needed (Example 6, page 12).

Lighting

Lighting accounts for about 10% of the total heat load and, again, you pay for this twice. The most efficient form of lighting is already used in most applications, but savings can be made by ensuring lights are switched off when not in use for long periods.

Defrosting

Defrosting should not have to increase the heat load on a cold store, but often does, and this depends on the method used and its control. A defrost termination thermostat, which terminates the defrost as soon as all the ice has melted, ensures that heat does not enter the store (again, you would be paying for this twice).

See Section 8 *Defrost*

Example 3

A frozen food store 10 m x 7.5 m x 2.5 m high at –18°C for 30 tonnes of wrapped frozen food is sited inside a warehouse with a constant ambient temperature of 18°C. **A 5% reduction in the overall heat load can be achieved by using 200 mm of polyurethane foam, instead of the usual 150 mm**, for the walls, ceiling and floor. The savings will be lower for higher temperature cold stores, and probably not worth considering for high temperature applications such as beer cellars.

Example 4

One PVC strip curtain manufacturer monitored the temperature of a small freezer store with and without a curtain. **Every time the door opened the store air temperature increased from –19°C to –2°C without a curtain, but only to –16°C with a curtain in place.** The time taken to reduce the store temperature back to –19°C was 12 minutes without the strips, and 3 minutes with (Figure 4).

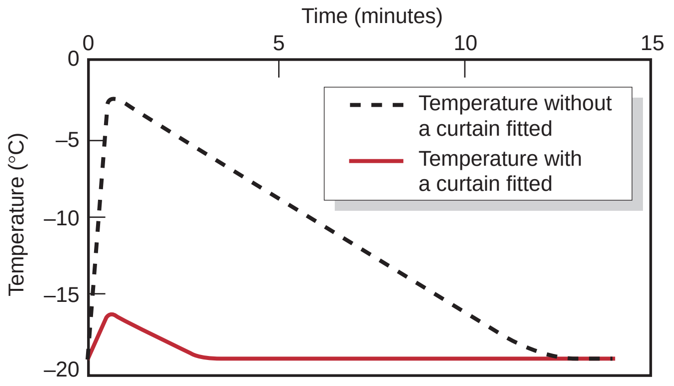

Figure 4 Graph of cold store temperature after door opening, with and without PVC strip curtains

Example 5

By swopping to lower power fans on a cold store at –22°C, the motor power input was reduced from 0.8 kW to 0.456 kW. The lower heat transfer of the evaporator reduced the capacity of the system by 5% but, because of the lower heat gain into the store from the new fans, the heat load on the system was also reduced.

Based on a 5p/kWh electricity cost, the changes reduced annual running costs by £115 per unit. The unit price increased by around £190, giving a payback of less than 20 months. **Thus, lower power fans can be very cost-effective.**

Example 6

Most cold stores and beer cellars run evaporator fans constantly to ensure even temperature distribution throughout the store. Where multi-fan evaporators are used, all fans need not be kept running. **One evaporator manufacturer has calculated that, for beer cellars using two- or three-fan evaporators, running only one fan per evaporator would cut running costs by up to 14%.** This requires a simple modification to the control circuit. One brewery running 2,500 pubs with these cellars is saving over £175,000 per year.

NB If you apply this saving, ensure that there is sufficient air flow in the store to give even temperature distribution when the system is off. You cannot turn off all the fans.

4.3 Minimising the load on the system: process plant

There is also scope for reducing heat loads in process cooling, such as:

- less energy intensive methods of cooling;
- using heat exchange;
- reducing extraneous loads.

The most significant reductions are possible in the field of high temperature process cooling. For example, mechanical refrigeration is often applied to cool products which have just left ovens or fryers, when a substantial part of the cooling load could be handled by alternative coolants such as filtered ambient air. Some processes afford the opportunity for ‘free’ cooling where the temperature of the hot product is such that it can be cooled by refrigerant (e.g. ammonia) evaporating within the air cooler at a pressure which enables the gas generated to be condensed without the need for a compressor.

Always investigate the use of less energy-intensive methods of cooling to handle the cooling load at least in part, if not in its entirety. The use of mains, bore-hole, river or tower water can often significantly reduce the required size of the refrigeration plant and consequently the power required.

Also consider the use of heat exchange. For many years it has been standard practice in the milk industry to use ‘regenerative’ cooling in the pasteurising process. Incoming milk is initially heated by heat exchange with the milk leaving the hot section of the pasteuriser, the latter thus being cooled significantly before reaching tower water and chilled water cooling sections (also see Example 7).

See GPG 89
Guide to compact heat exchangers

Carry out a careful analysis of the process and the temperatures before finalising the mechanical refrigeration load.

Extraneous heat loads must also be considered and kept to a minimum.

- Site refrigeration plant close to the process served, with pipework insulation of optimum thickness, to minimise transmission heat gains.
- Use short pipe runs to reduce energy consumption of secondary refrigerant circulation pumps and thus the heat generated by them.
- Run circulation pumps, air fans, etc. only when the process requires.
- Use variable speed drives on pump motors on fluctuating flow systems. The cost of the energy saved will exceed their additional capital cost within a short period (Example 8, page 14).

Remember to consider the areas through which pipes carrying low temperature fluids pass. A liquid line routed through a boiler house may provide the shortest run but also the maximum heat pick-up.

Example 7

An instant coffee producing plant used incoming cold water as a raw material which was heated under pressure to 180°C and then passed through a percolating process. The resulting coffee liquor was cooled from 30°C to 20°C, first using cooling tower water and then a water chiller. The flow rate of the outgoing liquor and the cold feed were very similar and the streams were located close together.

This system was replaced by a plate heat exchanger that simultaneously heated the incoming water to 70°C and cooled the coffee liquor to 20°C. **This use of heat exchange saved both heating and chilling costs, giving a payback of eight months, and also improved control of the system.**

See GPCS 124
Variable speed drives on secondary refrigeration pumps

Example 8

Variable speed drives were fitted to glycol distribution pumps in a brewery to deliver cold glycol at –7°C to various process operations, the main demand for cooling being from large fermentation vessels.

Heat released into the beer during fermentation has to be removed, and towards the end of the process the beer is 'crash cooled' from around 20°C to nearer 0°C. The cooling demand on the refrigeration system is peaky and unpredictable due to there being 20 vessels involved. When a vessel requires cooling, valves open to allow glycol to pass through a jacket and then close when the set-point is reached.

The glycol pumps have to deliver widely varying flows while maintaining the minimum delivery pressure to achieve the desired cooling rate. **Fitting variable speed drives to the pumps saved this brewery £25,000 a year from an investment of £40,000.** The pump speeds were varied so as to maintain a fixed delivery pressure whatever the load.

5

PLANT SIZING

It is often tempting to over-size plant. For example:

- you may be unsure about the accuracy of the load calculation because your customer's information is vague or the raw data are unavailable;
- you know that the data presented by some equipment suppliers are 'optimistic'.

It has, therefore, become common practice to add a safety/commercial factor. Typical over-sizing rules of thumb include adding a 10% capacity margin, sizing plant to meet a 24 hour load in 18 hours at the maximum load and ambient temperature condition, or both.

However, over-sizing is usually unnecessary to meet demand, and also results in the plant cycling more frequently and running less efficiently, reducing plant reliability.

Don't be tempted to over-size plant – it costs more to buy and run.

To avoid having to over-size plant:

- get accurate information from your customer regarding the usage of the plant – it may help to agree it in writing;
- use accurate raw data for calculating the load – most of the information you need is available in the ASHRAE Handbook;
- specify components from manufacturers whose data are certified (i.e. the components have been independently tested and the performance data checked for accuracy).

See Appendices C and D for ASHRAE details

6 OVERALL SYSTEM DESIGN

There are choices to be made with system configuration when the load is larger and/or variable. This section covers:

- single and multi-stage systems;
- absorption systems and cases where these may be beneficial;
- other system considerations, such as suction and liquid temperature and heat recovery.

Multi-compressor systems are covered in Section 10 *Compressor selection and control.*

6.1 System configuration – single-stage, compounded, cascade system

The system configuration most suited to a particular cooling requirement is usually determined by the pressure ratio (condensing pressure/evaporating pressure) which the compressor must achieve. For reciprocating compressors, if the compression ratio exceeds about 10:1, the re-expansion of compressed gas remaining in the cylinder at the end of compression (clearance volume) displaces so much of the cylinder's capacity for incoming suction gas that the compressor capacity falls off markedly. This results in very high discharge temperatures which cause breakdown of the lubricating oil.

Coping with high pressure ratios

Two-stage compression (Fig 5) can cope with high pressure ratios. It avoids both ineffective compression and high discharge temperatures. The gas is cooled to the saturation temperature corresponding to the intermediate pressure by passage through a pool of liquid in an intercooler vessel, or close to saturation temperature by liquid injection. The intercooler, or alternative heat exchanger, can also be used to sub-cool liquid refrigerant *en route* from condenser to expansion valve and evaporator, effectively increasing the capacity of the evaporator and improving COP.

As long as the screw compressor is designed for this, it can work effectively at much higher pressure ratios than reciprocating machines, due to minimal clearance volume at the end of compression. High discharge temperatures are avoided by cooling via the large amount of oil injected into the screw machine. The downside of this is the requirement for significant oil cooling. (Thermo-syphon cooling is more efficient than other forms of oil cooling.) Energy savings can still be made by operating screw compressors in tandem as a two-stage compression system, the savings primarily arising from the sub-cooling of the liquid flowing to the evaporators.

NB: Improvement in screw compressor COP can also be realised by taking advantage of an 'economiser' or intermediate pressure port, which allows input of gas part-way through the compression process. Again, this is used to sub-cool liquid flowing to evaporators and allows an increase in compressor refrigeration capacity in excess of the increase in compressor power consumption.

GPG 280 *Energy efficient refrigeration technology – the fundamentals* contains more information on cascade and two-stage systems.

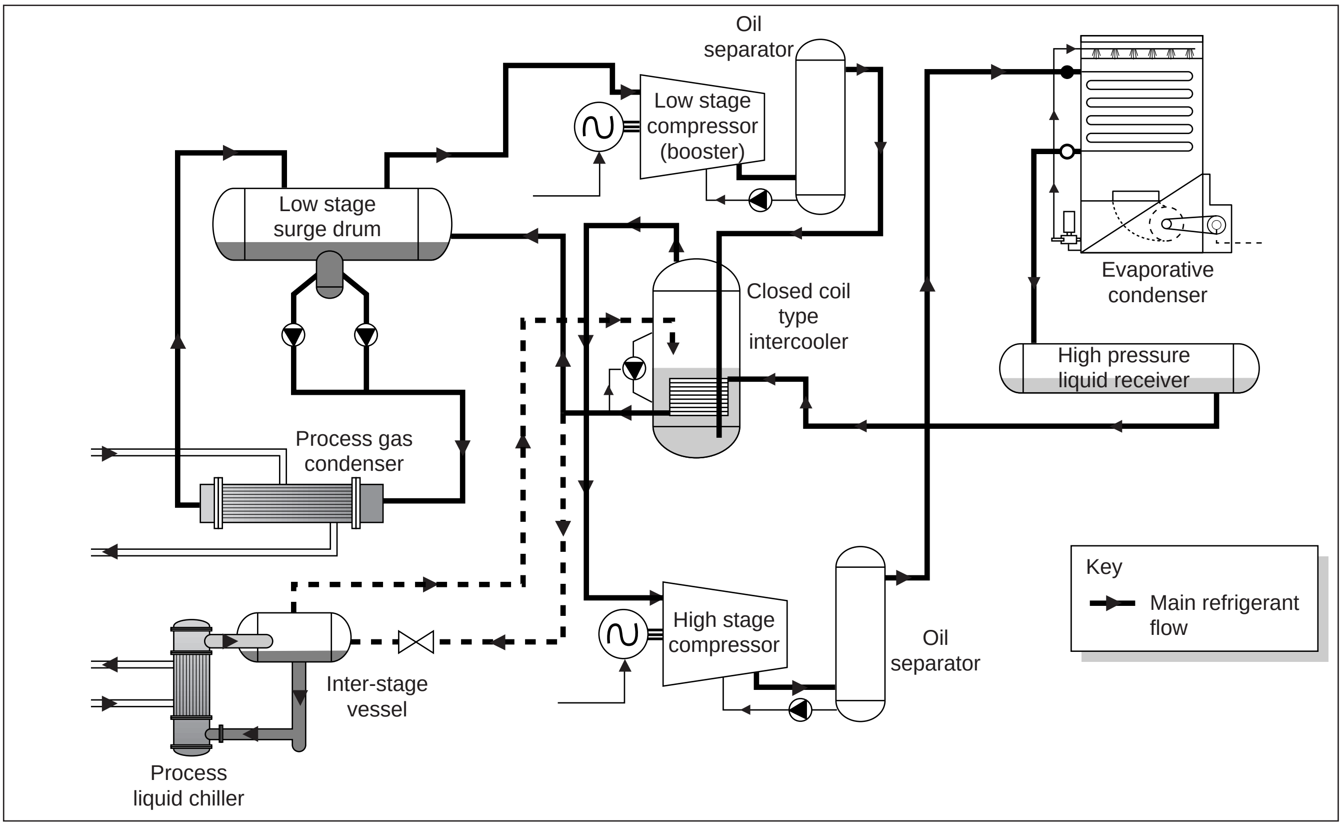

Figure 5 Schematic diagram of a two-stage system

Coping with very low temperatures

On low temperature applications with an evaporating temperature below about –50°C, cascade systems are the best option for many reasons including energy efficiency. The low pressure (or first stage) refrigerant has the necessary low saturation temperature (or boiling point) at a reasonable suction pressure. The condenser of the first stage is a heat exchanger acting as the evaporator for the second stage, which operates with a conventional refrigerant and hence has an acceptably low discharge pressure at the normal condensing temperature. A typical application suited to a cascade system could be ethylene liquefaction, or cooling for an aircraft windscreen test facility where cooling to –60°C is typical.

