Guide to Good Commercial Refrigeration Practice

Part 2

System Design and Component Selection

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2 SYSTEM DESIGN AND COMPONENT SELECTION

2.1 INTRODUCTION

2.1.1 General

System design is a complex subject made even more so in recent years by environmental matters and the safety legislation affecting the industry. Safety is paramount, all designs should conform to all regulations and Codes of Practice. (See Section 3)

The following points need to be considered:

1. Is the capital cost of the system a reasonable cost taking into account energy efficiency and total cost of ownership?
2. Will the system design be as simple as possible with due regard to the task it will be carrying out?
3. Will the equipment be reliable and give many years of trouble-free operation?

The essence of good design is simplicity – over-complication may lead to unreliable operation. It is important that before design work is even attempted, the design engineer is aware of all the information regarding the application to enable him/her to consider all aspects so that the client receives the most cost-effective and efficient result and receives a system meeting their operational requirements.

The vast majority of commercial systems are of the direct expansion air-cooled type and the design points refer primarily to this type of system.

Indirect systems are sometimes proposed for supermarkets and a number of these systems have been successfully installed. An indirect system, or secondary loop system, utilises a brine or other secondary fluid which is cooled by a liquid chiller. Ammonia or hydrocarbon could be used as the refrigerant in the chiller. The brine is circulated to coolers within the cool rooms or cabinets. The liquid chiller would normally be installed as a factory assembled component. Chiller and secondary refrigerant system designs are not covered by this guide.

In recent years the phase down of high global warming potential (GWP) refrigerants has resulted in the introduction of several new fluorinated refrigerants and blends. These offer much lower global warming potential but several of the new fluids are flammable. This has resulted in a major revision of the safety and classification standards, introducing a new class of "lower flammability" refrigerant. The new flammability class is called "2L". Guidance can be found in *Safety Code of Practice: Flammable Lower Toxicity Refrigerants (A2L, A2 and A3)*.

Carbon dioxide (CO_2) systems and flammable refrigerant systems have been successfully employed. These refrigerants requires special considerations, and may be employed either in direct expansion mode or as a secondary coolant. Detailed system specifications must be obtained from the installers of such systems, and guidance can be found in the Institute of Refrigeration (IOR) *Safety Code of Practice: Carbon Dioxide Refrigerant*.

2.1.2 Efficiency

The electrical energy consumed by refrigeration equipment in the United Kingdom is estimated at 17% of the national total and it is most important that energy is used efficiently because of its environmental consequences and cost. The following are the key steps for energy efficiency and reduction of environmental impact:

1. Reduce the cooling load. Examples: better insulation, pre-chill fresh air.
2. Correct system design. Examples: maximise suction pressure, split loads on to different suction levels, consider two-stage or economised cycles.
3. Correct control philosophy. Examples: consider conditions at which most running hours occur, avoid head pressure control, avoid fixed speed auxiliaries.

4. Optimise selection of components including compressors, heat exchangers, and refrigerant to ensure that they are suited to the overall design.
5. Proper operation and maintenance. Example: fix refrigerant circuit leaks so that proper charge level is maintained, ensure heat exchange surfaces are not dirty/fouled, etc.

2.1.3 Coefficient of performance (COP) and system efficiency index (SEI)

COP is a method of comparing the efficiency of refrigeration systems.

$$COP = \frac{\text{Refrigeration Capacity}}{\text{Power Input}}$$

The compressor COP definition simply relates to the cooling capacity and power input of the compressor, taking a simple cycle with no pressure drops or non-useful heat gains. A more meaningful definition is the Coefficient of System Performance (COSP) which takes actual evaporator cooling capacity and includes power input to auxiliaries such as fans and pumps.

The System Efficiency Index (SEI) is the ratio of COSP to the ideal COP for the process temperatures (load temperature and outside air temperature). A rating model for SEI is available from the IOR.

2.1.4 Methods of maximising coefficient of system performance (COSP)

The ratio of the evaporating and condensing pressures should be as low as possible. This increases the capacity and reduces the power required at the compressor. Minimising the compression ratio can be achieved by a combination of:

- The use of larger condensers.
- The use of larger evaporators.
- Low pressure drops in pipework and components.
- The non-useful heat gain between the evaporator and compressor should be kept as low as possible because it reduces effective compressor capacity and results in unnecessary power consumption. The non-useful superheat can be reduced by use of close-coupled systems and insulation of suction lines.
- Sub-cooling contributes to capacity without demanding more power at the compressor. Sub-cooling can be maximised by increasing condenser size to include sub-cooling coils but this will preclude the use of a liquid receiver. A suction/liquid heat exchanger is a useful way of reducing the heat loss/gain effects arising from long unavoidably long suction and liquid lines.
- Correct compressor choice (see section 2.2.1)
- COP can be improved with enhanced cycles such as economised/two-stage systems particularly for low evaporating temperatures. These solutions require proper calculation for optimum results, and software to assist with this is available from a number of sources.

2.1.5 Annual running cost

Low outdoor temperatures make up a high proportion of the UK annual temperature profile, and the hours spent running at the design condensing condition will be very few, and may be zero in some years. Most of the running hours will be at moderate condensing temperatures well below the design maximum. UK installations should be optimised for air temperatures in the 10°C to 15°C range, whilst retaining the ability to operate and deliver sufficient capacity at an outdoor air temperature of, say, 35°C. The efficiency comparison of compressor options should be made at realistic running conditions.

To gain maximum benefit from the outdoor air temperature profile, it is necessary to consider fan power in addition to the condensing temperature effect and the compressor characteristics. It may be advantageous to reduce fan speed or turn off condenser fans at low ambient conditions, in preference to allowing further condensing temperature reduction. Incremental compressor power reduction below 20°C condensing can be quite small. In other words it often becomes more effective (in overall COSP) to reduce the condenser fan power input.

2.1.6 Total equivalent warming impact (TEWI)

TEWI is used as a measure of the lifetime global warming impact of a system. It includes the effects of refrigerant leakage, refrigerant recovery losses and energy consumption. The largest element of the TEWI for the vast majority of refrigeration systems is energy consumption. TEWI should be calculated when comparing system design options for specific applications, and it should be noted that the load pattern must be identical for each system to be compared if a realistic comparison is to be made. Comprehensive method details with calculation examples are given in the BRA/IOR TEWI *Guidelines for Calculation*. It is important to use an up to date emissions factor, also known as Grid Carbon Intensity, which can be found online.

2.2 COMPRESSORS

2.2.1 General

Compressor performance data is normally given for a range of evaporating and condensing temperatures. The parameters used to select the compressor apply to the conditions at the compressor suction and discharge service valves. Allowance must be made for pressure drops between the evaporator and the compressor, and between the compressor and the condenser. The suction line pressure drop is the more important. If this is not taken into account the compressor can be undersized.

Rating conditions for all refrigerants should be checked with the manufacturer.

2.2.2 Suction return temperature and superheat

For compressor rating purposes all suction superheat is usefully obtained in the evaporator, i.e. contributes to the stated capacity. The actual operating conditions will vary from these rating conditions, therefore corrections must be applied to calculate the actual performance on any application.

In a real system the useful superheat would typically be between 5 and 10K (the superheat setting of the valve) and allowance must therefore be made for:

- the actual suction return temperature at the compressor
- the non-useful superheat

Compressor manufacturers’ software usually enables these corrections to be made. A reduction in useful superheat will tend to reduce the compressor capacity from that given at the rating condition, but without reducing the power input. The system efficiency will therefore be lower than the compressor efficiency indicated in the manufacturer’s published data. When making efficiency comparisons for different compressors, it is very important that identical operating conditions are used.

2.2.3 Limits of operation

The maximum and minimum pressures and temperatures shown in compressor performance data must be strictly adhered to. The maximum evaporating and condensing temperatures are usually determined by the motor capability. If exceeded, the motor may overheat, reducing its reliability and increasing power consumption.

The minimum evaporating temperature at any particular condensing temperature is usually fixed by the maximum discharge temperature limit. It may also be necessary to sustain sufficient mass flow to cool the motor. Vacuum operation should be avoided due to the possibility of air ingress.

For a given capacity and operating condition there may be a choice of model type. Some types are optimised for medium temperature applications and some for low temperature applications. A smaller motor may be used for lower evaporating temperatures. In addition to the primary requirement of delivering the necessary capacity at the design conditions, care should be taken to investigate operation at all likely conditions and efficiency at typical running conditions. Cost will also be a consideration.

Maximum suction return temperatures are often given. If these are exceeded, the discharge temperature may become too high and cause a high temperature trip. Without discharge temperature protection excessive temperatures cause oil breakdown and lubrication problems. This can result in partial or total seizure, thus increasing power consumption or causing complete compressor failure. The manufacturer's suction return gas superheat limits usually indicate where additional compressor cooling, such as a fan may be needed to avoid the danger of exceeding the operating limits.

The application of a capacity controlled compressor may be more restricted than the same machine operating on full load. This should be checked when selecting compressors for this type of application.

2.2.4 Sub-cooling

Some degree of sub-cooling is always necessary to avoid flash gas in liquid lines and the sub-cooling can vary depending on length of liquid lines. This sub-cooling increases the system capacity without increasing the power consumption. To assess the correct capacity and efficiency, the effect of sub-cooling must be known. Corrections for sub-cooling can usually be found in manufacturer's software. Further sub-cooling, sometimes termed "mechanical sub-cooling" may be provided by an additional heat exchanger with two-stage or economised systems (see 2.2.5), or by a suction/liquid heat exchanger. In the latter case the superheat will also be affected and the evaporator capacity is determined by the enthalpy difference between the expansion valve inlet and the evaporator outlet (not the heat exchanger outlet). CO_2 booster applications are more complex and require further consideration.

When calculating the capacity gain resulting from liquid sub-cooling it is important that the actual liquid temperature at the expansion valve entry is used, not the temperature of liquid leaving the sub-cooler.

2.2.5 Liquid injected and economised compressors

Some low temperature compressors incorporate liquid injection. This is to cool the compressor and allow operation over a wide range of condensing temperatures. Data supplied by the manufacturer will be dependent on use of the recommended liquid injection control method, and the availability of suitable sub-cooled liquid supply.

Economised scroll or screw compressors require an additional heat exchanger, usually a plate heat exchanger with a liquid stream expanding in the heat exchanger. This is used to sub-cool the working refrigerant before it is sent to the expansion valve(s) and evaporator(s). The vapour generated in this sub-cooling process is fed to an intermediate, or economiser port on the compressor. This process results in a significant gain in compressor capacity and COP. It is essential that the liquid reaching the heat exchanger expansion device is pre-subcooled to some extent. Natural sub-cooling in the line from the condenser will normally suffice, provided the system is correctly charged.

Data supplied by the manufacturer will refer to the liquid temperature at the outlet of the economiser heat exchanger, which may be considerably lower than the surrounding temperature. It is therefore essential to insulate the liquid lines from the heat exchanger to the expansion valves. Corrections to the published data may be necessary to allow for liquid temperature gain.

2.2.6 Efficiency

Users can either compare information from manufacturers or can obtain comparative information from the Energy Technology List ETL. Manufacturer's ASERCOM certified data has been independently verified. Energy efficient products are listed on the ETL website www.etl.beis.gov.uk. The performance of all compressors listed on the ETL has also been independently verified.

2.2.7 Additional important considerations

The performance, efficiency and operating range will be different for a compressor operating on capacity control. This must be examined when selections and comparisons are made.

Supplementary compressor cooling e.g. air fan or liquid injection may be necessary for some applications to reduce discharge temperatures. Such additional cooling must be fitted when recommended by the compressor manufacturer to maintain efficiency and reliability.

Crankcase heaters may be necessary to prevent liquid migration to the compressor during its off cycle. If such heaters are not fitted and correctly used, accelerated bearing wear can occur, reducing reliability and efficiency.

Where reduced starting current methods are used – star delta and part wind, for example – some method of reducing the starting load may be required. This prevents the compressor overload when starting and stalling, thus effectively starting direct on line and drawing the locked rotor current for a longer duration. Some compressor types have an internal bypass valve to route the compressor's discharge gas back into its suction via a solenoid valve just for the short start duration. Manufacturer's recommendations regarding starting should be followed.

System pressure switch settings should ensure that the compressor is prevented from operating outside its design range and an oil differential switch is normally applied where a high-pressure lubrication system (oil pump) is used in the compressor.

2.3 EVAPORATORS

2.3.1 General

Correctly sizing the evaporator is important, both to ensure that it will achieve the required air off temperature and the required space or product temperature. Maintenance of an acceptable level of humidity may be a consideration, but it is not so common in commercial applications, where selection is usually based on a standard humidity or SHR. The space or product temperature will have been specified by the user. Generally, a specific humidity level only needs to be considered for fresh unwrapped product and is primarily affected by the difference between the evaporating temperature and the air on/air off temperature – the higher the difference the more moisture is removed.

The following factors affect the efficient operation of an evaporator and need to be addressed when selecting a model to match a particular application:

- fin spacing
- air velocity
- method of defrosting, where necessary
- air on humidity
- large/small "air on" to "evaporator" temperature differences
- door openings to space
- refrigerant glide

FIN SPACING Fin spacing vary from 3mm (6 fins/inch) to 12mm (2 fins/inch). It is necessary to use the wider fin spacing on applications where there will be a build up of frost due to low evaporating temperatures, or where there could be airborne contamination. However, a narrower fin spacing gives a greater heat transfer surface area and hence a high duty, but would require more frequent defrosting on low-temperature applications.

AIR VELOCITY Low air velocity evaporators are used in areas which are occupied and to cool delicate produce. High velocities are typically used for blast freezing. The greater the air velocity over a given fin block the greater will be the cooling capacity of the evaporator. The air velocity must not be too high or it will blow free water into the cold room or remove moisture from the product.

DEFROSTING The evaporator must be selected to suit the type of defrosting operation that has been specified in the system design. Refrigerant pumpdown is required before defrost and snap cooling of the coil is required to freeze free water on to the fins prior to starting the fans after defrosting coil.

2.3.2 Evaporating temperature

Taking the air on to the evaporator temperature as the reference (room temperature), the maximum difference between this value and the product temperature should be selected as follows:

3K- products such as fruit and vegetables that dehydrate quickly (not normally possible with standard commercial evaporators)

5K- products needing less than 80% relative humidity

10K- products which will not dehydrate

A system with a higher evaporating temperature will operate more efficiently but may require a larger evaporator which will increase the initial capital cost of the installation. By using a larger evaporator, compressor size could be reduced, thus reducing overall cost. Where there is no specific humidity requirement, evaporators should be selected on a 6–7K temperature difference to optimize efficiency.

2.3.3 Evaporator selection methods

The most common method for selection of commercial evaporators is by use of suppliers' selection software. The required capacity, DT1 and either evaporating or air on temperature are entered.

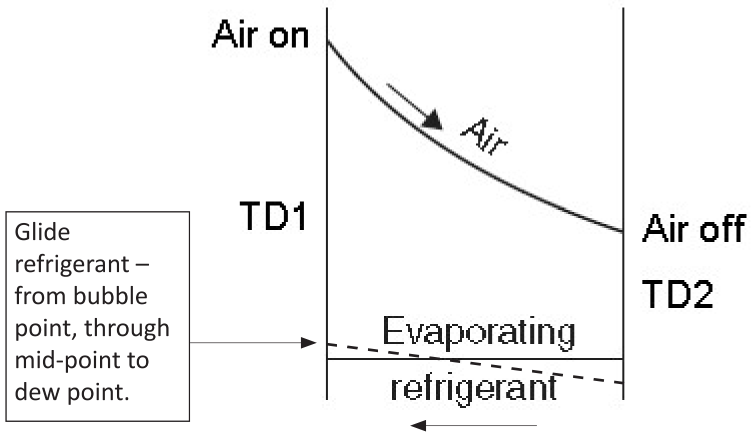


Figure 2.1: Temperature differences

Care must be taken when using refrigerants with glide due to the different evaporator size that will result from using the wrong evaporating value. This is particularly true when using a DT1 value that relates to experience or a specification for a non-glide refrigerant – mid-point must be used for the new selection. Using dew point would result in a smaller cooler with high DT1 and high moisture extraction, not enough surface for frosting and poor system efficiency.

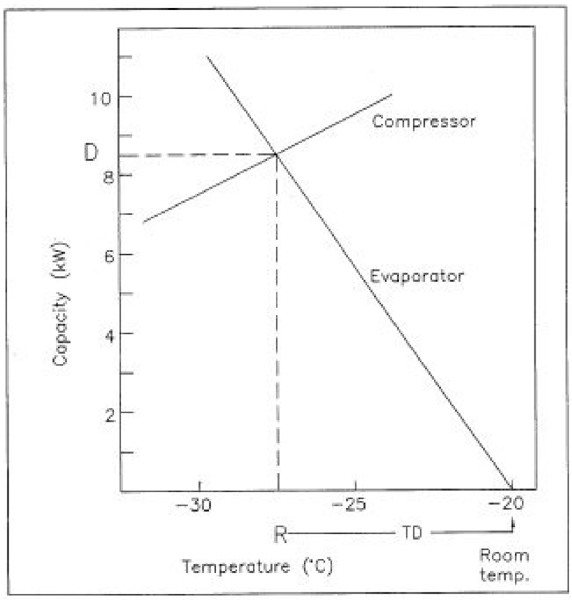
When using thermostatic expansion valves, it is important to consider the minimum superheat required for the valve to control accurately. This has to be calculated using the dew-point TD1.

Mean air temperature is rarely used for cooler selection and is not included in EUROVENT certified data or EN evaporator performance testing standards.

2.3.4 Actual evaporator selection – equipment balance

When sufficient information has been obtained to allow an evaporator to be selected, its actual operating conditions can be established by using a suitable computer programme or by constructing a balance graph with the compressor.

By incorporating different sized evaporators into a balance graph, a comparison of the operating conditions can be obtained, see Figure 2.2.



Balance Graph at Maximum Ambient Condition
(Drawn from compressor and evaporator performance data)

D = Duty of balanced equipment
R = Refrigerant temperature (evaporating temperature)
TD = Temperature difference between refrigerant and air return to the evaporator

Figure 2.2: Equipment balance graph example

It should be noted that the equipment illustrated in Figure 2.2 will only remain in balance if the conditions illustrated are prevailing. If the ambient temperature falls and causes the condensing unit/compressor duty to increase then a new balance will occur with the temperature difference increasing so that the duty of the evaporator matches that of the condensing unit/compressor.

Due allowance should be made when carrying out the above procedure for suction line pressure drop. The following methods to overcome the above should be considered:

(i) The Temperature Difference (TD) between the room temperature and the compressor saturated suction temperature should allow 1 or 2K for pressure drop in the suction line, i.e. for a room temperature of -20°C, assuming a suction line pressure drop equivalent to 2K and an evaporator TD of 6K the compressor saturated suction temperature would be -28°C.

(ii) The resulting (reduced) compressor capacity should be plotted against the actual evaporating temperature and TD -26°C and 6K.

2.3.5 Positioning of evaporators within the coldstore

Air circulation and the velocity of the air circulated contributes directly to the overall system performance. It is important that good air circulation is available. It may be economical to select for example two evaporators to produce a certain duty for a condition but four evaporators could give better air circulation. The number of evaporators selected must represent the most effective air circulation performance. In some instances the evaporator fan 'air throw' may be too small for the size of the coldstore and in this instance supplementary air circulation fans may have to be considered. These would produce additional heat that needs to be taken into account.

The following points should also be considered:

- Install evaporators opposite entrance doors
- Direct air flows along gangways and shelving
- Do not install transverse support steel work which can block air flow at high level
- Ensure free air flow over guards, i.e. do not stack too high
- Catalogue values for air throw will be for ideal conditions

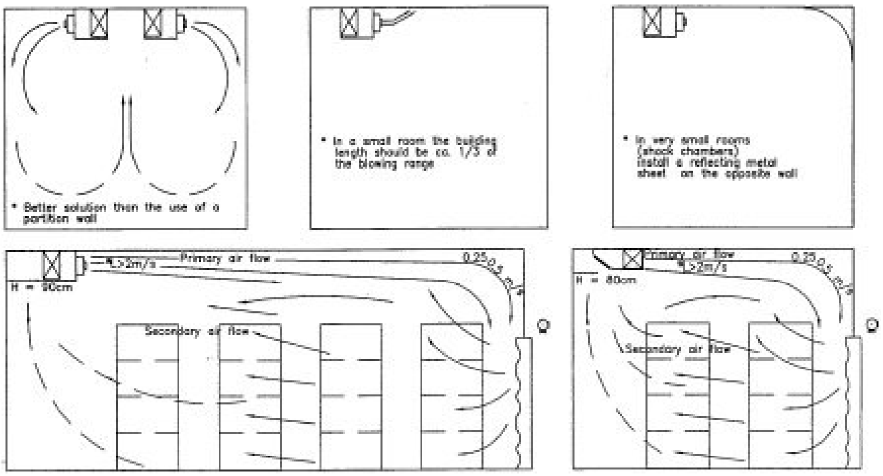


Figure 2.3: Examples of correct evaporator installation in cold stores

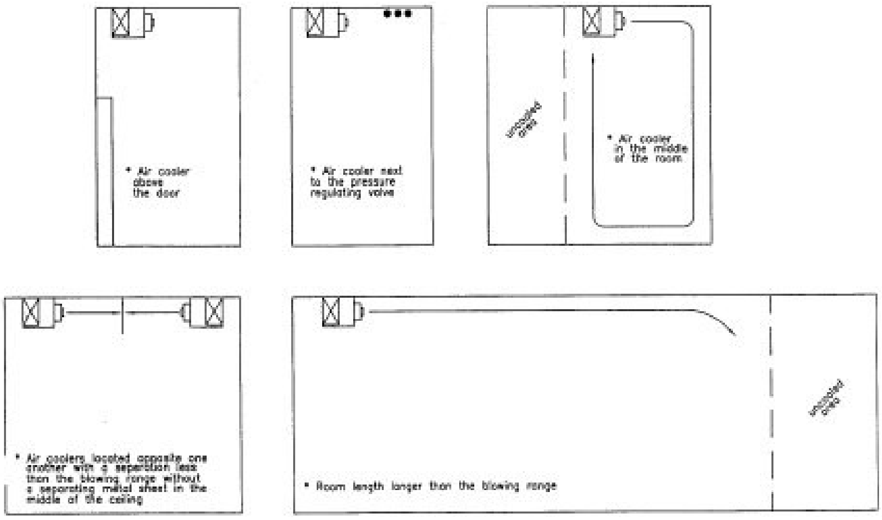


Figure 2.4: Examples of incorrect evaporator installation in cold stores

2.3.6 Additional important considerations

Evaporators in commercial coldrooms should not be defrosted separately regardless of the number of coolers in the room. If the evaporators are defrosted at different times moisture generated during defrost will migrate to the other units.