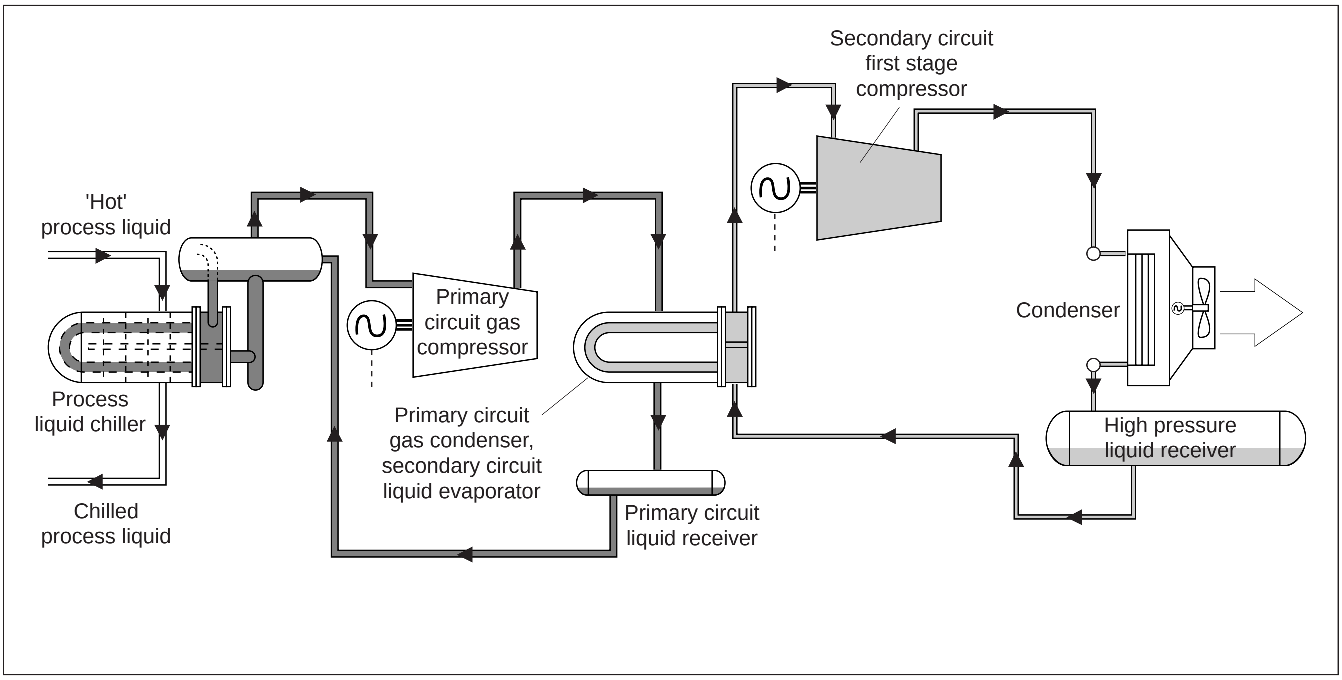

Figure 6 Schematic diagram of a cascade system

6.2 Absorption systems

See GPG 256 *An introduction to absorption cooling*

Absorption cooling offers better environmental performance and cost savings in particular circumstances, such as where your customer:

- is considering CHP (Combined Heat and Power);
- has a CHP unit, but cannot use all of the available heat;
- has waste heat available;
- has a low-cost source of fuel available (e.g. landfill gas);
- has a boiler which has a low efficiency due to a poor load factor (this is especially likely in summer).

Absorption cooling is also useful if the site is particularly sensitive to noise or vibration, or has an electrical load limitation that is expensive to overcome, but also has an adequate supply of heat.

Although the COPs for absorption systems appear poor compared with more conventional systems, in the above circumstances, the heat supply will be low cost and performance is good. Reliability has also improved in recent years.

It is common to integrate absorption systems for base load cooling with compression systems for peak and/or back-up cooling. Take care, however, to ensure that the absorption chillers do not see undue fluctuations in ‘flow’ and ‘return’ temperatures. Absorption plant should, therefore, usually be upstream from compression plant. More guidance is contained in GPG 256 *An introduction to absorption cooling*.

6.3 Sub-cooling

See GPG 280 *Energy efficient refrigeration technology – the fundamentals* for a definition of sub-cooling

Sub-cooling the liquid refrigerant before it enters the expansion device increases capacity without increasing power consumption. Sub-cooling is, therefore, very important to energy efficiency.

On a single-stage refrigeration cycle, sub-cooling of the liquid refrigerant prior to the expansion valve is achieved in:

- the condenser, as the removal of heat from the refrigerant continues after condensation;
- the receiver, when heat is lost to the surroundings as ambient temperature is below condensing temperature;
- the liquid line upstream of the expansion valve when passing through areas cooler than the liquid line temperature.

Sub-cooling within the condenser is achieved by providing additional heat transfer surface in the form of a sub-cooling coil. The refrigerant from the receiver is passed through the sub-cooling coil before entering the liquid line. This is especially recommended when the liquid line is long and/or rises vertically for some distance between receiver and expansion valve.

‘Free’ sub-cooling can be obtained by careful siting of the receiver – not in a hot plant room, not in direct sunlight and so on – and ensuring that the liquid line is arranged to avoid locations giving rise to heat gain.

A further method of achieving sub-cooling in the liquid line is to use a liquid-to-suction heat exchanger. Liquid is cooled by transferring heat to the cold suction gas leaving the evaporator. The effect on system capacity and energy consumption, however, depends on the refrigerant, as the liquid sub-cooling results in warming of the gas in the suction line, decreasing the refrigerant density and reducing the mass flow rate of the compressor.

On two-stage systems, the inter-stage pressure refrigerant should always be used to sub-cool the liquid from the receiver:

- generally to around 5°C above the inter-stage saturation temperature in a closed coil or heat exchanger;
- fully down to saturation temperature in an ‘open flash’ intercooler.

6.4 Superheat

See GPG 280 *Energy efficient refrigeration technology – the fundamentals* for a fuller explanation of superheat

Superheat is the increase in temperature of refrigerant gas above the evaporating temperature. The higher the suction gas superheat, the lower the gas density and, therefore, the lower the compressor mass flow rate. This reduces the compressor capacity without reducing its power consumption, increasing running costs.

Superheat can be classed as:

- useful – where the heat gained by the gas is picked up while doing useful cooling, e.g. in the final passes of a 'dry expansion' cooler with a thermostatic expansion valve;
- non-useful – e.g. in suction piping outside the space to be cooled, en route to the compressor where, if that heat is of no use, it is wasting energy.

The advent of electronically-controlled expansion valves has allowed lower superheat of gas leaving evaporators to be routinely achieved without risking liquid being carried back to the compressor in the suction gas. The latter was a problem with conventional mechanically-controlled or 'thermostatic' expansion valves.

Compressor refrigeration duty and system energy efficiency can be increased by using electronic expansion valves.

Efficiency is improved by both:

- the lower superheat keeping gas density up;
- allowing liquid wetting within the evaporator over more of the tubing length;

thereby increasing overall heat transfer and raising suction pressure.

Always insulate suction lines to prevent non-useful superheating.

6.5 Heat recovery from refrigeration systems

See Section 2.6 in GPG 280 *Energy efficient refrigeration technology – the fundamentals*

See GPG 141 *Waste heat recovery in the process industries*

The condensing refrigerant in a refrigeration system is warmer than ambient temperature. The amount of heat rejected in the condenser is the cooling effect plus most of the compressor input power. Heat can be recovered from:

- The discharge vapour, which can be as hot as 150°C. The heat is removed in a de-superheating vessel between the compressor and condenser. The amount of heat available is, however, relatively small.
- The condenser, which is normally 10 to 30°C above ambient temperature.
- The oil used on oil-cooled compressors, which can be between 60 and 80°C.

It is essential that the effect of heat recovery on the performance of the refrigeration system is carefully analysed. In many cases, recovering useful heat from the condenser forces the system to operate less efficiently (i.e. at a higher condensing temperature), and the savings in heating costs are usually less than the added refrigeration costs.

A ***de-superheater*** recovers high temperature heat from the discharge vapour leaving the compressor. The discharge temperature depends on the operating conditions of the system and the refrigerant. R22 and ammonia operate with significantly higher discharge temperatures than most other refrigerants.

In a well designed refrigeration system, the condensing temperature should be as low as possible. Any heat recovered from the ***condenser*** will be at a very low temperature and is very rarely useful.

In oil flooded screw compressors much of the motor heat is dissipated into the ***lubricating oil***. The oil usually enters the compressor at about 40°C and leaves it at 60 to 80°C. For an R22 system, about 38% of the motor power would be absorbed by the oil and consequently be available for recovery. For ammonia systems, this figure increases to about 60%.

Heat recovered from discharge gas or compressor lubricant (or, to a lesser extent, a water-cooled condenser) could be used, for example, to pre-heat water for a boiler. The heat recovered from an air-cooled condenser could be used to provide background heating, e.g. in a dry goods warehouse.

7

TEMPERATURE LIFT, EVAPORATORS AND CONDENSERS

See GPG 280 *Energy efficient refrigeration technology – the fundamentals*

Every system should be designed so that the difference between the evaporating temperature and the condensing temperature is as low as possible, thus maximising capacity and efficiency. A reduction in temperature lift of 1°C reduces running costs by 2% to 4%. Selecting large condensers and evaporators is the simplest way of achieving this design strategy. They obviously add to the capital cost of the system, but this is often partly or totally offset because smaller size compressors can be selected.

7.1 Evaporator selection

The type of evaporator is driven by the application, e.g. water chiller/air cooler, but the size of evaporator is up to the system designer.

A larger evaporator not only increases the evaporating temperature (remember a 1°C rise saves 2% to 4% in running costs), but also reduces:

- frost build up in air coolers;
- the size of compressor needed;
- the compression ratio of the compressor and hence its wear rate.