2.4 CHILLED AND FROZEN FOOD CABINETS

2.4.1 General

Selection of a retail display cabinet is dependent on a number of factors. The cabinet should be selected on its ability to perform the function required (in terms of temperature control, amount and the way food can be displayed and the overall usability of the cabinet). Additional features such as energy consumption, flexibility, life of the cabinet and hygiene and safety should also be considered.

2.4.2 Test standards and published information

In Europe the majority of cabinets are tested to BS EN ISO 23953 parts 1 and 2[1] (formerly EN441). This provides a measure of the temperature control and energy usage of the cabinet. In this

1 BS EN ISO 23953-1:2015 *Refrigerated Display Cabinets. Vocabulary.* BS EN ISO 23953-2:2015 *Refrigerated Display Cabinets. Classification, Requirements and Test Conditions.*

standard, temperature control is divided into several levels as shown in Table 1. It is usual for manufacturers to state that their cabinet complies with one of these temperature levels, although sometimes tighter temperature bands are applied to cabinets designed for special functions.

Table 1: Temperature levels defined in BS EN ISO 23953

Class	Highest temperature of warmest M-package[2] less than or equal to	Lowest temperature of coldest M-package greater than or equal to	Lowest temperature of warmest M-package greater than or equal to
	°C		
L1	-15		-18
L2	-12		-18
L3	-15		-15
M0	4	-1	
M1	5	-1	
M2	7	-1	
H1	10	1	
H2	10	1	
S	Special classification		

BS EN ISO 23953 also defines a method to measure energy consumption. For integral cabinets (those operating directly from a plug and with a refrigeration system contained on board) the energy directly used by the cabinet is stated (termed Total Energy Consumption (TEC)).

For remotely operated cabinets the heat extracted by the evaporator is converted to an equivalent electrical power consumption called Refrigeration Energy Consumption (REC). The conversion is based on typical efficiencies of the refrigeration circuits to which they will be attached. The direct energy used by fans, lights, defrost heater etc. are then added to provide the Direct Energy Consumption (DEC). DEC + REC equals the TEC.

For remote cabinets the majority of manufacturers define REC and DEC separately. This is because in most large supermarkets the energy used to extract heat from the evaporator will be part of the refrigeration pack electrical load and the direct energy will be part of the shop services electrical load.

The heat extracted by the evaporator at a defined evaporating temperature may be supplied in addition to, or as an alternative to REC.

The ambient conditions which cabinets are tested in are defined in the standard. These conditions are the conditions at which the ratings apply. For retail cabinets designed for supermarkets the majority of cabinets are tested in climate class 3 conditions (see Table 2). In supermarkets the ambient conditions are generally around 20–21°C and are lower in humidity than those used to test the cabinets. This may lead to a lower heat extraction rate in reality than in the tests and it is common for manufacturers to make an adjustment for this.

For commercial service cabinets for kitchens where the ambient conditions are warmer the climate class conditions are generally climate class 4.

Table 2: Climate class levels defined in BS EN ISO 23953

Test room climate class	Dry bulb temperature °C	Relative humidity	Dew point	Water vapour mass in dry air g/kg
0	20	50	9.3	7.3
1	16	80	12.6	9.1
2	22	65	15.2	10.8
3	25	60	16.7	12.0
4	30	55	20.0	14.8
5	27	70	21.1	15.8
6	40	40	23.9	18.8
7	35	75	30.0	27.3
8	23.9	55	14.3	10.2

2.4.3 Practical considerations

The conditions defined in the standard require lighting to be on for 12 hours out of every 24 hour period. In modern supermarkets trading 24 hours a day it should be noted that lighting load is likely to be higher than this. For cabinets with doors the standard defines door openings over a 12 hour period. Actual usage may be more severe, or in some situations less severe, and therefore the location and type of supermarket needs to be taken into account and the heat extraction rate for cabinets with doors adjusted accordingly.

A number of display cases can either be connected to one condensing unit, one stub, or branch of a central plant installation. Each of the cabinet extraction rates are added together to make the total load for the condensing unit or stub or branch. In such a system the compressor(s) must operate to achieve the lowest evaporating pressure required by the attached cabinets. Therefore, care needs to be taken to ensure that a few cabinets do not require a lower evaporating temperature than others on the same pack as this will raise energy consumption unnecessarily.

In addition, as most cabinets in UK supermarkets are not operated with evaporator pressure regulators, the cabinets must operate at the pack pressure and this may differ from the cabinet design conditions.

2.4.4 Choice of cabinet

Cabinets are usually selected to store a certain food in a certain manner. The location of the cabinet and size of the retail outlet usually define whether a remotely operated or integral cabinet is preferable. Whether the food stored in the cabinet should be chilled or frozen is also usually clear. Once these are defined the type of cabinet selected is dependent on how food will be merchandised, the available area to locate the cabinet and any other features such as energy consumption.

2.4.5 Performance and energy efficiency

Once a cabinet type has been determined users can either compare information from manufacturers or can obtain comparative information from sources such as Eurovent or the ETL. Eurovent (www.eurovent-certification.com/index.php.en) is a trade scheme that certifies the performance ratings of refrigeration products according to European and international standards. Manufacturers can list products if they are members of the scheme and a certain number of products are independently verified each year. The ETL is a UK Government operated scheme that identifies energy efficient products and lists them on the ETL www.etl.beis.gov.uk. The performance of all retail cabinets listed on the ETL should be independently verified.

Both the above schemes use BS EN ISO 23953 to define performance of cabinets. The overall performance is usually listed as the TEC divided by the Total Display Area (TDA). TDA is a measure of the visible area of the food within the cabinet; a measure of the merchandising field of view that the consumer sees. It should be noted that TDA is not a good reflection of cabinet volume. If storage volume is important in a cabinet selection then TDA should not be considered in isolation.

Even though such schemes help identify the best performing cabinets there is still a large variation in performance of cabinets listed on such schemes. By selecting the very best performing cabinets considerable energy savings can be achieved.

Figure 2.5 presents data from the ETL showing TDA against TEC for M1 and M2 cabinets. It can be seen that in certain instances there is considerable differences between cabinets and that by careful selection of a cabinet considerable energy savings can be achieved.

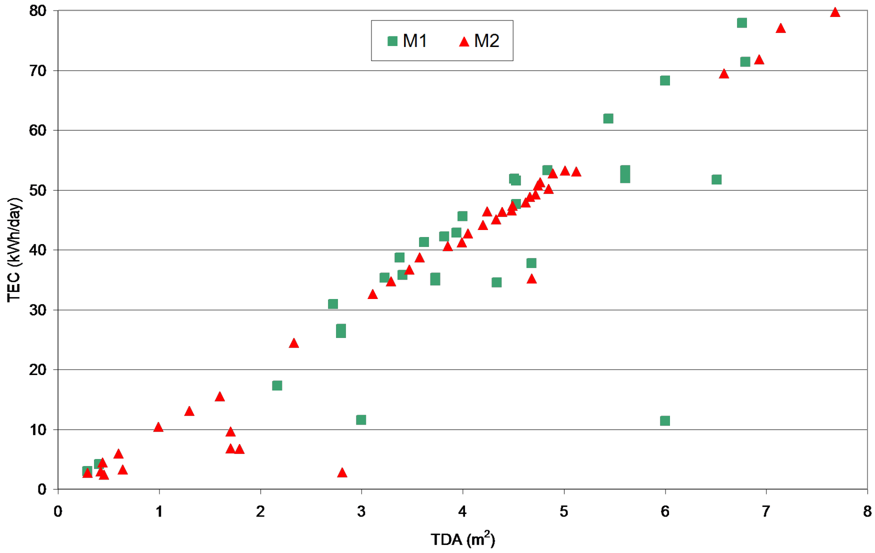


Figure 2.5: Data from ETL for M1 and M2 cabinets (January 08)

2.4.6 Additional energy efficiency features to be considered

Table 3: Features and benefits

Technology	Application	Notes
Adding doors to display cases	Chilled and frozen multi-decks	Reduced refrigeration heat load
Anti sweat heater controls	Freezer cabinets	Savings dependent on store conditions
Defrost optimisation	Freezers (chillers should be operated on off-cycle)	May be part of supermarket control system rather than cabinet itself
Efficient 'standard' lighting system	All cabinets with lights	Most options already used
Evaporator coil rifling	All cabinets	Enhanced heat transfer, Increased UA
Evaporator fan ECM motors and high-efficiency fan blades	All cabinets with forced air convection	High efficiency fans may have additional benefits such as improved air distribution
LED lighting	All cabinets with lights	Saving depends on number of lights and radiant effect of 'conventional' lighting
Liquid-suction heat exchangers	All cabinets	
Low emissivity/reflective glazing ('K' glass)	Cabinets with glass (half glass doors, full glass doors, delicatessen)	Reduction in radiant heat gain

Technology	Application	Notes
Night blinds	Chilled multi-decks, well freezers	Assumes retail outlet closes for part of day, correct fitting is essential to capitalise on energy saving
Night covers	Chilled multi-decks, well freezers	Assumes retail outlet closes for part of day, correct fitting is essential to capitalise on energy saving
Optimisation of air curtain	Chilled multi-decks, well freezers	Use of weir plates or shelf edge air flow technologies to reduce infiltration, cabinet with low infiltration will have low heat extraction
Strip curtains	Chilled multi-decks	Assumes correct fitting, energy saving depends on usage
Thicker/better Insulation	All cabinets, primarily frozen	

2.5 CONDENSERS

2.5.1 General

There are two types of condenser used in the commercial refrigeration industry:

- air cooled
- water cooled

The condenser must be sized to reject all of the heat absorbed by the refrigeration system at all of the plant’s operating conditions, so it is important that this is correctly calculated for accurate condenser selection. Heat rejection consists of:

Compressor capacity + Power absorbed by the compressor – heat losses

Note that the actual compressor capacity for condenser selection also incorporates non-useful superheat. Heat losses represent heat dissipation from the compressor body, and this is normally less than 10% of the power input. This figure can be higher in the case of air cooled compressors. In many cases the heat rejection can be found from compressor manufacturer’s software.

It is important that the total heat of rejection is also checked at conditions other than the basic design condition. For example applications that involve artificial forms of defrost or blast freezer/ blast chiller applications i.e. where a rise in suction pressure may occur, which can give rise to increased capacity and hence increased heat rejection.

Defrost and blast freezer/blast chiller applications will, during the process, cause evaporating temperature rise and therefore the equipment will have a larger duty. It is important that this increase in the duty of the equipment is taken into account during the selection of the condenser.

Generally the largest practical condenser should be selected since the energy efficiency is higher at the lower condensing pressures. The operating costs are very important when selecting condensers – a higher investment in a larger model can usually be offset by a reduction in the running costs within a reasonable period. However, if with an air cooled condenser, the larger model has more fans, then the extra power required to run them may severely reduce or even cancel any saving that is to be gained from the more efficient operation of the compressor.

Typically maximum condensing temperatures should not be more than 40–43°C. It is more efficient to increase the condenser size to reduce condensing temperature than to incorporate sub-cooling coils.

2.5.2 Air cooled condensers

The performance of an air cooled condenser will be a function of the following parameters:

- size of the fin block
- fin spacing (usually 2–3 mm) and type of fin coating
- fin block material and coating
- air flow through the fin block
- refrigerant
- temperature of the air onto the condenser, including air recirculation
- cleanliness

Some condensers are designed to incorporate water spray for adiabatic cooling

Data will be available from manufacturers giving the capacity of a condenser for a particular refrigerant, as a function of the TD of the air into the fin block and the refrigerant condensing dew point temperature. The capacity of different condenser models will be affected by the first four factors listed above. Users can either compare information from manufacturers or can obtain comparative information from sources such as Eurovent. Eurovent (www.eurovent-certification.com/index.php.en) is a trade scheme that certifies the performance ratings of refrigeration products according to European and international standards. Manufacturers can list products if they are members of the scheme and a certain number of products are independently verified each year.

As with evaporators, care must be taken when using glide refrigerants. However, with condensers the consequences are not significant unless the TD is below 10K.

2.5.3 Maintenance of discharge pressure

Expansion valves that will operate effectively over the full range of desired condensing temperatures should be chosen. If the valve requires a minimum pressure drop to operate effectively then the condensing pressure can be regulated to achieve this. But the additional compressor running cost resulting from maintaining head pressure in this way needs to be considered in conjunction with an alternative valve option.

Electronic expansion valves and balance port thermostatic expansion valves are not so reliant on pressure drop and therefore systems may be able to operate at lower condenser pressures/temperatures. With these types of valves it is acceptable during low ambient conditions to aim for discharge pressure equivalent to 20°C condensing temperature for systems operating at high temperatures, and discharge pressure equivalent to 15°C for those operating on low temperature.

Some thermostatic expansion valves need a pressure drop across them of 6 bar to operate effectively. To maintain this type of pressure drop the following methods of condenser control should be considered:

1. Pressure switch control of the condenser fans
2. Speed control of the condenser fans
3. Splitting the condenser heat exchange surface area into a number of passes and isolating the pass to reduce the surface area of the condenser.

Where consideration is given to the switching of the condenser fans to maintain discharge pressure, the following should be considered in respect of the control of the refrigeration system. With all fans off, 10–15% of the capacity of the condenser will still be available on high air volume condensers. With low air volume condensers the same heat exchanger may be capable of 30% of the design capacity. The isolated section should have adequate means of transferring refrigerant to the operational part of the system.

Furthermore, and particularly on twin coil condenser models, it is necessary to consider the arrangement of the staging of the condenser fans:

- Always run the fan or fans nearest the inlet and outlet connections first and then cycle the fan next to it and so on.
- On larger (12 fan condensers) the fans can be run in pairs over each coil, on smaller condensers (four fan) it may be preferable to select an 'in line' four fan model.

Care must be taken when operating in low ambient conditions because liquid accumulation can occur in the condenser causing liquid shortage.

2.5.4 Position of remote air cooled condensers

Remote air cooled condensers should not be positioned in full sun on flat roofs as this can add up to 10°C in solar gain and will affect performance. Screens may be used to reduce solar gain.

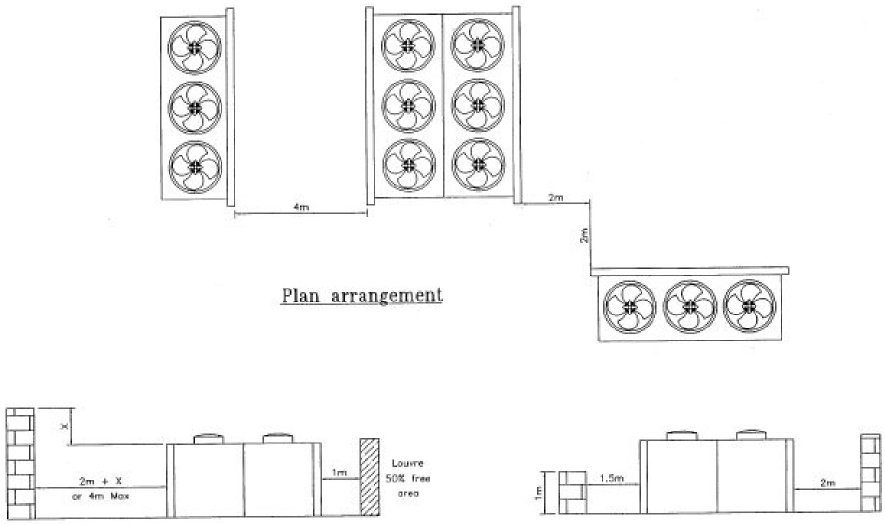


Figure 2.6: Location of remote air cooled condensers

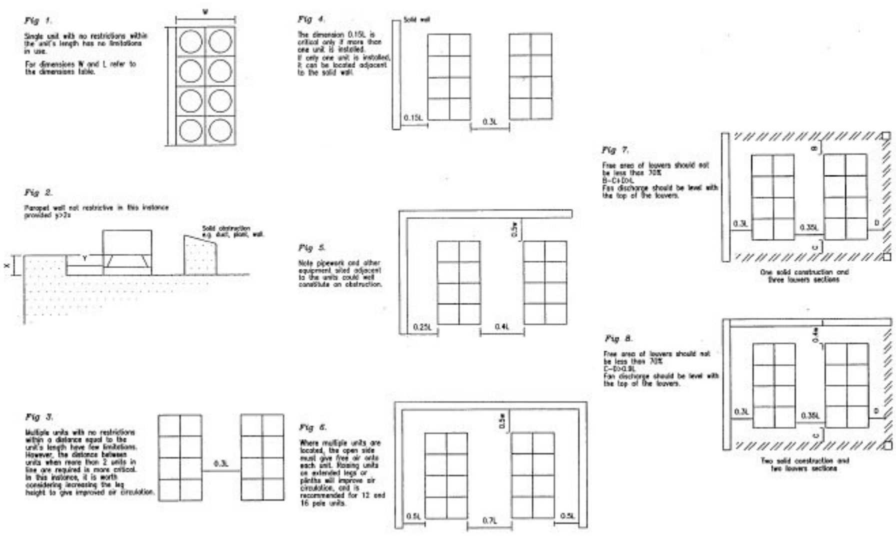


Figure 2.7: Installation and Location of remote air cooled condensers

2.5.5 Fin block material

Aluminium is the most commonly used but corrodes in salt or polluted atmospheres. Some examples of alternatives are:

- Polyester coating
- Copper fins
- Copper fins electro tinned
- Spray-on protective coating

2.5.6 Water cooled condensers

The performance of a water cooled condenser is a function of a number of parameters similar to air cooled models:

- surface area of the tube bundle
- tube design and material – plain or extended surface tubes
- tube size and pitch
- refrigerant
- quantity of water flow
- water temperature entering the condenser
- arrangement of the water side baffling
- number of passes water side
- water fouling factors

Again different models will have capacity ratings which are a function of the first three parameters. The water temperature entering the condenser will depend upon its source. The flow rate will be a function of its supply; either a pump if it is a circulation system using a cooling tower, or the mains pressure.

In general the design engineer has a greater selection with water cooled condensers than with air cooled types, but the same overall factors affect energy efficiency.

Control of head pressure can be achieved by the use of a 3-way valve controlling water flow through the condenser. The valve can either be temperature or pressure controlled.

Evaporative condensers are not generally employed on commercial refrigeration systems. There are several methods of introducing water so that it evaporates to lower the air temperature over the coil. Any such method must utilise purpose designed condensers and water treatment systems to avoid build up of deposits and any risk of legionella.

2.6 EXPANSION VALVES

2.6.1 Introduction

The most general type of expansion device used on commercial refrigeration installations are the standard thermostatic expansion valve (TEV) or electronic expansion valve (EEV). As long as it is properly selected and adjusted for the correct superheat, it is a reliable and efficient control device for systems with reasonably constant loads.

If wide variations in the operating pressures are expected, as with systems having capacity control or a floating head pressure, then an alternative valve having more flexible control characteristics should be used – for example, a balanced port valve or an electronic expansion valve.

The balanced port valve is similar in appearance and operation to a standard TEV. However, its internal design and construction allows it to operate with much closer clearances. This allows the valve to maintain an acceptable level of control on the flow of refrigerant over a much wider range of pressure differentials. The cost compared to a standard valve is typically 20% more.

2.6.2 Thermostatic expansion valve selection

Accurate selection of expansion valves is essential to achieve good system control and optimum efficiency. EEVs give a good performance over a wide range of operating conditions. Special care must be taken when selecting TEVs for systems which operate with capacity control to prevent liquid return to the compressor on part load.

Data on expansion valves for use with a particular refrigerant is usually presented in tabular format covering both a range of evaporating temperatures and high to low side pressure drops. The data is rated for a given superheat and liquid temperature. When operating at different temperatures, correction factors have to be applied.

The pressure drop across the expansion valve must be calculated. This is the difference between the condensing and evaporating pressures taking all the relevant pressure drops into account. It will include losses in pipes and fittings, static pressure drops in vertical liquid lines, and the effect of a distributor if one is used with the evaporator.

The valve's capacity will vary if the liquid refrigerant temperature is different from that of the rated condition. Correction factors are published by the manufacturer, which must be applied to the stated capacity.

An externally equalised valve must be used if the pressure drop through the evaporator exceeds the level specified by the valve manufacturer. This will vary with the evaporating temperature and refrigerant. If the evaporator is fitted with a distributor, the resulting pressure drop will necessitate the fitting of an externally equalised valve.

On evaporators with a low pressure drop there is no disadvantage in specifying an externally equalised valve.

Different phial charges are available depending upon the system and operating conditions. Typical of the variations available are charges which limit the operating pressure (Maximum Operating Pressure (MOP) charge). This will prevent an overload condition at the compressor if peak periods of high load are possible – for instance, just after a defrost operation.

2.6.3 Low ambient conditions

The expansion valve capacity relates to the pressure drop across the valve. In low ambient conditions the pressure drop across the valve is smaller and therefore the duty of the valve is smaller. It is therefore important that when expansion valves are selected, the minimum operating condensing pressure is considered. In some instances this may make the expansion valve twice the size of the one selected for the design ambient conditions but expansion valves can throttle down to, and control, at least 20% of their design capacity.

2.6.4 Capacity control

To prevent liquid returning to the compressor on single evaporator systems an expansion valve with a capacity as close as possible to the full load condition should be selected. Most valves will operate without causing liquid return at capacities down to 20% of their rated full load condition. If variations greater than this are expected, two or more valves fitted in parallel may be required. Alternatively a more versatile type can be used, such as an electronic or balanced port valve.

2.6.5 Additional important considerations

TEVs are available with alternative phial charges and connections, so these must be specified by the system designer, as must the requirement for the valve to be internally or externally equalised.

At their rated capacity, valves are not fully open. This allows some compensation for intermittent problems such as:

- short periods of increased load or low ambient conditions;
- occasional small quantities of flash gas;
- slight shortage of refrigerant;
- minor errors in selection.

Although there is some latitude in the sizing of expansion valves, it is important to ensure that the selection is as accurate as possible. Both over and under sizing a valve can cause inefficiencies.

Over sizing a valve can lead to the following system faults:

- liquid refrigerant returning to the compressor;
- expansion valve hunting.

Liquid refrigerant returning to the compressor will often result in failure or reduced efficiency.

If the valve is too large for the required duty, any adjustment made in an attempt to control the superheat will result in excessive fluctuations in the flow of refrigerant. This will cause alternate flooding and starvation of the refrigerant in the evaporator, and subsequent loss of efficiency. This is known as hunting.

Suction accumulators can be used as a preventative measure to avoid liquid carry over to the compressor.

Under sizing a valve will starve the evaporator of refrigerant as there will be insufficient capacity through the valve to maintain the required exit superheat. This will again reduce the evaporator's capacity.

2.6.6 Electronic expansion valves

Electronic expansion valves are available in two types:

1. Stepper motor
2. Pulse solenoid (Pulse Width Modulated (PWM))

In each case a controller is used in conjunction with the valve. The controller is pre-configured for the refrigerant and valve type and it receives the information from sensors, for example, pressure and temperature at the evaporator outlet. This enables the superheat to be determined. The output signal to the valve initiates the orifice adjustment. The stepper motor arrangement operates on the principle that each individual step of the motor will open the orifice by a certain percentage.