Example 9

This example compares several evaporator sizes (measured as evaporating surface area) for the same application, and shows the effect on evaporating temperature, running cost and compressor size. As shown, **use of a larger evaporator allows higher evaporating temperature and, hence, results in lower annual running cost.**

Process requirement: Frozen food store, temperature –20°C to –22°C
Cooling load 100 kW
Single-stage plant
8,000 hours/year operation
Electricity cost 4p per kWh
Assume fixed condensing temperature of 30°C

Evap temp (°C)	COP	Evap surface (m^2)	Evap cost (£)	Annual running cost (£)	Comp size (m^3/s)
–28	2.1	625	7,500	15,168	0.083
–31	2.0	436	6,050	16,320	0.090
–34	1.8	335	5,150	17,600	0.103
–37	1.7	272	4,550	18,880	0.117
–40	1.6	229	4,100	20,288	0.138

These data show a good payback (less than eight months) for even the largest evaporator, without taking into account differences in compressor size. At the highest evaporating temperature, the compressor is 40% smaller than at the lowest. In most cases, the saving in compressor cost will be similar to, or greater than, the increase in evaporator cost.

Example 9 simply shows the effect of different sized evaporators at one condensing temperature, in this case an average condition. To get an accurate picture of the annual running costs you should calculate the operating conditions at a range of predominant ambient temperatures. When the ambient temperature falls, the compressor is able to pump more, so the evaporating temperature falls. This not only affects the COP of the system, but also may affect whether or not frost build up will occur, for example.

See Section 2
Calculating and comparing running costs

Whenever evaporators are selected, the cost of their associated fans and pumps must be taken into account. Larger evaporators usually have more fan/pump power.

Example 10

It is worth considering fan power carefully. Two evaporators run with very similar temperatures at the predominant system load and ambient temperature. One has three fans with a total power of 320 W and the other only one larger fan at 210 W. With the single fan evaporator, the power absorbed by the fan, and the heat input to the cold store, is lower.

Alternatively, Example 6 shows that significant savings can be made by switching off some fans with multi-fan evaporators, when the refrigeration system is not running.

Read evaporator selection data with care, especially when selecting evaporators for use with a refrigerant which is a zeotropic blend (i.e. has temperature glide).

See Section 11
Refrigerants

Selection of the saturation temperature at which the system operates is, therefore, key to achieving an energy efficient design. A further important factor is selection of the level of superheat that is to be used. Superheat is the amount by which the actual temperature of the refrigerant gas is raised over and above its saturation temperature, once evaporation of all the liquid has taken place.

The level of superheat required is affected by the choice of expansion device. Direct expansion systems with thermostatically-controlled expansion valves (TEV) require a certain level of superheat in the evaporator (or at least upstream of the TEV sensing bulb), to provide an effective signal for the TEV to control the flow of liquid refrigerant into the cooler. The normal level of superheat in the gas leaving the evaporator is 5°C and this is the factory setting for most TEVs. Electronically-controlled expansion valves enable small levels of superheat to be safely used.

Any increase in the level of superheat in the evaporator reduces the efficiency of the system. This is because heat transfer is most effective on the ‘wetted’ parts of the evaporator, i.e. those parts in contact with liquid refrigerant which can absorb large amounts of latent heat. Heat transfer across the ‘dry’ surface area of the cooler, in contact with (superheated) refrigerant in the gas phase only, is substantially less. This reduces the capacity of the evaporator which, in turn, causes the evaporation temperature to fall and, for a given condensing temperature, the temperature lift to be increased.

Superheat also causes lower efficiency beyond the evaporator. Increased superheat further reduces the density of the gas entering the compressor. For a given volumetric flow handled by the compressor, this reduces the mass flow through the machine and hence the capacity. To minimise the level of superheat in the gas at the inlet to the compressor, pay particular attention to:

- the size of expansion valve – an undersized valve results in increased superheat;
- liquid line sizing – undersized liquid lines can result in excessive pressure drop, which can cause vapour generation upstream of the expansion valve, compromising valve capacity;
- suction line sizing – undersized suction lines in effect increase superheat by lowering saturation temperature;
- suction line insulation – insulation will keep heat increase to a minimum;
- suction line position – avoid running suction lines through areas of elevated temperature.

Suction line heat exchangers, with the TEV bulb placed downstream, result in a fully-wetted surface within the evaporator, but with the associated penalty of slightly lower suction pressure.

See GPG 280, Section 2, for further information on suction line heat exchangers

See GPG 280 *Energy efficient refrigeration technology – the fundamentals*

7.2 Condenser selection

The choice of condenser is between:

- air-cooled;
- water-cooled;
- evaporative (a mixture of air and water).

A water-cooled condenser takes advantage of the lower wet bulb ambient temperature (especially in the summer). The wet bulb temperature is often up to 5°C lower than the dry bulb temperature, even more in summer. The condensing temperature with water-cooled and evaporative condensers will, therefore, be at least 5°C lower than with air-cooled types in summer.

Some of the associated savings are offset, however, by the additional power needed to drive water pumps and cooling towers. *All the power users associated with a condenser must be taken into account when comparing different types.*

The larger the condenser, the lower the condensing temperature (again, a 1°C lower condensing temperature means 2% to 4% lower running costs). But don't be tempted to specify condensers that are too large! This can cause low pressure problems. As a guide, aim for 10K temperature difference (between condensing temperature and ambient) for air-cooled and evaporative types, and 5K for water-cooled systems.

Example 11

This example compares several condenser sizes (measured as condensing surface area) for the same application. As shown, **use of a larger condenser allows lower condensing temperature and, hence, results in lower annual running costs.**

Process requirement:	Brine chiller
	Cooling load 100 kW
	Cooling water flow at 23°C, return at 26°C
	Ammonia plant
	6,000 hours/year operation
	Electricity cost 4p per kWh
	Assume fixed evaporating temperature of –8°C

Cond temp (°C)	COP	Cond surface (m^2)	Cond cost (£)	Annual running cost (£)	Comp size (m^3/s)
28	4.1	27	5,650	5,855	0.035
31	3.7	14	3,800	6,432	0.035
34	3.4	9	3,000	7,008	0.036
37	3.2	7	2,550	7,632	0.036
40	2.9	6	2,250	8,208	0.037

There is no significant reduction in compressor size with an increase in condenser size (unlike increasing evaporator size). Looking at the payback over a standard sized condenser, the optimum condenser size is probably one giving 30°C:

Cond temp, °C	Payback, years
31	1.4
28	3.2

Remember to include the fan power when making a comparison.

A further aspect of condenser operation which affects the system energy efficiency is the degree of sub-cooling used. Sub-cooling is the degree to which the condensed refrigerant liquid is cooled below the temperature at which it condensed.

Condensers with the refrigerant flowing through the 'Tube Side' of the unit, i.e. evaporative and air-cooled types, can, under certain conditions operate with the bottom one or two rows of tubes fully flooded with refrigerant in the liquid phase. As these tubes are exposed to the cooling medium, sensible heat is drawn from the liquid, lowering its temperature to below its saturation temperature, i.e. sub-cooling it.

The greater the degree of sub-cooling of the liquid refrigerant, the less 'flash gas' is generated in expansion. Sub-cooling occurs initially in the condenser and then from the surfaces of pipework and any receiver, through heat loss to ambient air. With greater sub-cooling, the compressor has to handle less gas, reducing its energy consumption and increasing the energy efficiency of the system.

On single-stage systems, in addition to sub-cooling in the condenser, similar results can be achieved by use of a suction line heat exchanger, where liquid flowing to the expansion valve gives up heat to the suction gas flowing to the compressor. Degrees of sub-cooling achieved in the condenser can exceed 10°C at low ambient temperatures, with slightly lower values achieved in the summer. With a suction line heat exchanger, further cooling by a similar amount can be achieved with, however, an increase in suction gas superheat and suction line pressure loss. The latter increasing temperature lift across the compressor (see Fig 12 on page 39).

On two-stage systems, the liquid is normally sub-cooled in the coil of a closed intercooler to within 5°C of the inter-stage saturation temperature. Thus with liquid leaving the receiver at 30°C, 35°C of sub-cooling could be achieved, with an inter-stage saturation temperature of -10°C.

Many single-stage systems incorporating screw compressors have the option of connecting to ports at intermediate pressure levels along the machine. This enables the compressor to handle additional vapour generated in separate sub-cooling units or economisers more efficiently than would be possible if the same gas had to be handled at the lower suction condition.

7.3 Condensing unit systems

There is scope to improve efficiency even on small systems where you will be selecting condensing units. You usually have no control over the condenser used (the manufacturer decides this), but you do have control over the evaporator. Also, condenser sizing varies between different manufacturers, so it is worth making comparisons even of 'standard' equipment.

Example 12

This example shows how it is worthwhile making comparisons even of standard equipment, and also how influential ambient temperature is on performance.

A retail Group needed to specify condensing units and evaporators for beer cellars. Three standard systems were tested at a cellar temperature of 10°C, in ambient temperatures of 32°C (the design condition) and 15°C (the most prevalent ambient temperature when cooling is required). The refrigerant used was R407C.

	System A	System B	System C
32°C ambient (design)			
Evap temps, °C bubble, dew	–1.3, 4.4	–4.6, 1.2	–4.0, 1.7
Cond temps, °C bubble, dew	44.2, 49.1	47.0, 51.8	46.3, 51.1
COP	2.17	1.92	2.09
15°C ambient (most prevalent)			
Evap temps, °C bubble, dew	–2.1, 3.6	–5.5, 0.3	–6.1, 0.3
Cond temps, °C bubble, dew	28.5, 33.8	29.8, 35.1	38.6, 43.7
COP	3.37	3.11	2.55
Estimated annual running cost, £	590.44	621.71	634.50

The operating conditions show which will be the most efficient. System A has a higher evaporating and lower condensing temperature at both ambient temperatures.

The cost of the three systems was very similar, but system A is 7% cheaper to run than system C. As these running costs are repeated at each of the Group's considerable number of sites, the testing, design and selection work proved well worthwhile.

8

DEFROST

Frost will build up on evaporators whenever the evaporating temperature is below 0°C. Where the store temperature is above 3°C, the store air can be used to 'naturally' defrost the coil.

See GPG 280 *Energy efficient refrigeration technology – the fundamentals*

Frost build up reduces heat transfer and air flow and thus the evaporating temperature, increasing the temperature lift. Frost build up can be minimised by careful design. The defrost method and control has a significant effect on efficiency.

Build up of frost on evaporators can be minimised at the design stage by:

- ensuring the evaporating temperature is as high as possible – if the evaporating temperature can be kept above 0°C, no defrost will be required (Example 13 overleaf);
- specifying evaporators where frost build up is not so critical (Example 14 overleaf);
- minimising the ingress of warm, humid air through doors/other openings by ensuring good door management and installing strip curtains or an air lock.

Frost build up will vary with ambient and load conditions, so the best form of defrost control is **defrost on demand.** This uses sensors to initiate defrost when the frost has built up to a pre-set level, i.e. before it significantly reduces the capacity and efficiency of the evaporator (Example 15 overleaf).

Sensors are also used to terminate the defrost immediately that all the frost has been cleared, otherwise the defrost heat enters the cooled space and has to be removed by the refrigeration system, effectively being paid for twice. However, if the ice has not all cleared when defrosting is terminated, performance will be reduced and ice build up will increase making the situation worse.

If defrost is timed, make sure defrosting occurs outside peak hours, e.g. for three defrosts a day, time the defrosts to occur at 8 am, 4 pm and midnight, to reduce the impact of defrosting.

Drain lines should be heated and insulated where the store temperature is below 0°C to ensure they do not become blocked and reduce the effectiveness of defrosting.

The most efficient type of defrosting will depend on the overall system design. Usually electric defrosting uses up to twice as much energy as cool gas or reverse cycle because, when the refrigerant is used to melt the ice, the heat is distributed where needed and at a lower temperature. With electric defrosting there are fewer electric heaters than evaporator tubes, so the heat has further to travel and is dissipated less easily over the whole of the fin block, with more heat entering the room.

The use of socks on the fans and cowls on the coil face is claimed to reduce defrost by as much as 40%. These fabric tubes fit over the evaporator outlet and prevent heat loss to the cold store during defrost heating. They also claim to reduce frost build-up around the evaporator.

Example 13

In a beer cellar the evaporator is sized so that the evaporating temperature never falls below 1°C. This ensures that ice will not build up on the coil and defrosting will not be necessary.

Example 14

Following on from Example 5 (cold store at –22°C), the fin spacing was changed from 4 mm to 6 mm to make frost build-up less critical. The next size evaporator had to be used, but the number of defrosts per day was reduced. The saving was £35.50 per year, with a list price increase of £108. Further savings also accrued from the higher evaporating temperature that was then possible with the larger evaporator.