In the case of the PWM valve it is the relationship between the opening and closing which determines the capacity of the valve. The valve is either open or closed and each time interval of a few seconds will include an opening period depending on the signal.

The PWM valve also eliminates the need for a liquid line solenoid valve and in most cases evaporator pressure regulators. Liquid hammer can occur with this type of valve and refrigerant liquid line velocities should be no more than 1m/sec.

The other important advantage in applying electronic expansion valves is that they can operate on any type of refrigerant, therefore if the refrigerant is changed, the valve does not require replacing. They are more costly than TEVs.

The turn-down capability of the electronic expansion valve means that unlike its conventional counterpart, the valve can operate satisfactorily at virtually any condensing pressure, providing it is properly selected. This means that operation with very low condensing pressures is possible (e.g. during the winter months).

2.7 OTHER COMPONENTS

2.7.1 Pressure relief valves

BS EN378-2 sets out in detail requirements for pressure protection. This includes safety valves, bursting discs and other pressure limiting devices, including for example, high pressure cut out switches.

2.7.2 Temperature controls

Temperature controls cover a wide spectrum of types, from the simple bi-metal strip arrangement, through the capillary and bellows type, to the modern electronic type. The main function of a thermostat is to either stop or start a condensing unit or compressor, dependent on the temperature setting. This can be achieved by operating a solenoid such as described under pump down cycle or by directly controlling the operation of the condensing unit/compressor.

Temperature controls are also used for the following purposes:

1. Defrost termination
2. Evaporator fan delay after defrost (to cause a snap freezing of any free water on surface of coil after defrost)
3. Liquid injection control

Capillary temperature controls/thermostats
Capillary temperature controls have a bellows attached to a remote bulb by a length of capillary tube. Bellows, capillary tube and bulb are evacuated and partly or wholly filled with a media (some types with two media).

Increase or decrease of temperature at the bulb causes pressure changes within it. These pressure changes are transmitted through the capillary tube into the bellows located in the thermostat. The bellows expands or contracts as the pressure inside of the bellows changes and moves the mechanism of the electrical contact.

The medium used to convey the temperature change information may vary relative to the requirement, and the temperature range. Sensing bulbs also vary in size and shape dependent on the charge. The types of charge are detailed below.

Electronic controls
Electronic expansion valves are described in section 2.3.2. Case/coldroom controllers consist of small printed circuit boards situated within a small electrically safe protective box. Most controllers need a handheld unit to access the memory. Once this is available it becomes convenient for all the commissioning information and settings to be assigned to the controller; when service is required with the handheld unit in place, access to all data is available.

Display case/coldroom controllers can carry out the following functions:

1. Initiate defrost
2. Terminate defrost
3. Control air on and air off temperature by operation of liquid line solenoid valves
4. Initiate temperature alarms

The type of information that is available from case/coldroom controllers is as follows:

1. Defrost initiation times
2. Defrost termination temperature
3. Air on evaporator temperature
4. Air off evaporator temperature
5. Superheat across the evaporator
6. Alarm temperature settings

Electronic controls offer the following advantages:

a. Energy saving

   Electronic expansion valves can save energy cost because they can operate at lower superheats and therefore the evaporator contains more liquid refrigerant.

   Floating discharge pressure can save up to 30% in energy costs. This is more likely to be achieved using an electronic expansion valve because the pressure difference across the valve has a less significant impact on the valve operating efficiently.

b. Reduction in maintenance costs

   Case/coldroom controllers are convenient to use and give access to information quickly; this reduces maintenance and trouble shooting time.

c. Product quality enhancement

   Temperature control and defrost control play a big part in the life of a product under refrigerated storage. Storage life can be shortened considerably if the correct temperature within the case/coldroom is not achieved, or if defrost is not terminated correctly. With case/coldroom controllers this problem is reduced, due to the ease at which temperature settings etc., can be checked and adjusted.

2.7.3 Temperature sensors

Platinum resistance sensors

Platinum resistance sensors are more commonly known as PT100s or PT1000s. These have resistances of 100 and 1000 ohms respectively at 0°C. Their resistance increases at around 0.35 and 3.5 ohms/degree respectively. More accurate characteristics are published in BS1904. The sensors are moderately priced, can be used in a fairly wide range of temperatures, and because their characteristics depend on the basic properties of platinum, they are internationally available. The output signals are small however and require moderately accurate resistors to recover temperature information in Celsius.

Thermistors

Thermistors are resistors whose resistance varies rapidly with temperature. Thermistors are made from semiconductors and in refrigeration normally take the form of a disk encapsulated in a metal or plastic housing. A typical thermistor might have a resistance change of about 4% per degree C. Thermistors are particularly suitable for measuring temperatures normally encountered in refrigeration. Thermistors come in two types, Positive Temperature Coefficient (PTC) and Negative Temperature Coefficient (NTC) thermistor. PTC thermistors are those whereby the resistance of the probe increases with temperature. There is a type of thermistor where the resistance changes quite abruptly from some 10s of ohms to 10,000s at some specified temperature. They are quite commonly used for thermal protection applications.

The other type NTC thermistors, are those whereby the resistance of the probe decreases with temperature. The thermistors are used by the controller to measure the temperature within the cases or coldroom. The exact temperature to resistance relationship will be dependent on the probe type and will be important to the operation of the controller.

Thermocouples

Thermocouples are economical, small, fast acting, suitable for a wide range of temperatures, and can be made to withstand harsh environments. The thermocouple senses temperature by measuring small voltage differences that occur when connecting dissimilar metals. This means that the signal output is small (typically 40uV per degree). The thermocouple measures the temperature difference between the sensing element and the indicating device. A controller which uses thermocouples therefore needs an additional sensor to measure its own temperature.

2.7.4 Suction/liquid heat exchangers

This heat exchanger is used to sub-cool liquid refrigerant entering the expansion valve, using refrigerant vapour exiting the evaporator. Consequently, there is an increase in the superheat of the compressor suction vapour. The increase in suction superheat can also increase discharge temperatures at the compressor and so great care must be taken to ensure that the compressor operating limits are not exceeded. Suction/liquid heat exchangers are used for insurance against the possibility that any liquid refrigerant leaving the evaporator will reach the compressor. Higher suction line temperature can result in less non-useful heat pick up, and this will enhance system efficiency, but heat exchangers should only be used after careful consideration of the system design and operation. The beneficial effect of suction/liquid line heat exchangers does vary with different refrigerant fluids and it is important that careful compressor performance modelling with the selected refrigerant be carried out to determine whether this is the optimal option.

2.7.5 Oil separators

All compressors pump oil from the crankcases to a greater or lesser extent. The amount of oil carry over depends on the compressor type and the load on the compressor itself.

Oil within refrigerant systems can cause problems due to its miscibility and solubility with refrigerants. System pipework is designed to keep pressure drops as low as possible and at the same time keep refrigerant velocities high enough to transport oil.

Where varying loads occur on systems the velocities will at times fall and this can lead to accumulation of oil in the low pressure part of the system. Large amounts of oil in the evaporator can cause an insulating effect on the evaporator heat exchanger surface which results in duty loss and poor operation of the system at a lower evaporating temperature than the original design.

An oil separator can significantly reduce the amount of oil circulating through the system and should always be considered – particularly on low temperature systems with remote evaporators where the viscosity of the oil/refrigerant mixture in the evaporator can be high. The lubricant recommended by a compressor manufacturer should be used, since an account will normally have been taken of its behaviour in the systems for which the compressor is intended.

It is important to recognise that no separator removes 100% of entrained oil, and systems must always be designed for adequate oil return velocities.

Oil separators should be selected in accordance with the manufacturer's information. Most manufacturers recommend selection is based on the maximum mass flow of the compressor within the system, and that the inlet and outlet connections of the oil separator are not smaller than the system discharge lines.

Oil separation is dependent on the discharge gas velocity. With multiple compressors it is advisable to ensure that under low load conditions the velocity is sufficiently high to maintain efficient oil separation. Individual oil separators may be required for each compressor. When more than one separator is used a check valve should be fitted at the oil return line from each separator. When subject to low ambient conditions, the oil separator shell should be heated or insulated to eliminate refrigerant condensing within the shell. A solenoid valve should also be installed in the oil return line to the crankcase so that if liquid refrigerant is present within the oil separator shell, it cannot enter the crankcase during the compressor off cycle.

Oil injected compressors such as screw compressors depend on a high oil flow through the machine and usually have a built-in separator and oil injection arrangement.

2.7.6 Pressure regulators

Pressure regulators are frequently used to control the refrigerant pressure in an evaporator at a condition higher than that of the compressor suction. There are two situations in which this type of control is required.

Evaporator pressure regulation
Evaporator pressure regulators (EPRs) can be used on systems with multiple evaporators working at different temperatures, or they can act as a safety device to prevent water chillers working at unacceptable low temperatures.

EPRs control the flow of refrigerant gas by monitoring and regulating the conditions at the exit of the evaporator. If this pressure drops below the set point of the valve, an internal piston progressively closes, restricting the flow of gas until the evaporator's pressure rises to the set point. If the suction pressure of the compressor, and hence the evaporator, is above that of the valve's set point it will remain in the fully open position.

For the EPR to maintain an accurate control there will always be a pressure drop through this type of valve. The magnitude of the pressure drop will depend upon the selection of the valve which should meet the system duty and the prevalent operating conditions. However, any pressure drop in the suction side of a system will reduce the plant's efficiency.

EPRs should not be installed close to the compressor as this may cause hunting due to the very small refrigerant volume between the valve and compressor suction.

Crankcase pressure regulation
Where a compressor is limited at higher evaporating temperatures by motor size it is possible for the motor to become overloaded during the pull-down period, and crankcase pressure regulation can be used to limit the suction pressure.

Control is arranged so that when the outlet pressure drops, the valve opens to allow a greater flow of refrigerant gas to pass through it to the compressor. Conversely, as the compressor suction pressure reaches the valve's set point it closes, throttling the pressure to the compressor.

The compressor motor need not be sized for the maximum load or pull-down if this is either of a short duration or if it can be extended over a longer period. Use of a smaller motor means that during the majority of the operating cycle, when the load is at its normal level the motor will be better matched to the application and thus operate more efficiently.

Pressure regulators should always be selected on a duty/pressure drop basis and NOT on pipe size.

2.7.7 Liquid line sight glasses/moisture indicators

Liquid line sight glasses should be fitted to all condenser systems and should be installed after the liquid line drier and immediately before the expansion valve.

All sight glasses should be complete with a moisture indicator appropriate for the refrigerant charged in the system. For copper pipework brazed copper connections should be used.

2.7.8 Non return valves

A non return valve (NRV) should be fitted in the discharge line when using remote air cooled condensers if the discharge pipework could allow liquid refrigerant migration during the off cycle. NRVs on this type of application should be suitable for the system discharge temperatures and pressures.

NRVs are also used in saturated and hot/cool gas defrost systems on a bypass arrangement around the expansion valve. These valves should be of the sweat type with copper tails and sized for full liquid line flow.

Care should be taken not to oversize NRVs because the valve may chatter at reduced flows.

2.7.9 Hand shut off valves

Hand shut off valves should be installed where appropriate to isolate components and machinery that will require maintenance and to eliminate the loss of refrigerant during maintenance.

All shut off valves should be designed with gas tight stem. Reliance should not be placed on a cap to prevent leakage because the valve will leak gas when the cap is left off or lost.

2.7.10 Oil differential pressure switches

This control should be fitted to all compressors that incorporate a mechanical high pressure oil pump. The timer delay range of the pressure switch arrangement should be of the type recommended by the compressor manufacturer. Intelligent oil safety controls that eliminate the need for pressure connections and joints are preferred.

2.7.11 Safety cut outs (switches)

These should be set and sealed by the commissioning engineer and labelled with their respective settings.

Safety means SAFETY not CONTROL.

Double bellows type pressure switches are preferred to reduce refrigerant leakage.

Pressure switches that connect directly into the main pipework should be considered, as should electronic arrangements using pressure transducers. Refer to EN378.2:2016

2.7.12 Solenoid valves

Solenoid valves installed in the liquid line should be as close as possible to the expansion valve particularly when a pump down control system is being used. These valves should also be fitted in the oil return line from the oil separator to the compressor where individual compressor receiver or condensing units are being applied.

These components should be sized on refrigerant flow. Where servo operated valves are used, pressure differences should be checked to make sure they are sufficient for the valve to operate.

2.7.13 Pressure control tubing

Flexible nylon capillary tubing with capillary hose fittings may be used for connecting instruments and pressure controls provided the material is of suitable permeability in for the refrigerant to be used. Copper tubing should have brazed connections throughout.

2.8 REFRIGERANT PIPEWORK

2.8.1 Introduction

The pipework should be sized taking into account the following factors:

1. Pressure drop
2. Oil return to the compressor
3. Minimum refrigerant charge
4. Liquid refrigerant control
5. Prevention of vapour locks
6. Low noise
7. Flexibility over entire operating range
8. Minimum heat pickup

Some of these parameters are dealt with elsewhere in this guide. Oil return, refrigerant charge and noise are covered here.

2.8.2 Pressure drop

Pipework should be selected to minimise pressure drop. The effect of pressure drop on performance and efficiency is more significant in suction lines than in other pipework. A relatively small pressure drop in the suction pipework of a low-temperature installation will cause a serious reduction in the system's efficiency.

A pressure drop in a discharge line will have no significant effect on efficiency. As the pressure ratio of a system decreases so does the effect of a suction line pressure drop.

2.8.3 Sizing

It is very important that pipe sizing calculations are carried out. Selecting pipes according to the connection sizes on compressors and other components will not guarantee the best design. These connections are sized to suit a wide range of applications, but will not be correct for every installation.

Charts and software which give pressure drops for a given refrigerant and system capacity over a range of evaporating or condensing temperatures are available from a number of sources.

Fittings and components, such as pipe bends and valves, have pressure drops which are commonly expressed as an equivalent length of straight pipe. These can be obtained from manufacturers' catalogues. The pressure drops for fittings specified in systems should be checked as there may be options with lower pressure drops. For example, a swept bend will have a lower pressure drop than a right angle bend and will also have a longer life.

Where there is any choice, long vertical upflow pipe runs should be avoided. In liquid lines the static head from such a rise must be converted into a pressure drop. This will enable the equivalent loss in liquid subcooling due to the pressure reduction to be ascertained and indicate whether evaporation of liquid refrigerant could occur before the TEV.

2.8.4 Oil return

The sizing of pipework to ensure that good oil return is maintained is contrary to the parameters for minimising pressure drops. Unless an adequate gas velocity is maintained, oil will accumulate in the low pressure side of the system and possibly result in compressor lubrication failures.

The final pipe size selection has to be a compromise between a small diameter pipe to ensure that there is sufficient refrigerant velocity to carry oil, and a large diameter pipe to minimise any pressure drop. Some pressure drops can be reduced by careful attention to the design and installation of pipework. Limiting the number of fittings and bends which increase the pressure drop is one way of achieving this.

Note: Slope discharge pipework down from the compressor

To give good oil return the pipework should slope towards the compressor. Minimum slope of 1 in 400, is recommended.

2.8.5 Refrigerant velocity

As stated previously, refrigerant velocities in pipework are important for the recovery of oil back to the compressor from the system. As a general guide, the following velocities should be considered:

Liquid Ex The Condenser	0.5m/s (100 fpm)	–	0.76m/s (150 fpm)
Liquid Ex The Receiver	0.5m/s (100 fpm)	–	1.0m/s (200 fpm)
Suction	4.0m/s (800 fpm)	–	17.8m/s (3500 fpm)
Discharge	10.0m/s (2000 fpm)	–	17.8m/s (3500 fpm)

2.8.6 Oil and liquid traps

It is important that traps are incorporated into pipework systems, in order to prevent liquid refrigerant draining back to the compressor(s) during off cycles, and to aid oil return to compressors during run cycles. This particularly applies where vertical pipework is involved. Details as follows:

- Liquid Traps: On vertical discharge pipe runs, it is usual to take the pipework to the highest point of the condenser before returning to the inlet connection of the condenser. This ensures any liquid from the condenser does not drain backwards down the discharge onto the cylinder head of the compressor(s) during off cycles. Another method of eliminating this problem is to fit a check valve in the discharge line at the lowest point after the pipework leaves the compressor.
- Oil Traps: Oil traps are installed in suction lines to ensure oil return to the compressor(s), they are usually installed at the ends of vertical risers

2.8.7 Vertical suction risers

To maintain oil return vertically rising suction pipe may need to be of reduced size. The increased pressure drop can be offset by increasing the diameter of the horizontal part of the suction line. Oil traps should also be provided at 6m intervals to assist oil return.

Where capacity control is utilised, the mass flow may not be high enough to give the correct velocity for oil return. Double risers should then be considered.

2.8.8 Double suction risers

Double suction risers consist of two pipes, one pipe sized to provide oil entrainment with the compressor operating at partial load, and the two together capable of lifting oil at maximum operating conditions.

Oil traps must be provided between the two risers. At partial load operation when the gas velocity is insufficient to return oil through both pipes, the trap will gradually fill with oil until the second riser is sealed off. The refrigerant gas then travels up the first pipe only with sufficient velocity to carry the oil along with it into the horizontal pipe section. Precautions must be taken to make the trap as small as possible by close coupling the fittings. This ensures that the oil holding capacity of the trap will not seriously lower the compressor crankcase level. It also avoids slug-back of oil to the compressor when the trap clears on the resumption of increased loading.

The second riser pipe must be formed at it's upper limit with an inverted loop to enter the main horizontal suction pipe from above. This prevents oil draining into this riser whilst idle during partial load operation.

When selecting the combination of pipes for a double suction riser, the diameter of a pipe that will ensure oil entrainment at maximum compressor duty; must be established and also the diameter of the pipe that will give satisfactory operation at minimum duty. The combined cross section area of the two complementary parallel risers must not exceed that of the single pipe which is suitable for the entrainment of oil at maximum duty.

A close coupled trap should be formed at the lower end of any suction pipe at the point of rising to a higher level for the collection of fall-back oil during off-cycle time. An oil lock must be formed to retain as little oil as possible preventing serious reduction in crankcase oil level when the plant operates at partial duty.

In the examples in Figure 2.8 pipe 'A' is sized to provide oil entrainment at minimum duty, and pipes 'A' & 'B' together provide oil entrainment at maximum duty when the oil lock clears due to increased pressure drop in pipe 'A'. Pipe 'B' must enter the horizontal suction pipe from above to avoid oil carried up pipe 'A' from draining into pipe 'B' when the plant is operating at partial duty.

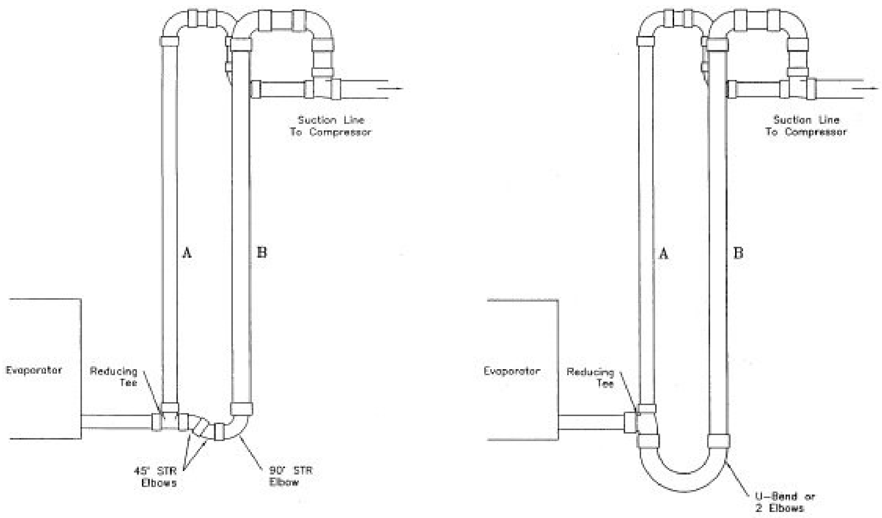


Figure 2.8: Examples of double suction riser layout

2.8.9 Multi stub

This term is used to describe a design in which a number of refrigerated fixtures are assigned to one suction and liquid line. The suction and liquid lines usually run from a group of fixtures back to a multi compressor pack. The group of fixtures should operate at approximately the same evaporating temperature.

This system is relatively simple to design using pressure drop and velocity graphs, or computer software, all that is required is the total duty and the lowest evaporating temperature of the group of fixtures. This system has been applied in conjunction with saturated or hot gas defrost, which are now less used.

Liquid distribution to the multi-evaporator system is very important if poor performance is to be avoided and there are several methods of achieving this:

- Header Method: The liquid feed to a number of evaporators is fed to the centre point of a header. This feed line should have sufficient capacity to ensure that the header remains full of liquid refrigerant at all times.
- Ring Main Method: This method is usually employed where many evaporators are fed from a large capacity single stub. The ring main ensures that liquid refrigerant feeds the supply line to each evaporator from two directions.

- Reducing Method: With this method the liquid refrigerant feed line reduces in size immediately after the tee-off to each evaporator, ending finally with the feed pipe size used to feed each evaporator.

2.8.10 Radial main system

On a radial main system, the liquid and suction pipework for coolers most distant from the compressors is comparatively small because it is sized for only one set of refrigerated fixtures. The added effect of fixtures nearer the compressors requires the main suction liquid lines to be increased to allow for the increased duty. The branch suction and liquid lines are sized in exactly the same way as a multi stub project. On this type of system it can be seen that the actual duty can vary tremendously on the main suction and liquid line. It is therefore very important that refrigerant suction lines slope back to the multi compressor pack; all vertical risers need a minimum of double pipe arrangement, if not to three pipe arrangement, depending on the fluctuation in load.

2.8.11 Guidance data for computer selection of pipework

LT System

Superheat (K)	10K
Liquid temp	5K below condensing temp
Suction pressure drop	2K
Head difference	0
Liquid line pressure drop	1K

Risers to be designed to return oil at minimum 33% capacity

HT System

Superheat (K)	10K
Liquid temp	5K below condensing temp
Suction pressure drop	2.5K (2. OK) condensing units
Head difference	0
Liquid line pressure drop	1K

Risers to be designed to return oil at minimum 33% capacity
Liquid temp factor assumes no mechanical sub-cooling

2.8.12 Pipework marking

All pipework carrying refrigerant should be marked in the following manner.

The label shall be in accordance with BS1710 *Specification for the Identification of Pipelines and Services*

An adhesive label shall be provided which will not under normal circumstances pull off or fall away from the copper pipework or insulation. The label must be affixed at intervals of no more than 2m intervals.

The label shall contain the following information.

Refrigerant No
Direction of Flow
Rated Pressure

The information shall not be easily removed from the label.