Example 15

A freezer room defrost system was set to operate twice a day for each of the six coils. The total cost of defrosting was £2,000 per year. A defrost on demand system was installed, using infrared ice sensors. In winter only two defrosts a week were necessary. **The overall saving from defrost on demand was £1,500 per year for an investment of £3,000.**

9

AVOIDING HEAD PRESSURE CONTROL

Wherever possible, the head pressure of a system should be allowed to float with ambient temperature, to take advantage of the prevalent low ambient temperatures in the UK (Example 16).

It is, however, common practice on systems with thermostatic expansion valves to control the system so that the head pressure is held artificially high, usually because the thermostatic expansion valve does not control well when the condensing pressure falls significantly. Fitting electronic valves or balanced port valves overcomes this problem. They are more expensive than standard types but have the following additional benefits:

- closer control of the superheat;
- no capital cost for the head pressure control system;
- electronic expansion valves can be coupled with other system controls and monitoring/management systems.

See GPG 280 *Energy efficient refrigeration technology – the fundamentals*

Remember that the evaporating temperature falls with the condensing temperature because of the extra compressor capacity. This was taken into account in the example below where the evaporating temperature fell from –5.8°C to –7.5°C.

Even if it is necessary to limit the minimum condensing pressure, you should make sure it is set as low as possible. Specify the minimum condensing pressure to the installation/commissioning technician.

Liquid line pumps can be useful in facilitating lower head pressures – see Section 14 *Other components*.

Example 16

A system with a fixed head pressure operates with a condensing temperature of 42°C and with a COP of 2.84. By modifying the system to allow the head pressure to float, the average condensing temperature falls to 24°C and the COP rises to 4.26. **Avoiding head pressure control reduced running costs from £907 to £605 per year, an annual saving of £302.**

Good Practice Case Study 302 *Improving refrigeration performance using electronic expansion valves* gives another example of how savings were made by eliminating head pressure control through the use of electronic expansion valves.

10 COMPRESSOR SELECTION AND CONTROL

There is great scope for wasting money due to selection of the wrong compressor, because there are so many different types available. On larger systems, decisions also need to be made regarding control of system capacity to meet varying loads, and some ways are much more efficient than others.

GIL 52 *The engine of the refrigeration system: selecting and running compressors for maximum efficiency* contains additional guidance on compressors.

10.1 Understanding compressor data

It is important to understand how compressors are rated. The capacity and power input data are usually presented at the range of evaporating and condensing conditions and a ‘rating point’. This rating point comprises:

- useful superheat (the superheating of the gas in the evaporator);
- non-useful superheat (the superheating of the gas between the evaporator and the compressor);
- liquid sub-cooling (the cooling of the liquid between the condenser and the expansion device).

See GPG 280 *Energy efficient refrigeration technology – the fundamentals,* Section 2.2 *Compressors*

The rating conditions used by compressor manufacturers are not typical. On a real system these conditions will be different, significantly affecting the capacity and power input. When comparing and selecting compressors, use realistic operating conditions, rather than just the design point. The compressor data should include correction factors. If a compressor manufacturer’s software is used to select compressors it should allow you to input the actual operating conditions of the system, thus making the calculation much easier.

See GIL 56 *Energy savings from motor management policies*

When considering open compressors, take into account the efficiency of the motor and the drive/coupling. Most open compressor data give the power input, but this is to the compressor shaft. The actual power input to the motor will be higher because of losses in the motor and drive/coupling. Motors are typically 80 to 95% efficient, varying with size. The new EU motor efficiency labelling scheme gives guidance on motor selection for efficiency – ask your supplier, or see General Information Leaflet 56 *Energy savings from motor management policies*. Losses in the drive/coupling can be up to 5%.

Example 17

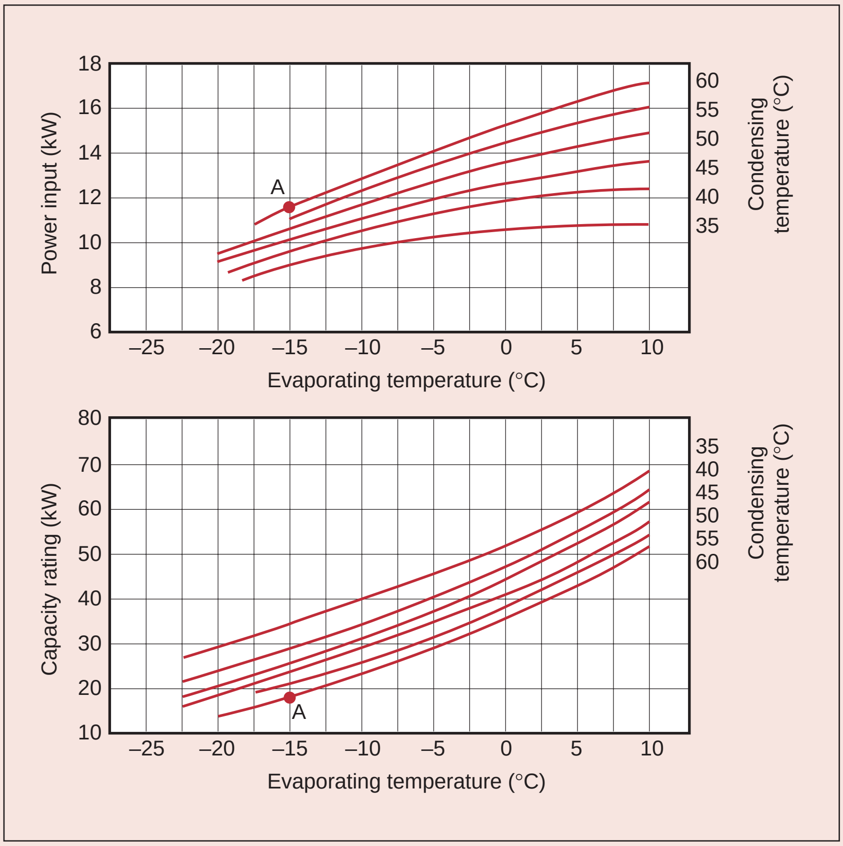

Figure 7 Compressor performance curves

Figure 7 uses typical manufacturer's compressor capacity and power input data, and assumes rating conditions of 25°C suction return temperature and 0°C liquid sub-cooling. However, these 'standard' conditions are totally unrealistic, as they assume that the 25°C suction temperature is all achieved in the evaporator (i.e. is useful superheat). Most systems will operate with between 5K and 10K useful superheat. The non-useful superheat will depend on the length of the suction line and how well it is insulated. (Non-useful superheat should be kept to a minimum to maintain compressor capacity and efficiency.) There is also always some liquid sub-cooling, again the amount depending on the length of the liquid line and condenser construction. **Therefore, use correction factors (usually available with the compressor data) to adjust the capacity and power input to realistic superheat and sub-cooling conditions.**

The COP of the compressor at –15°C and 60°C condensing (point A on the graphs) at the rating conditions is 1.61. When more realistic conditions are selected (6K useful superheat, 0°C suction return and 10K liquid sub-cooling) the COP rises to 1.70.

See GPG 280 *Energy efficient refrigeration technology – the fundamentals*

10.2 Evaluating compressor types

Selecting the most efficient compressor type depends on many criteria including:

- operating conditions;
- capacity required;
- whether capacity variation is necessary.

For small applications with compressors below 5 kW motor power (beer cellars, small cold stores, small liquid chillers, cabinets and other commercial appliances), the decision is quite straightforward (Figure 8).

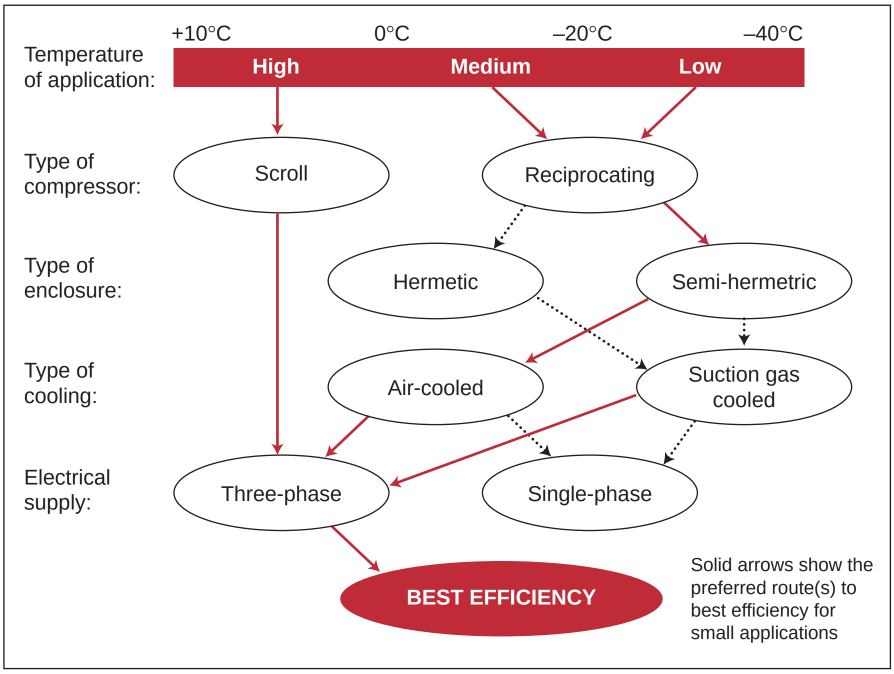

Figure 8 Selection of small (below around 5 kW) compressors: decision tree

The decision tree is a general rule of thumb (remember that different makes of compressors have different efficiencies), and the comparison may not always work out as shown.

Scroll compressors tend to be more efficient than other types at high temperatures because they have been primarily developed for air conditioning uses. Compressors with a semi-hermetic enclosure are usually more efficient than hermetics because they have the option of being air-cooled, thus minimising the temperature of the gas just before it is compressed. Three-phase motors are generally more efficient than single phase.

For larger compressors, the choice is between reciprocating and screw machines (and sometimes centrifugal types) with open or semi-hermetic enclosures. The most efficient option is not as clear cut as for smaller machines and for each application, different options should be investigated by calculation of energy consumption from manufacturer’s technical data.

Example 18

The COP figures in the table below were generated to determine the best compressor/refrigerant combination for a beer cellar held at 10°C at the maximum ambient condition (32°C).

	Reciprocating semi-hermetic	Reciprocating hermetic	Scroll
R22	2.95	2.78	3.19
R134a	2.85	2.94	2.61
R404A	2.62	2.40	3.00
R407C	2.77	2.62	3.10

The most efficient choice of compressor varies depending on the refrigerant used. R134a is often used in this application, so a reciprocating hermetic compressor would be the most efficient option. R407C is increasingly being used because of its superior performance, in which case the scroll compressor would prove best. The best COP at the design point is the scroll with R22, an option which is now not feasible as a long-term solution due to phase-out of HCFCs.

The comparison should also be carried out at more predominant operating conditions, i.e. 15°C ambient, 2°C evaporating, as well as at the design point. (At lower ambient temperatures cooling is less likely to be needed.)

	Reciprocating semi-hermetic	Reciprocating hermetic	Scroll
R22	4.30	4.03	4.91
R134a	4.28	4.35	4.13
R404A	3.34	3.20	4.60
R407C	4.12	3.42	5.02

Examining these results shows that the scroll compressor with R407C is the best option overall, and this was the option that was selected.

Example 19

This example shows how comparisons can be easily made to find the most efficient solution for a particular application.

The table shows a comparison of reciprocating and screw, open and semi-hermetic compressors at two operating conditions, all with R407C. This information has been generated from software provided by a compressor manufacturer for its compressors. The rating conditions are 5K liquid sub-cooling, 10K superheat.

	Capacity (kW)	Power input (kW)	COP
0°C evaporating 50°C condensing			
Reciprocating, open	63	22.39	2.81
Reciprocating, semi-hermetic	60.2	20.41	2.95
Screw, open	63	24.98	2.52
Screw, semi-hermetic	57.1	24.10	2.37
–15°C evaporating 40°C condensing			
Reciprocating, open	55.7	24.19	2.30
Reciprocating, semi-hermetic	53.4	21.96	2.43
Screw, open	56.0	25.89	2.16
Screw, semi-hermetic	53.1	23.86	2.22

With the open drive compressors, a loss of 3%, due to the coupling, and a motor efficiency of 88% have been assumed. The reciprocating machines run at 1,450 rpm, the screws at 2,900 rpm.

In Example 19, calculations for this particular situation show the semi-hermetic reciprocating machine to be the most efficient option. The results of this analysis would be different for each situation, depending on:

- size of compressors used (e.g. the efficiency of open motors and screw compressors increases as the motor size increases);
- operating conditions (e.g. screw compressors are most efficient at a particular volume ratio);
- refrigerant (e.g. only open drive compressors can be used with ammonia).

There are other general considerations to take into account. Screw compressors:

- can have a fixed volume ratio (V_1) making it important to ensure they closely match the load at the most prevalent operating conditions, however, machines with variable volume ratio are now available which automatically adjust their V_1 to match prevailing operating conditions;
- are generally good for constant high loads where they have been well matched to the load, but are poor if they operate for long periods at low loads, although this limitation does not apply to the variable speed screw compressors that are becoming increasingly available;
- are useful in situations where precise control of evaporating temperature is required, for example, in water cooling where operation is close to 0°C;
- require ancillaries, such as oil coolers and pumps, and any power used by these must be taken into account when calculating COP.

10.3 Capacity control

See GPG 280 *Energy efficient refrigeration technology – the fundamentals,* Section 2.2 for an explanation of capacity control

It is often necessary to operate compressors at part load. Most medium-to-large compressors can be supplied with some form of capacity control, but all these methods reduce the efficiency of the compressor.

The least efficient method is where the discharge gas is bypassed back to suction, as the compressor input power remains the same as on full load and the compressor tends to overheat. Avoid this method wherever possible.

Variable speed drive compressors are becoming more widely available, and this form of capacity control gives the lowest efficiency penalty.

The efficiency of other methods varies according to manufacturer and operating conditions. When selecting a compressor which will operate on capacity control, compare different types at prevalent part load conditions as well as at full load.

10.4 Compressor configuration for large, varying loads

The load profile is essential when selecting the best compressor configuration. Different options should be compared at the most common operating conditions as well as at the design condition.

When the refrigeration load is large and can vary significantly, there are many options for matching the compressor to the load, as illustrated in Figure 9:

- one large compressor with inbuilt capacity control;
- two or more smaller compressor of the same size, which are cycled to match the load[2];
- two or more smaller, unevenly-sized compressors, which are selected and cycled to match the load.

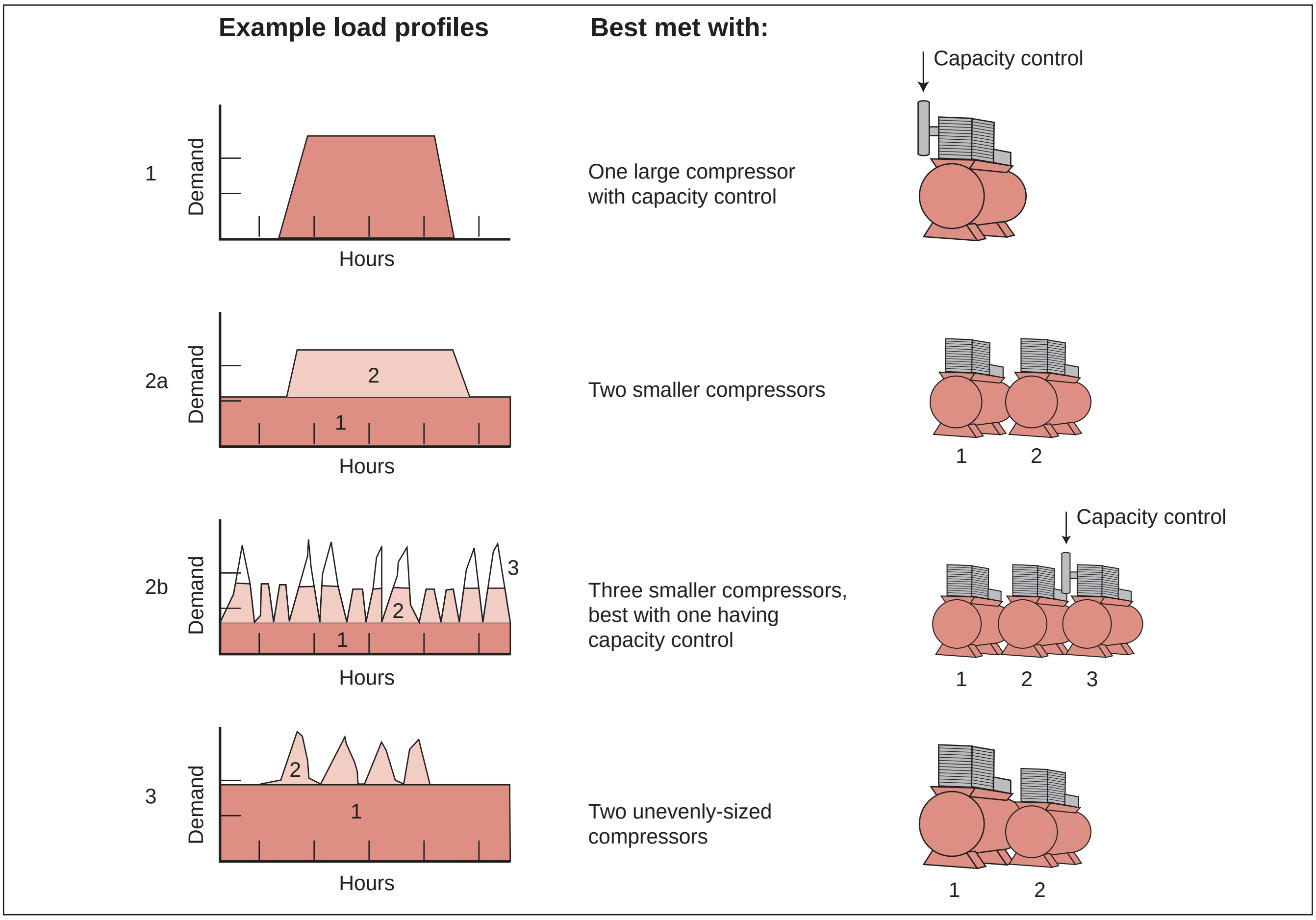

Figure 9 Matching compressor number and size to best suit the load profile

[2] Two scenarios for this are shown for different load patterns – 2a and 2b. A further improvement would be to have one compressor with inbuilt capacity control to more accurately match the load variation.

The efficiency of the different options varies enormously, and there is no hard and fast rule as to which is the best solution. When the load profile demands that a compressor runs on capacity control, select the compressor which performs best at this part load condition.

Switching a compressor off to reduce the system capacity is the most efficient method of meeting a reduced load. The efficiency of a compressor operating on inbuilt capacity control is always lower than when it operates at full load.

As a general rule of thumb, one compressor with capacity control is rarely the most efficient option. A pack of unevenly-sized compressors will usually produce the most efficient system, but the control system would need to be carefully designed.

10.5 Control strategy

Regardless of the option selected to meet a load, the control of the compressors is important. The control strategy should be designed to:

- select the most efficient mix of compressors to meet the load;
- avoid operation on inbuilt capacity control when possible (if capacity control is necessary, try to ensure the reduction in capacity is as low as possible);
- avoid operation at low suction pressures when possible;
- avoid short cycling any compressor.

Selecting compressors of different sizes and designing a good control strategy to cycle them to match the most common loads accurately is often the most efficient option.

Example 20

This example shows how the control strategy is especially important with screw compressors. A control programme which ramps a number of machines up and down in parallel can result in poor energy efficiency.

Figure 10 shows a curve of percentage capacity vs percentage full load power that is typical of a screw compressor. At 60% capacity, the power absorbed is approximately 80% of full load power. Therefore, three machines controlled in parallel, each running at 60% capacity (total 180% capacity) would use 240% full load power. With one compressor as base machine fixed at 100% capacity, however, and one running as a trim machine at 80% capacity (absorbing 88% full load power), the power use falls to 188% full load power.

The case is actually worse than shown because the capacity will be lower than 60% with the slide position at 60%.

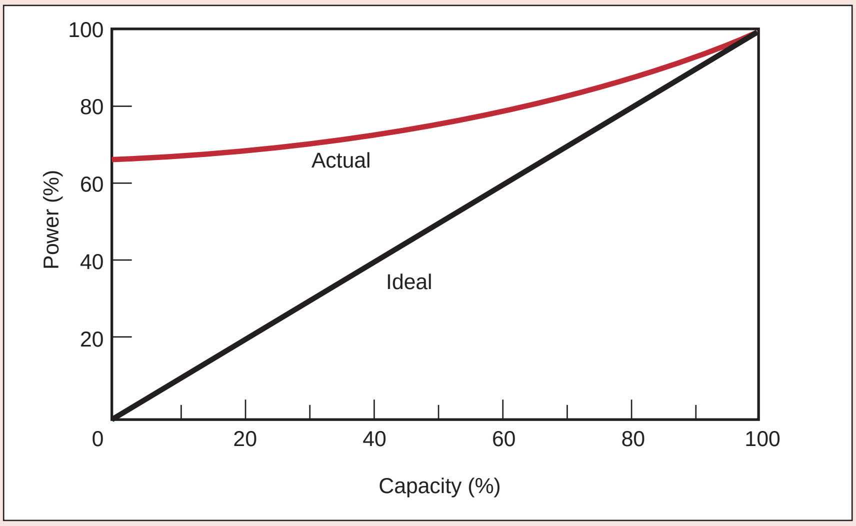

Figure 10 Power versus capacity curve for a screw compressor

10.6 Compressor location

Compressors need to be cooled adequately to maintain their performance and life. Make sure you specify:

- plant room cooling (usually forced ventilation with air in and out);
- cooling fans to be fitted to the compressor where necessary;
- that the compressor is to be operated within its application range, especially with regard to suction return temperature.

11 REFRIGERANTS

There are many and varied refrigerants available on the market and very few general rules as to which are the most efficient. The efficiency depends on:

- operating conditions;
- compressor type;
- size of application.

There are many other factors which govern refrigerant selection, the most important are listed below.

- Safety issues – ammonia and hydrocarbons are often the most efficient refrigerants to use, but their applications can be limited by their flammability and, in the case of ammonia, by its toxicity and reaction with copper. Designing and installing systems to BS4434:1995 *Safety and environmental aspects in the design, installation and operation of refrigerating systems and appliances* will reduce these risks to levels similar to other refrigerants.

- Long-term availability – many refrigerants are based on HCFCs which will be phased out of production and restricted in their use. It is no longer sensible to offer HCFC refrigerants and blends containing HCFCs.

See GPG 280 *Energy efficient refrigeration technology – the fundamentals,* Section 2.2 for an introduction to refrigerants

Example 18 on page 31 shows how energy efficiency is affected not only by refrigerant type, but also by compressor type and operating conditions (ambient temperature in this case). The example illustrates the importance of comparing different refrigerants for each application, and not assuming that because a certain refrigerant has been used in that application, it is the best to use. Over the last few years R134a has been the HFC refrigerant of choice for most beer cellars, but the example shows it is not the most efficient to use.

11.1 Computer comparisons

Computers are often used to predict which refrigerant will be most effective for a particular application. The accuracy, and therefore the usefulness, of these is limited, however, because they do not take account of:

- the different heat transfer rates of refrigerants;
- the effect of oil mixed with the refrigerant;
- the operation of blends in some systems.

The only accurate method to determine the best refrigerant is by independent test work which mimics typical, real operating conditions.

11.2 Refrigerant blends

Many new refrigerants are zeotropic blends of two or more substances. In the evaporator and condenser these refrigerants do not behave as a single substance, but instead evaporate and condense at a range of temperatures (the temperature glide). This can give an improvement in capacity for no additional power input if the evaporator and condenser are designed to take advantage of the glide.

For example, take the case where there is air blowing across the coil of an air cooler. With a single substance refrigerant or an azeotropic blend (i.e. one that behaves as a single substance), the air cools and the refrigerant continues to evaporate at the same temperature along the coil. The temperature difference between the air and refrigerant therefore reduces across the coil, and the heat transfer rate also reduces as the rate is proportional to temperature difference.