2.8.13 Pipe selection

It is extremely important that the correct gauge of copper pipe is selected and the following publications should be consulted:

EN 378-2:2016 *Refrigerating Systems and Heat Pumps Safety and Environmental Requirements, Piping*

2.8.14 Pipework insulation

Insulation is essential on all suction lines to eliminate condensation and reduce non-useful superheat. Liquid direct from a condenser or receiver benefits from additional subcooling (natural subcooling) which can occur in the line, and so insulation is not then required. Where a heat exchanger is employed to provide subcooling liquid lines should be insulated to reduce heat pickup.

The liquid line should not be passed through an area of high ambient temperature as this will reduce the sub-cooling and may cause evaporation of liquid before the expansion device. If liquid lines must pass through areas of high ambient temperatures (i.e. where ambient temperatures exceeds liquid refrigerant temperature) they must be insulated.

Thus liquid lines on conventional systems do not normally require insulating especially if passing through areas where the ambient temperature is lower than the liquid refrigerant temperature. However, in some cases such as false ceilings, insulation may be required to reduce heat pickup.

Insulation of discharge lines should be considered for safety reasons. The insulation specification depends on such environmental conditions as maximum ambient temperature, air dew point, relative humidity and air flow. Manufacturers' technical information should be consulted for determining the required insulation.

Care should be taken to make sure that the insulation conforms to the correct fire regulations relating to the site.

It is important that pipework insulation is carried out by competent technicians suitably trained in such installation work. All joints should be glued to make an air tight assembly.

2.9 DESIGN CONSIDERATIONS

2.9.1 Design for minimal leakage

Priority must be given to minimising potential sources of leaks in all aspects of the design.

2.9.2 Suction and condensing pressure

Examination of compressor performance data will illustrate the effect of condensing

Pressure, power consumption and capacity:

Example 1:

Bitzer 6HE-28Y R449A Evaporating at -5°C (Dew point)

Condensing °C	Capacity kW	Absorbed kW	C.O.P.
43	69.50	24.00	2.9
30	83.00	19.91	4.17
20	93.40	16.19	5.77

Example 2:

Bitzer 6HE-28Y R513A Evaporating at -5°C

Condensing °C	Capacity kW	Absorbed kW	C.O.P.
43	45.70	15.27	2.99
30	55.00	12.82	4.29
20	61.90	10.48	5.91

Performance figures courtesy of Bitzer UK Limited.

The compressor capacity increases as condensing temperatures fall, the net result is less running time for the compressor.

Note that:

- Considerable savings can be made even with small condensing temperature reductions.
- Heat recovery projects, where very high condensing pressures are maintained throughout the year, may prove to be less effective when compared to the above savings which they preclude.
- With water cooled condensers, there can be significant variations in the water temperature which necessitate some form of condensing pressure regulation for conventional expansion valves to work correctly. Use of electronic expansion valves allows lower temperatures, which apply for most of the year.

The increased efficiency in a system that can be achieved if the suction pressure is increased is illustrated below.

EXAMPLE: Bitzer 4J-13.2, R404A, Condensing at 43°C

Saturated suction °C	Capacity kW	Absorbed kW
-10	33.4	14.43
-8	36.1	15.01

Performance figures courtesy of Bitzer UK Limited

Although the power consumption may increase, the increased capacity will result in a compressor running less hours.

2.9.3 Refrigerant charge and isolation for maintenance purposes

The first consideration is charge limitation. Pipe sizes and pipe runs should be kept as small as possible to reduce system working charge. Head pressure control by flooding the condenser with liquid refrigerant should not be considered. However due consideration should be given to the pressure drop of the selected pipework. Refrigerant receivers should be as small as possible, horizontal refrigerant receivers require larger working charges than vertical types. This is because the area between the bottom of the receiver and the dip tube is larger on horizontal receivers than the vertical type.

Pump down of the full working charge should be accommodated within the receiver plus the remote air cooled condenser heat exchange surface area.

To enable components to be changed without loss of refrigerant, hand operated isolation valves with sealing caps should be considered.

Bypass arrangements so that liquid line driers can be changed whilst the equipment is still operating should be installed on the larger type of system.

Consideration should be given to the isolation of equipment such as solenoid valves and expansion valves. EN378-2:2016 should be consulted regarding maximum charge of refrigerant and restrictions on use. Consideration should be given when checking isolation valves to ensure liquid refrigerant is not trapped in pipework and allowed to rise in temperature, this causes a hydraulic effect and dangerously high pressure can occur causing fracturing of pipework etc.

2.9.4 Pump down cycle

When a refrigeration system is not in operation for long periods of time, and if the compressor becomes colder than other parts of the system, the refrigerant charge may migrate to the compressor. At the compressor crankcase this refrigerant vapour may condense into the oil, until the oil is completely saturated with refrigerant, causing lubrication problems.

The most effective method of reducing the migration of liquid to the compressor crankcase is by the use of a pump down cycle. This involves the isolation of the refrigerant in the high side of the refrigeration system, Figure 2.9.

The cycle operates by closing a solenoid valve located in the liquid line between the expansion valve and the receiver. The liquid line solenoid valve is controlled by a thermostat reading the fixture temperature.

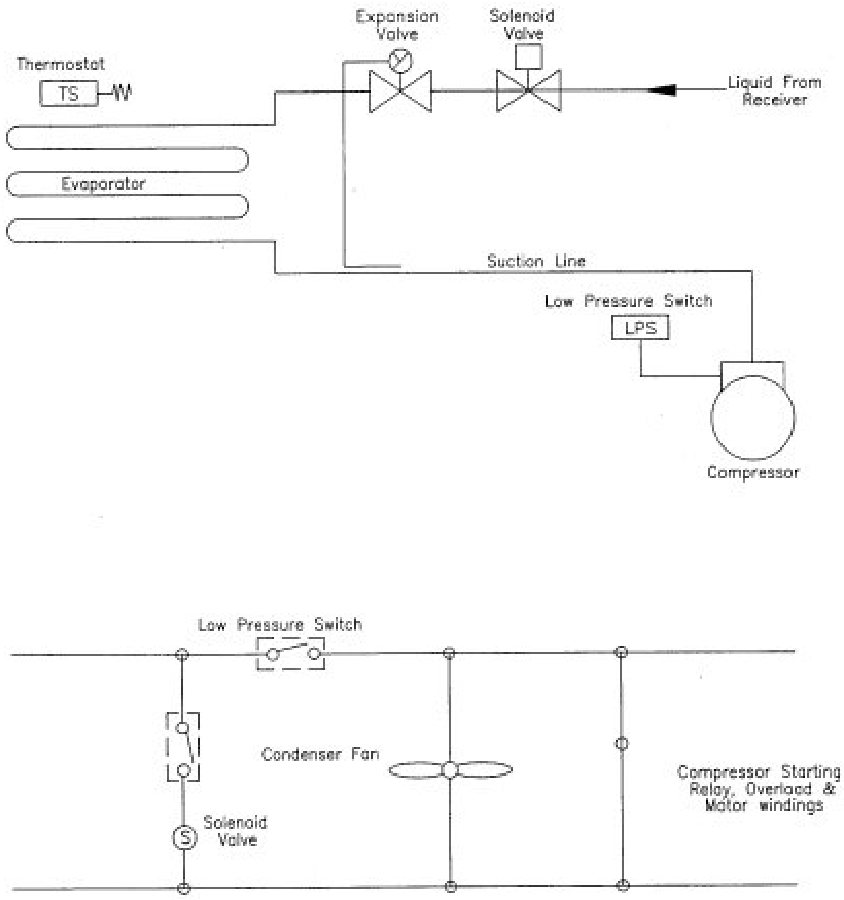


Figure 2.9: Physical and electrical arrangement for a typical pump down cycle

The compressor continues to operate, pumping the refrigerant into the condenser, receiver and liquid line. The operation of the compressor is controlled by the use of a low pressure switch sensing the suction pressure. The suction pressure drops until the compressor is switched off by the low pressure switch. This type of cycle can be used on most superheat regulated systems, providing the high side of the system is adequate in capacity to hold the system charge.

After the pump down cycle, it is likely that small amounts of refrigerant will leak past the solenoid valve. This will cause the suction pressure to rise again. Therefore this process must be repeated.

A crankcase heater may also be fitted to the compressor to keep the oil warm enough to prevent refrigerant from condensing into the oil. The crankcase heater should be switched on at all times when the compressor is not operating.

2.9.5 Defrost methods

There are four common methods of defrost used in commercial refrigeration.

a. Off cycle
b. Electric
c. Hot gas
d. Saturated gas

The advantage/disadvantage of these systems can be summarised as follows:

a. Off cycle

Off cycle defrost does not use any form of artificial defrosting technique such as electric heater, or refrigerant gas, to raise the temperature of the evaporator for ice removal.

Off cycle defrost is not employed on applications where the temperature is below +2°C. In some instances the ice build up that may occur on the evaporator heat transfer surface is removed during the normal off cycle. An alternative method is to employ a definite off cycle process when a timer shuts down the refrigeration equipment for a fixed period of time.

b. Electric

Electric defrost is simple to operate, control, and does not rely on any external service for the supply of heat. Electric defrost can be used on all types of system.

c. Hot gas

Hot gas defrost uses the discharge gas as the source of heat, i.e. discharge gas is circulated through the evaporator and an external heat source is not required. However, coldrooms/display cases on the system need to be in the refrigeration mode to give a heat source for defrosting. This may lead to inefficient operation of the system during low load and low ambient conditions.

Repeated expansion and contraction of pipework and joints during the defrost cycles may lead to refrigerant leakage and premature failures of pipework and evaporators. For this reason, these systems are no longer widely used.

d. Saturated gas defrost

Saturated gas defrost uses refrigerant vapour which is formed above the liquid in the refrigerant receiver. Therefore the temperature of the gas is the same as the condensing temperature for the system. The disadvantages and advantages are similar to hot gas defrost. Expansion and contraction of pipework is not quite so critical, but nevertheless these systems are no longer widely used.

2.9.6 Electric defrost – method of operation

A controller/timing device initiates defrost by disconnecting the electricity supply to the liquid line solenoid valve. At the same time the controller connects the electricity supply to the defrost heater, on the larger type of application this may be via an electrical contactor. Where coldrooms are concerned, the evaporator fans are switched off once defrost is initiated, this eliminates heat from the defrost heaters being blown into the coldroom.

Defrost will be terminated on either time, or by sensing the temperature of the evaporator heat exchange surface using a thermostat or sensor.

Once defrost is completed the defrost heaters are switched off, the liquid line solenoid valve is then opened and refrigeration re-commences.

In the case of coldrooms, the fans on the evaporators will not start to operate until the moisture on the evaporator heat exchange surface has been re-frozen. This eliminates moisture being blown into the coldroom.

Condensing unit applications may employ a pump down cycle as described in section 2.9.4.

For supermarket applications it is common practice to have up to four sections of the same type of display cases connected either to one stub on a multi compressor application, or to one condensing unit. Individual case defrost termination is probably the most energy efficient and cost effective in terms of product enhancement.

All sections are defrost initiated at the same time. However, as each case section will have its own defrost electrical control, once the case section has achieved defrost, the defrost termination thermostat would interrupt the electricity supply to the defrost on this specific case section.

All case sections would terminate and refrigeration would then be initiated by a timer. Therefore a wait period would occur between all the case sections being defrost terminated on their defrost termination thermostat and the refrigeration being initiated by the timer.

Individual case control can be achieved by a thermostat/controller per case section, thus controlling a solenoid valve situated in each case liquid line. In this situation the defrost initiation timer would interrupt the electricity supply to all the individual case liquid line solenoid valves, defrost termination would proceed as previously described.