With a zeotropic blend, the evaporating temperature increases as the refrigerant evaporates. If the air and refrigerant are in counter-flow, the temperature difference between the air and refrigerant can be maintained, and so the heat transfer rate will be more constant.

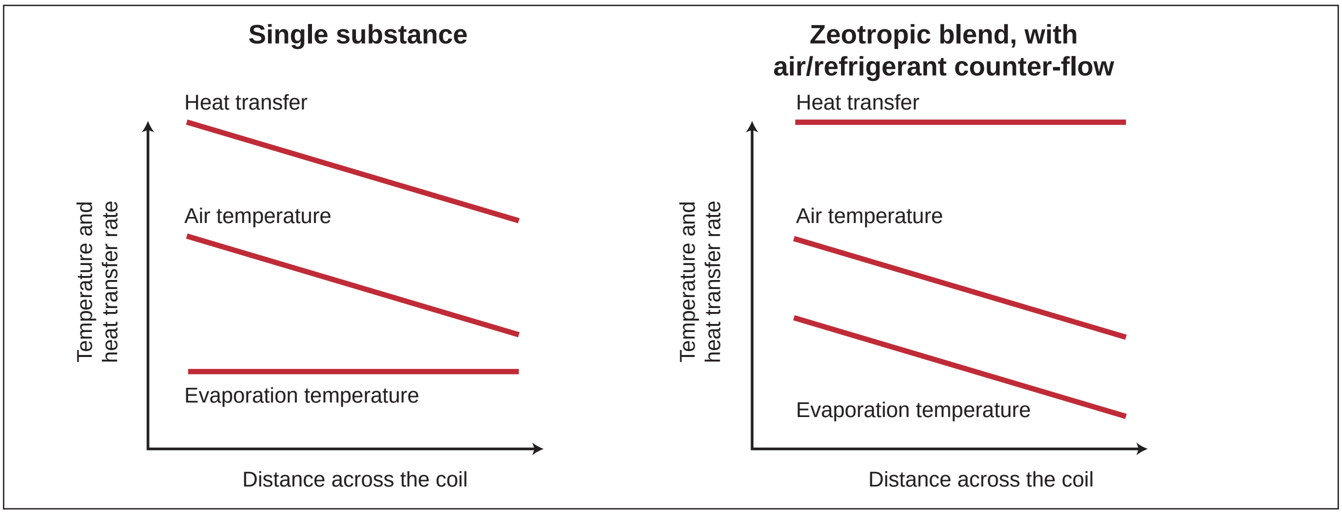

Figure 11 Temperature and heat transfer across the surface of a coil

See GPG 280 *Energy efficient refrigeration technology – the fundamentals*

Zeotropic blends give a capacity and efficiency advantage whenever the refrigerant and cooled medium is in counter-flow. If they are not in counter-flow, there is no advantage and there may be a reduction in performance.

Zeotropic blends are most suited to direct (dry) expansion systems. Check with the refrigerant supplier before specifying them for flooded systems.

11.3 Reducing refrigerant emissions

The efficiency of a system reduces as refrigerant leaks out, so the system should be designed to be as leak proof as possible by:

- Using as few joints as possible.
- Using brazed joints wherever possible. Where these cannot be used, manufactured flares are better than flares made by the installation technician on site.

These points apply to the main pipework and the gauge and control lines.

Specifying pressure relief devices which relieve the pressure from the high to the low side, rather than into the air, can also reduce emissions. There must be a relief device that can vent the refrigerant from the low side into air as well.

The system will leak less if the high side pressure is kept as low as possible by:

- using a refrigerant which operates with low pressures (although you need to bear in mind that, in general, higher pressure refrigerants have a higher capacity and efficiency);
- using a large condenser (this is consistent with good design for energy efficiency).

See GPG 178 *Cutting the cost of refrigerant leakage*

The amount of refrigerant should also be kept as low as possible to reduce the impact on the environment of a leak by:

- designing systems with as small a refrigerant charge as possible (but don't reduce the evaporator or condenser size!) – plate heat exchangers can be useful here;
- making sure the evaporator is as close to the compressor/condenser/ receiver as possible;
- using vertical receivers (which need less refrigerant to maintain the minimum level);
- specifying the amount of refrigerant charge needed (to prevent the system being overcharged during commissioning).

12

PIPEWORK

In Section 7, the importance of low temperature lift is emphasised. The level of energy efficiency achieved by installing generously-sized evaporators and condensers can, however, be lost by penny pinching on interconnecting pipework, although pipework generally represents a small fraction of the overall project cost. 'Tight' sizing of the suction and discharge lines can result in a significant increase in the temperature lift across the compressor, reducing energy efficiency (Figure 12).

See GPG 280 *Energy efficient refrigeration technology – the fundamentals*

It is important, however, that all line sizes are not simply increased without careful consideration.

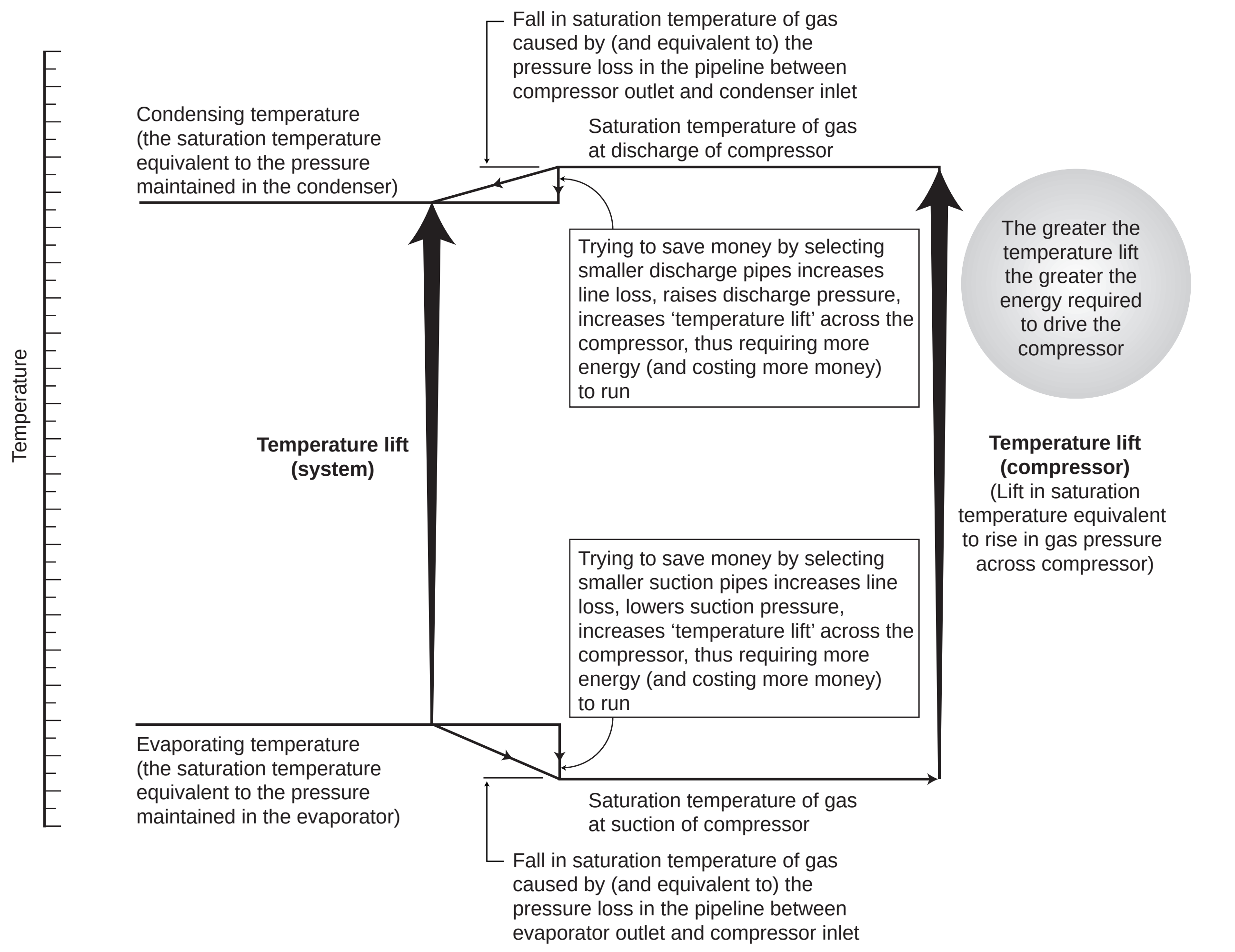

Figure 12 How pipe selection increases the effective temperature lift seen by the compressor

In direct (or dry) expansion systems it is necessary to maintain an appropriate minimum gas velocity in the suction lines to the compressor, so that oil leaving the evaporators is drawn back to the compressor on a continuous basis. Low velocity can allow the oil to 'fall out' of the gas stream, building slugs of oil which can cause operational problems or even system damage. Similarly there is a need, in vertical discharge lines, to entrain lubricating oil which has left the compressor.

For flooded expansion evaporators, high gas velocity must be maintained in any rising sections between evaporator and suction separator or surge drum. Where the need to rise cannot be avoided, use a short vertical section with pipe bore sized to maintain gas velocity. 'Horizontal' sections must be laid to a fall to assist liquid flow, and generously-sized pipework will help to minimise the overall pressure drop in the wet return piping. Avoid liquid traps (low points) in lines carrying a mixture of liquid and gas, as the liquid will settle out and cause surging flow.

Size liquid lines to minimise pressure drop as necessary, to avoid generation of 'flash gas' which occurs when the pressure in the liquid line falls below that equivalent to saturated pressure at the temperature of the liquid in the line.

The line between condenser and receiver is generally run continuously down hill and larger than the main liquid line to allow 'sewer' type flow. This allows liquid to flow from condenser to receiver in the lower part of the pipe, and gas to return from receiver to condenser in the upper part without impeding liquid flow. This 'reverse' gas flow is necessary to allow condenser and receiver pressures to equalise at a time of falling condensing pressure, e.g. after an additional condenser fan has cut in. If reverse gas flow is not possible, liquid backs up in the condenser, increasing condensing pressure unnecessarily (and also leading to possible interruption in liquid flow to evaporators). An alternative approach is to run a separate gas or 'equalising' line between receiver top and condenser inlet. The receiver should always be below the condenser.

It is a false economy to install pipelines that result in gas and liquid velocities at the top end of the acceptable range. Pipes and fittings only form a small part of the overall project cost, so the increase in cost associated with larger pipes and fittings is a very small percentage of the total. Pipes that are too small result in an operating cost penalty that continues for the operational life of the plant.

Piping for heat transfer fluids (chilled water, secondary refrigerants or brines) should also be sized for reasonably low velocities to avoid increased pumping power. The circulation pump power has to be included in the system energy required to achieve the desired refrigeration effect, and hence influences the COP for the overall system. A simple rule for choosing the size of heat transfer fluid pipework is:

- for pipe less than 150 mm internal diameter, use 2 m/s;
- for pipe more than 150 mm internal diameter, use 3 m/s.

Always chose a size to keep the refrigerant velocity below these values.

13

DESIGN FOR MONITORING AND MAINTENANCE

All the good work carried out in achieving an energy efficient design can be wasted if the plant is subsequently poorly maintained. The design of plant can have a big impact on how easy and cost-effective it is to maintain, and this in turn will impact on its long-term running costs and reliability for the user.

Good plant maintenance is helped by monitoring or logging system operation. Build in, at the design stage, system monitoring equipment to help ensure the plant works efficiently throughout its life. At the very least, this should include gauges and appropriate thermometer points. With larger industrial plant, it is usually cost-effective to include a computerised monitoring system based on a SCADA package (statistical control and data acquisition). Monitoring equipment is invaluable at the commissioning stage too.

See GPCS 302 *Improving refrigeration performance using electronic expansion valves*

The exact type of equipment that should be incorporated depends on the size and complexity of the plant – see Table 2 overleaf.

You need to specify:

- what should be measured and how;
- how often should it be measured;
- how the information should be used.

See GPG 279 *Running refrigeration plant efficiently – a cost-saving guide for designers*

13.1 What to monitor

Table 2 includes the conditions which should be logged – they are useful when determining how efficiently a system is operating, and in detecting trends which may lead to failure. For smaller systems (e.g. small-to-medium cold stores, beer cellars, milk tanks, packaged chillers) only those conditions given in bold are usually necessary.

Table 2 Conditions which should be logged

Basic measurements useful for ***all*** plant	***Additional*** measurements useful for medium-sized and larger plant
Compressor suction pressure	Compressor suction temperature
Compressor discharge pressure	Compressor discharge temperature
Compressor hours run	Compressor current draw
Compressor oil level	
Refrigerant level	
Cooled space temperature	
Condenser air on/water inlet temperature	Condenser air off/water outlet temperature
	Secondary refrigerant temperatures
	Ambient temperature
	Electricity consumption
	Evaporator air on/liquid inlet temperature
	Evaporator air off/liquid outlet temperature

13.2 Design for easy access

Refrigeration plant is often poorly maintained because it is difficult to access, or because insufficient isolation valves are included in the plant design. To ensure a system can be easily maintained and serviced:

- site the plant where it is safely and easily accessible (e.g. make sure there is safe access to air-cooled condensers for cleaning, that the compressor and liquid level sight glasses are visible);
- include gauges and thermometer points for good monitoring and fast diagnosis of faults;
- include receiver liquid level indicators/sight glasses for accurate charging and monitoring of refrigerant level;
- include valves at strategic points in the system to allow key components such as filter driers, to be replaced with minimum downtime;
- include an air purging point if the suction pressure will be below or close to atmospheric pressure during operation.

14

OTHER COMPONENTS

Other components in refrigeration systems that affect efficiency are covered in this Section.

Other components **Page**

See GPG 280 *Energy efficient refrigeration technology – the fundamentals* for definitions of the most common types of expansion device

14.1 Expansion devices

Thermostatic expansion valves are the most common type used in commercial systems. They do not control well if the pressure drop reduces (as it will when the head pressure floats with ambient temperature). Use electronic or balanced port valves as often as possible to allow for this.

The size of the valve is important. Too large and it will not control correctly to let the required amount of refrigerant into the evaporator, resulting in inefficient operation and sometimes compressor failure. Too small and it cannot allow enough refrigerant into the evaporator, reducing its capacity and the system efficiency.

The evaporator will be most efficient when it is full of evaporating refrigerant. With most systems, this is not practical as liquid refrigerant will be allowed to return to the compressor, causing damage and possible failure. The expansion valve should be set so that the refrigerant is superheated by about 5K when it leaves the evaporator. This ensures all refrigerant has evaporated and gives a safety margin for when load conditions change.

Flooded evaporators generally employ liquid feed control via:

- a float valve with modulating opening as the expansion device;
- a pilot float controlling a modulating main valve for larger systems;
- a manually adjusted expansion valve where liquid flow is turned on-off by a solenoid valve upstream, the solenoid being controlled by a float (level) switch.

An important advantage of all of these systems is that they can be sized for adequate liquid flow at low condensing pressure and hence low liquid line pressure. At times of higher condensing pressure, the valve modulates to a smaller opening, or the solenoid is closed for a larger percentage of the time. Thus full advantage can be taken of compressor power reduction due to falling condensing pressure at times of cooler ambient and/or reduced duty on the system.

14.2 Fan motors and pumps

Heat gain from fan motors adds to the load in a refrigerated space, resulting in energy consumption in the refrigeration system to counteract this additional heat load, in addition to the direct use of power by the fan itself. Heat gain can be minimised by:

- running fans only when cooling is required;
- using higher efficiency, low energy motors;
- installing the motor outside the refrigerated space.

The first two points also apply to primary and secondary refrigerant pumps. The latter is usually not justified except in very special applications, e.g. very low temperature chambers.

On plants with a fluctuating load, variable speed pumps or variable or two-speed evaporator fans should be considered and the associated payback period determined.

Example 21

A large warehouse holding beer at 9°C used two fans in a ducted air system. The fans ran continuously, even when refrigeration was not required, absorbing 122 kW of power each. **Installing variable speed drives saved £80,000 per year, against a capital cost of £63,000** (including a variable speed drive for a pump, and a more sophisticated controller), giving a payback of less than 10 months. The disappointing aspect of this saving is that it took a year to persuade the end-user to make the investment.

14.3 Solenoid valves

Where pressure losses are particularly important, for example, in the wet return lines from flooded evaporators, solenoid valves can represent a large pressure drop. An alternative is to use a valve actuated by refrigerant gas pressure. The gas pressure is applied or isolated by a small, pilot solenoid valve in much the same way as a pneumatic valve would operate.

14.4 Liquid line pumps (liquid pressure amplifiers)

Liquid line pumps are used to increase the pressure level between the condenser/receiver and expansion valve in direct expansion systems. The increased pressure of the sub-cooled refrigerant is argued to increase capacity and system efficiency, and also helps to overcome flash gas-related problems that often arise in medium-sized and larger systems. Significant energy savings and improvements in reliability have been reported by users of liquid line pumps.

Hermetically-sealed magnetic-drive pumps are available to install between the condenser/receiver outlet and the expansion valve. The head developed may be only 1 bar or 2 bar, but evidence suggests that this pressure change is adequate to have the following beneficial effects:

- helping overcome the effects of pressure drops in the liquid line and its components, which should reduce flash gas, thereby assisting effective operation of the expansion valve and restoring refrigerating capacity;
- allowing the valve to control well, even when the head pressure floats down with falling ambient temperature, so lowering the temperature lift that the compressor sees and reducing energy consumption;
- improving the return of oil, through the improved percentage of liquid-to-vapour in the evaporator;
- reducing superheat by injecting a small portion of the pumped liquid into the discharge line of the compressor, which effectively reduces discharge temperature and increases condenser capacity.

Use of a liquid line pump is most beneficial when incorporated at the initial design stage. Retro-fit is also possible, and designers should consult suppliers for advice.

NB: The pump may require condenser fans to run for longer periods and this must be taken into account in energy calculations. Designers must ensure that oil return, defrost functions and distribution of liquid within the evaporator are not adversely affected by the reduction in head pressure and flash gas.

15

INSTALLATION, COMMISSIONING AND MAINTENANCE

The installation and maintenance procedures should be specified as part of the design process, and commissioning information should be provided. Without these, the plant is unlikely to perform as designed. Provision of appropriate guidance from designers will avoid problems for users and contractors at, and after, installation. The designer's reputation is also at stake.

Other guides in this series provide relevant information:

- GPG 281 *Installation and commissioning for efficient refrigeration plant – a guide for technicians and contractors*;
- GPG 282 *Service and maintenance for efficient refrigeration plant – a guide for technicians and contractors*;
- GPG 278 *Purchasing efficient refrigeration – the value for money option*.

The first two Guides are written for technicians, the third for non-expert owners/users. All three contain much advice on what readers should expect from the designers and suppliers whose plans they implement or buy. Designers would be well advised to be familiar with them.

15.1 Installation

Installation should be according to BS 4434. In particular, designers should specify:

- pipework details, including size, routing and type of joints, paying particular attention to pressure drops as discussed in Section 12 *Pipework*;
- siting of components, especially air-cooled condensers to ensure adequate flow of cool air, shading from direct sun and access for maintenance;
- plant room design, in particular ventilation to ensure heat does not build up to affect operation;
- pressure and leak test pressures to comply with BS 4434;
- the amount of refrigerant charge, to ensure the charge is adequate for all load conditions and that the system is not overcharged – both of these have a serious impact on efficiency.

15.2 Commissioning information

Commissioning involves checking that the equipment is installed and the controls are set correctly, as well as training in-house personnel in the plant's operation. If the plant is poorly commissioned it will run inefficiently. To avoid this, designers need to provide written procedures for commissioning well in advance that include:

- settings for all controls, including pressure switches, expansion valves and head pressure control if fitted;

See GPG 281 *Installation and commissioning for efficient refrigeration plant – a guide for technicians and contractors*

- discharge and suction pressures/temperatures for a range of operating conditions;
- commissioning as closely as possible to conditions of maximum load in the summer months;
- instructions to record the operating conditions so that the user/maintenance contractor can refer back to them later;
- carrying out a thorough leak and strength test;
- ensuring that the system is suitably evacuated and dehydrated before final charging (see GPG 281 *Installation and commissioning for efficient refrigeration plant – a guide for technicians and contractors* for further details).

As a minimum, commissioning should involve checking:

- operating pressures and temperatures;
- the storage/produce temperature, including thermostat settings;
- the electrical supply and operating parameters of the compressors and all other electrical motors;
- the function of any special features built into the compressor(s), including off loading or capacity control devices, etc.;
- oil level in the compressor(s);
- the settings and operation of any pressure safety devices;
- setting all system controls, including the superheat of the thermostatic expansion valve(s);
- the operation of the defrost system, where one has been used;
- the level of vibration.

15.3 Performance tests

After commissioning, the contractor should conduct one or more performance tests to ensure that the plant is meeting its design specifications. The designer should indicate which test is appropriate. There are four main tests.

1. **A demonstration run.** Operating the plant under supervision for a reasonable period and noting the performance data from the normal instrumentation. The results can then be compared with the specification and the contractor's tender. Clearly, this is best carried out at times when the plant is operating close to design capacity.
2. **A formal full-load test.** Not common due to the cost, but it is often possible to make special arrangements to ensure that the plant operates at its design conditions. This may involve simulating cooling loads with electric heaters, and additional calibrated instrumentation.
3. **A factory test.** This is suitable for many types of factory-built, packaged units, but not applicable to site-installed systems.
4. **Monitoring over an extended period.** The plant is monitored over a period and its performance checked against predicted running costs. However small a plant is, it is always worth monitoring. As a minimum, suction/discharge gauge pressures should be logged. If they change when they shouldn't it indicates that something is wrong, and the contractor can be alerted.

Proving compliance with design specification will increase user confidence and may well lead to follow-on business. Monitoring is more fully discussed from the users point of view in GPG 279 *Running refrigeration plant efficiently – a cost-saving guide for owners*.

15.4 Maintenance procedures

The type of maintenance to be carried out and its frequency should be specified by the designer to ensure the performance of the plant is maintained, and work should be carried out to any relevant codes of practice and British Standards. At the very least the procedures should include:

See Appendix D for information on standards

- condenser and evaporator cleaning (take into account local environmental conditions when specifying frequency of air-cooled condenser cleaning);
- leak testing;
- checking the compressor oil level;
- checking the state of insulation and door seals/strip curtains/air locks where necessary;
- checking defrost;
- checking the operation of safety and control devices.

For larger plant (over 25 kW installed motor power) a pressure vessel inspection scheme should also be specified.

15.5 Documentation requirements

After commissioning and performance testing, the supplier should provide at least two complete sets of operating documentation for the plant. The documentation should include:

- the design operating parameters;
- a complete set of installation diagrams together with system piping and wiring diagrams;
- the operating and servicing instructions for any major items of equipment included in the installation;
- quality assurance documentation on pressure vessels, installation and testing;
- a list of recommended spare parts for critical equipment;
- a complete set of control settings;
- the records from the commissioning procedure, a maintenance plan and, where appropriate, a pressure vessel inspection scheme.

A

LOAD PROFILE AND ENERGY RUNNING COST FORM

The following form can be used to calculate annual running costs and hence lifetime costs for a refrigeration system.

Ambient temperature range	**Under 5°C**	**5°C to 10°C**	**10°C to 15°C**	**15°C to 20°C**	**20°C to 25°C**	**25°C to 30°C**	**Above 30°C**
Days per year in this range Consult local weather data from CIBSE, the Meteorological Office or equipment suppliers, or call the Environment and Energy Helpline on 0800 585794							
Heat load (kW) Calculate this for each ambient temperature range using the midpoint temperature (see Section 4)							
System capacity (kW) Calculate this for each heat load/midpoint ambient condition							
Power input (kW) Calculate this for each condition as above							
System COP COP = capacity/power input (not essential, but indicates how performance varies)							
Compressor running time (%) % = 100 x load/capacity							
Compressor running time, hours per day (hours) Hours = % x 24							
Energy input at ambient range (kWh) kWh = compressor power x hours/day x days/year							

Annual energy input (kWh) = sum of all numbers in the last row of the table above (kWh)

Annual cost (£) = annual energy input (kWh) x cost of electricity (£/kWh)

B

TEWI CALCULATION EXAMPLE

This example is taken from the booklet *Guideline methods of calculating TEWI*, published by the BRA. For full details on calculating TEWI, consult this booklet.

See Appendix C for BRA contact details

Complex System An industrial process refrigeration system (for example a brewery) operating with R22 as the refrigeration working fluid in a direct refrigeration system.

In this application, the refrigeration load and operating conditions are dependent on seasonal demand and climate conditions. As shown below, the use of seasonal segmentation is important in assessing the true annual refrigeration load and power demand and the load model needs to be used (see Table 3). These differing load conditions affect the compressor performance (see Table 4). In addition there are significant auxiliary power loads associated with the systems (see Table 5).

All these data are used in the TEWI Calculation spreadsheet.

Table 3 Refrigeration load definition

Segment number	**Segment name**	**Cooling load (kW)**	**Nominal cooling temp (°C)**	**Heat rejection temp (°C)**	**Hours per year**
1	Normal	n/a	n/a	n/a	8,760
2					
3					
4					
5					
6					
7					
8					
9					
Total					**8,760**

Table 4 Annual compressor energy consumption calculation

Segment number	Segment name	Compressor cooling load (kW)	Evaporating SST (°C)	Condensing temp (°C)	Liquid temp at TXV (°C)	Return gas temp (°C)	Compressor power draw (kW)	Hours per year	Compressor energy consumption (kWh)
1	Summer load – peak	1,500	–15	38	28		563	500	281,500
2	Summer load – base	900	–12	26	20		226	1,000	226,000
3	Summer load – standby								
4	Autumn/Spring load – peak	1,200	–15	25	20		330	1,800	594,000
5	Autumn/Spring load – base	700	–12	20	18		147	3,600	529,200
6	Autumn/Spring load – standby								
7	Winter load – peak	1,000	–15	17	18		220	600	132,000
8	Winter load – base	500	–12	15	18		90	1,260	113,400
9	Winter load – standby								
Total								**8,760**	**1,876,100**

Table 5 Annual energy consumption – ancillary components

Segment number	Component	Power draw (kW)	Annual utility factor (%)	Annual energy consumption (kWh)
1	Fans: condenser	60	50	262,800
2	Fans: evaporator			
3	Fluid circulation pumps	50	100	438,000
4	Defrost heaters			
5	Others (state)			
			Total	**700,800**

TEWI Spreadsheet: Standard Method of Calculation

Job Reference: .... Example TEWI

Date: ....

INPUT DATA

Application sector:	Liquid chiller (industrial)		1
Refrigerant fluid:	R22		2
Refrigerant charge:	1,200	kg	3
Annual energy consumption (kWh)			
Compressors:	1,876,100		4
Ancillary:	700,800		5

SECTORAL FACTORS

System operational lifetime:	15	yr	6
Refrigerant GWP:	1,700		7
L1	2		
L2	0.5		
S1	0.25		10
S2	0		11
Recover efficiency:	0.95		12
CO_2 emission factor, β (kg CO_2/kWh)	0.53		13

TEWI CALCULATION

a) Direct effect

Refrigerant loss (operational) = (3) x (6) x (8 + 9 + 10 + 11)/100	495	14
Refrigerant loss (retirement) = (3) x 1.00 – (12)	60	15
Total lifetime refrigerant loss (kg) = (14) + (15)	555	16
CO_2 equivalent (kg) = (16) x (7)	943,500	17

b) Indirect effect

Indirect effect (kg CO_2) = ((4) + (5)) x (13) x (6)	20,486,355	18

c) TEWI

TEWI = (17) + (18)	21,430,000	kg CO_2
= ((17) + (18))/1000	21,430	tonne CO_2

C

USEFUL CONTACTS

The following are a list of useful contacts for further information on refrigeration and air conditioning.

American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE)
1791 Tullie Circle, NE, Atlanta, Georgia, GA30329-2305, USA
Tel: +1 404 636 8400
Fax: + 1 404 321 5479
Internet Home Page: www.ashrae.org

Building Research Establishment (BRE)
Bucknalls Lane, Garston, Watford, Hertfordshire WD2 7JR
Tel: 01923 664000
Fax: 01923 664790

BRECSU
Building Research Establishment, Garston, Watford WD2 7JR
Tel: 01923 664 000 Switchboard
Tel: 01923 664 258 Energy Efficiency Enquiries Bureau
Fax: 01923 664 787
Email: brecsuenq@bre.co.uk

BRECSU manages the Government's Energy Efficiency Best Practice Programme for buildings applications, including air conditioning.

Federation of Environmental Trade Associations (FETA)
Henley Road, Medmenham, Marlow, Bucks SL7 2ER
Tel: 01491 578674
Fax: 01491 575024
Email: info@feta.co.uk
Internet Home Page: http://www.feta.co.uk

This Federation comprises a number of independent but related associations allied to the refrigeration and air conditioning industry, including the BRA and the Heat Pump Association.

British Refrigeration Association (BRA)
Henley Road, Medmenham, Marlow, Bucks SL7 2ER
Tel: 01491 578674
Fax: 01491 575024

The BRA is a trade association of suppliers of refrigeration equipment and services. It has sections for designers, manufacturers, distributors and installers of components and systems and also for users. The BRA has an active interest in training.

British Standards Institution (BSI)
389 Chiswick High Road, London W4 4AL
Tel: 020 8996 9000
Fax: 020 8996 7400

The BSI publishes a complete range of British Standards covering the manufacture, installation, testing and safety of refrigeration systems and components. A BS number followed by EN indicates that the Standard is also a Euronorm.

Building Services Research and Information Association (BSRIA)
Old Bracknell Lane West, Bracknell, Berkshire RG12 7AH
Tel: 01344 426511
Fax: 01344 487575

Chartered Institute of Building Services Engineers (CIBSE)
Delta House, 222 Balham High Road, London SW12 9BS
Tel: 020 8675 5211
Fax: 020 8675 5449

CIBSE produces Codes of Practice for the installation and commissioning of refrigeration and air conditioning systems.

Cold Storage and Distribution Federation
Downmill Road, Bracknell, Berkshire RG12 1GH
Tel: 01344 869533
Fax: 01344 869527

Department of Trade and Industry
Environmental Division, 151 Buckingham Palace Road, London SW1W 9SS
Tel: 020 7215 1018

The DTI publishes a number of booklets regarding environmental aspects concerning the refrigeration and air conditioning industry, and has also published market studies into the use of refrigerants.

Health and Safety Executive (HSE)
HSE publications are available from:
HMSO Publications Centre, PO Box 276, London SW8 5DT
(mail, Fax and Tel orders only)
Tel orders: 020 7873 9090
Fax orders: 020 7873 8200
General enquiries: 020 7873 0011

The HSE publishes a wide range of books and information leaflets regarding regulations concerning safety in all branches of commerce and industry. All HSE priced publications can be bought from any HMSO Bookshop or their agents (see Yellow Pages).

The Heating and Ventilating Contractors' Association (HVCA)
ESCA House, 34 Palace Court, Bayswater, London W2 4JG
Tel: 020 7229 2488
Fax: 020 7727 9268

The Institute of Refrigeration (IoR)
Kelvin House, 76 Mill Lane, Carshalton, Surrey SM5 2JR
Tel: 020 8647 7033
Fax: 020 8773 0165
Email: ior@ior.co.uk
Internet Home Page: http://www.Ior.org.uk

The IoR is the professional body of the refrigeration industry. It provides information to the industry through published papers, seminars, Codes of Practice, etc.

Local Refrigeration Societies

Local societies run monthly meetings in their areas throughout the winter period. They are usually of a practical nature, aimed at providing information to application, installation and service technicians.

Local societies are run for:

- Scotland – Glasgow;
- South West – Bristol;
- Northern Ireland – Belfast;
- East Midlands – Grimsby;
- East Anglia – Norwich;
- North West – Liverpool;
- Yorkshire – Huddersfield;
- London

Contact the IoR for details.

D

SPECIFICATIONS, CODES OF PRACTICE AND REFERENCE SOURCES

Publications from the Institute of Refrigeration

See Appendix C for IoR contact details

Safety Code for Compression Refrigerating Systems Utilising Ammonia
Part 1: Design and Construction.

Safety Code for Compression Refrigerating Systems Utilising Ammonia
Part 2: Commissioning, Inspection and Maintenance.

Safety Code for Compression Refrigerating Systems Utilising Chlorofluorocarbons
Part 1: Design and Construction.

Safety Code for Compression Refrigerating Systems Utilising Chlorofluorocarbons
Part 2: Commissioning, Inspection and Maintenance.

Cold Store Code of Practice
Part 1: Design and Construction of Cold Store Envelopes.

Cold Store Code of Practice
Part 2: Design and Construction of Refrigerating Systems.

Code of Practice for the Minimisation of Refrigerant Emissions from Refrigeration Systems.

Also sets of technical papers covering the proceedings of past seminars and conferences.

British Standards

See Appendix C for BSI contact details

BS4434: 1995	Safety and environmental aspects in the design, construction and installation of refrigerating appliances and systems.
BS7671: 1992	Requirements for electrical installations – IEE Wiring Regulations, Sixteenth Edition.
BS EN 60529: 1992	Specification for degrees of protection provided by enclosures (IP code).

ASHRAE Guides

ASHRAE produces handbooks (in hard copy and electronic versions) covering all aspects of refrigeration and air conditioning and self-directed learning courses. They are available to non members. Products include:

See Appendix C for ASHRAE contact details

1997 Handbook – Fundamentals (SI)

1998 Handbook – Refrigeration (SI)

1999 Handbook – HVAC Applications (SI)

BRA Publications

Guideline Methods of Calculating TEWI

The BRA also publishes Fact Finder sheets dealing with topical issues and recommended procedures.

See Appendix C for BRA contact details

CIBSE Guides

A series of substantial reference guides is available from CIBSE, with more in preparation.

See Appendix C for CIBSE contact details

Trade Journals

Refrigeration and Air Conditioning (RAC)

Published by:
EMAP Business Communications, 19th Floor, Leon House, 233 High Street, Croydon CR0 9XT
Tel: 020 8277 5412
Fax: 020 8277 5434
Email: AndrewB@Trenton.emap.co.uk (for the editor).

AC and R News

Published by:
Faversham House Group, 232a Addington Road, South Croydon, Surrey CR2 8LE
Tel: 020 8651 7100
Fax: 020 8651 7117
Email: paul@fav-house.com (for the editor).

ACR Today

Published by:
Battlepress Ltd, Nithsdale House, 159 Cambridge Street, Aylesbury, Bucks HP20 1BQ
Tel: 01296 425151
Fax: 01296 435091
Email: info@aydee.com

The Government's Energy Efficiency Best Practice Programme provides impartial, authoritative information on energy efficiency techniques and technologies in industry, transport and buildings. This information is disseminated through publications, videos and software, together with seminars, workshops and other events. Publications within the Best Practice Programme are shown opposite.

Further information

For buildings-related topics please contact:
Enquiries Bureau
BRECSU
Building Research Establishment
Garston, Watford, WD2 7JR
Tel 01923 664258
Fax 01923 664787
E-mail brecsuenq@bre.co.uk

For industrial and transport topics please contact:
Energy Efficiency Enquiries Bureau
ETSU
Harwell, Didcot, Oxfordshire,
OX11 0RA
Fax 01235 433066
Helpline Tel 0800 585794
Helpline E-mail etbppenvhelp@aeat.co.uk

Energy Consumption Guides: compare energy use in specific processes, operations, plant and building types.

Good Practice: promotes proven energy efficient techniques through Guides and Case Studies.

New Practice: monitors first commercial applications of new energy efficiency measures.

Future Practice: reports on joint R & D ventures into new energy efficiency measures.

General Information: describes concepts and approaches yet to be fully established as good practice.

Fuel Efficiency Booklets: give detailed information on specific technologies and techniques.

Energy Efficiency in Buildings: helps new energy managers understand the use and costs of heating, lighting etc.