2.9.7 Vibration and noise

There are some basic measures that can be taken which can significantly help to reduce both vibration and noise to an acceptable level. Attenuation is a specialist subject and noise and vibration specialist should be consulted if problems arise, or specific requirements are specified or needed.

How pipework assemblies are designed and installed has an important impact on controlling the generation and transmission of noise and vibration in a refrigeration plant. Care should always be taken when assembling pipework to avoid rigid fixing.

To avoid amplification of compressor vibrations, the compressor should be mounted according to manufacturer's recommendations. This is particularly important for multiple compressor installations.

A distance of approximately 15 pipe diameters should be allowed between the compressor and the first pipe support on suction and discharge lines. This allows a long enough length of pipe to provide flexibility. It should be noted that the distance to the pipe support does not need to take into account the otherwise lengthening effect of the bend. If the pipe rises vertically from the compressor valve, this can also include the bend which is placed a short distance from the valve.

Components carried in pipelines, such as liquid traps and mufflers, should be supported. The additional heavy mass of some of these items can pose significant extra stress on pipeline assemblies. To reduce noise generated through the contact of a pipe within a pipe clamp, a small piece of resilient and deadening material (such as neoprene) should be used to stop metal-to-metal contact and thereby reduce "chatter" which may provide refrigerant leakage at a later date.

When a pipe passes through a wall, a liner should be placed within the hole and the opening properly rendered. The opening should be made larger than the pipe diameter to avoid the possibility of direct contact. The intervening space should be filled with resilient material such as fibre glass or polyurethane foam to seal it, deaden sound and protect from ingress of dust and the elements.

Flexible hoses/vibration eliminators

In some refrigeration and air conditioning applications the refrigerant lines must be flexible, for example connections to controls or gauges. (See section 2.7.13). This tubing is usually of a smaller diameter, e.g. ¼ " OD - 5/8" OD being the most common sizes.

Where vibration may be a problem i.e. pipework connecting to compressors, vibration eliminators are often installed in the suction and discharge lines as near to the compressor as possible. Vibration eliminators should always be installed in parallel to the compressor crankshaft, i.e. horizontally for piston compressors and vertically for vertical scroll machines.

Discharge mufflers

A muffler in the compressor discharge line can dampen discharge gas pulsations. The purpose of the muffler is to reduce vibrations and noise which would otherwise be transmitted and amplified by the pipework system. A muffler consists of a perforated pipe passing through a large diameter cylinder which forms a chamber with baffles.

The device is usually placed in the horizontal or vertical pipe immediately leaving the compressor. It is not usual to fit a discharge line muffler in systems equipped with an oil separator; the oil separator can itself function as a muffler.

2.10 LOAD CALCULATIONS

2.10.1 Introduction

When a refrigeration system is at the design stage, or when evaluating the suitability of a given system for a particular purpose, it is necessary to know the maximum duty required. A more detailed understanding of cooling loads allows designers to:

- Identify opportunities to eliminate or reduce loads.
- Optimise the temperature levels of the refrigeration system.
- Correctly select, size and control compressors.
- Evaluate energy savings, for example the use of more efficient compressors, heat exchangers with larger surface areas and the possibility of thermal storage.

2.10.2 Gathering data

Gathering relevant data for the cooling requirements is not always an easy task. For each cooling load, detailed information is required. This information is summarised below.

The data is required for all cooling loads if a full analysis is to be carried out.

To collect the information it is necessary to understand in some detail the process or system that requires cooling.

If the task of selecting the refrigeration equipment is left to personnel who do not have no access to the relevant information for the duty requirements, then there is a possibility that inefficient plant will be selected and installed.

2.10.3 Cooling load data

- Maximum cooling demand.
- Variations in cooling demand.
- Timing of the demand.
- Duration of the cooling demand.
- Initial and final temperatures of the products or fluids to be cooled.
- Specific heat, latent heat and respiration of the products.
- Other data includes constraints on condensing medium and diversity of heat generating equipment for example.

The best source for this information is discussions with key personnel, production managers and engineers.

2.10.4 Types of cooling load

Cooling loads can be grouped as follows:

- Sensible: the cooling of fluids or solids.
- Latent: freezing and condensing.
- Heat of respiration from fruit and vegetables.
- Insulation: cold rooms: pipes and vessels: etc.
- Infiltration: through cold store doors.
- Ancillaries: evaporator fans, chilled water pumps and lighting.
- Other: people and trucks in cold stores: etc.

2.10.5 Sensible cooling loads

Sensible cooling loads are the reduction of temperature of a substance without a change of phase occurring.

Examples include the chilling of water, air, or a product.

The cooling load is calculated as follows

$Q = mCp\ (Ti - Tf)$ Equation 1

where:
Q = Cooling load (kW)
m = Mass (1kg)
Cp = Specific heat capacity (kJ/kg °C)
Ti - Initial temperature (°C)
Tf = Final temperature (°C)

Specific heat capacity changes with temperature, although it can often be approximated as a constant. Values for specific heat capacities for common substances can be obtained from standard refrigeration and engineering references.

2.10.6 Latent heat

Latent heat is heat added or removed at constant temperature when a change of state takes place i.e. evaporation, condensing, melting and freezing.

The latent heat of fusion is the energy required to change a liquid to a solid or vice versa.

Fusion (freezing or melting) occurs at the fusion temperature (freezing/melting point) of the substance. Fusion temperatures vary slightly with pressure.

The latent heat of condensation or evaporation is the energy required to change a gas to a liquid or vice versa.

Condensation or evaporation takes place at a temperature which is highly dependent on prevailing pressure.

The latent cooling load is calculated as follows:

$Q = mL$ Equation 2

where:
Q = Cooling load (kW)
m = Mass (kg)
L = Specific latent heat of fusion or vaporisation (kJ/kg)

Values for freezing points, saturation temperatures/pressures and latent heats of fusion and vaporisation can be found in standard texts (see Appendix 1).

2.10.7 Heat of respiration

Products such as fruit and vegetables are living tissues and therefore produce heat which has to be considered as part of the heat load calculations.

This respiration effect can occur also in chemical cooling or fermentation and is known as the heat of reaction.

The heat of respiration is calculated as follows:

$Q = MR$ Equation 3

where:
Q = Cooling load (kW)
M = Mass flow (kg/s)
R = Respiration heat factor (kJ kg ºC)

People also generate heat which has to be allowed for in heat load calculations. The amount of heat generated will depend on their activity. Tables are available with the appropriate value in respect of both activity and room temperature. For air conditioning calculations both the sensible and latent loads should be considered. Whereas for refrigeration applications total heat loads are used.

2.10.8 Conduction heat gains

This load is the heat gain through the walls, ceiling and floor and for a cold store, chill room or air-conditioned area.

The load is calculated as follows:

$Q = U\ A\ (T2 - T1)$ Equation 4

where:
Q = Cooling load (kW)
U = Heat transfer coefficient (kW/m^2 °C)
A = Surface area (m^2)
T1 = "Cold" side temperature (°C)
T2 = "Hot" side temperature (°C)

The "U" value is based upon the thermal conductivity of the insulation material (wall, roof, floor, etc) and the thickness of the material:

$U = k\ l$ Equation 5

where:
U = Heat transfer coefficient (kW/m^2 °C)
k = Thermal conductivity (kW/m °C)
l = Thickness of material (m)

For a composite wall or roof structure the overall "U" value must be calculated based upon the thickness and thermal conductivity of the individual elements. "k" and "U" values are published for a wide range of construction materials in standard texts.

The internal ("cold side") temperature of a cold room is normally constant throughout the year. The external ("hot side") varies with the seasons. It is important to consider the impact of the variation in seasonal external temperatures on the peak cooling loads.

If the structure being cooled is subject to direct solar radiation, an allowance must be made for the increase in surface temperature (i.e. T2 the "hot side" temperature).

The increase in temperature is dependent upon:

- The time of year.
- Emissivity and the orientation of the surface.

The elevated temperature caused by solar radiation is known as the "sol-air" temperature. Values of sol-air temperatures can be found in standard texts.

Cooling loads for large cold stores may be reduced at the design stage by careful orientation of the building and choice of external surface colour to reduce heat absorption.

Pipes and vessels containing or circulating cold fluids also represent a significant cooling load.

2.10.9 Infiltration

Infiltration loads (the ingress of warmer air into a cold space) are significant, particularly for cold stores. The freezing of moisture entering with the air will be a latent load in the store and subsequently the frost removal will be an additional defrost load.

The infiltration cooling load is calculated as follows :

$Q = m\ (hi - hf)$ (kW) Equation 6

where:
Q = Cooling load (kW)
m = Air ingress (kg/s)
hi = Inlet enthalpy (kJ/kg)
hf = Final enthalpy (kJ/kg)

Values for enthalpy can be read from a psychometric chart or table, if the temperature and humidity inside and outside the store are known.

The primary leakage of air into cold and chill stores is through open doors.

After the opening of a cold store door a velocity profile is quickly established across the door opening with dense cold air falling out of the bottom of the opening and warm moist air entering near the top of the opening.

The infiltration cooling load caused by cold store doors can be reduced by having the minimum possible height of door, reducing opening time (by using automatic closing systems).

The use of strip or air curtains can reduce the infiltration load.

In practice, it is difficult to calculate infiltration loads without the aid of computer software, although equations can be found in some standard texts.

An alternative method entails the use of standard tables after a decision has been made into the amount of infiltration that will occur, i.e. Low, Moderate or High.

Q = M3 x ACR x gh　　　　Equation 6a

where:　Q = Cooling load (kW)
M3 = Cubic capacity of enclosure (m^3)
ACR = Air change rate
gh = Heat removed in cooling air to storage temperature (kJ/m^3)

Average number of air changes per 24 hours for storage rooms due to door opening and infiltration:

Above 0 °C

Room volume m^3	Air change per 24 hrs	Room volume	Air change per 24 hrs	Room volume	Air change per 24 hrs	Room volume	Air change per 24 hrs
2.5	70	20	22	100	9	600	3.2
3.0	63	25	19.5	150	7	800	2.8
4.0	53	30	17.5	200	6	1000	2.4
5.0	47	40	15.0	250	5.3	1500	1.95
7.5	38	50	13.0	300	4.8	2000	1.65
10.0	32	60	12.0	400	4.1	2500	1.45
15.0	25	80	10.0	500	3.6	3000	1.3

Below 0 °C

Room volume m^3	Air change per 24 hrs	Room volume	Air change per 24 hrs	Room volume	Air change per 24 hrs	Room volume	Air change per 24 hrs
2.5	52	20	16.5	100	6.8	600	2.5
3.0	47	25	14.5	150	5.4	800	2.1
4.0	40	30	13.0	200	4.6	1000	1.9
5.0	35	40	11.5	250	4.1	1500	1.5
7.5	28	50	10.0	300	3.7	2000	1.3
10.0	24	60	9.0	400	3.1	2500	1.1
15.0	19	80	7.7	500	2.8	3000	1.05

Comments.

1. For heavy usage the above figures should be multiplied by 2.

2. The above values have been established for many years and should only be used for traditional chill and cold rooms with one door. The tables must not be used for rooms with more than one door, dock loading bays or areas that are through loaded. Interpolation is permitted, but extrapolation is not.

Heat to be removed in cooling air to storage room conditions (kJ/m³)

OUTSIDE AIR CONDITION

		5C	**D.B.**	**10C**	**D.B.**	**15C**	**D.B.**	**20C**	**D.B.**	**25C**	**D.B.**	**30C**	**D.B.**	**35C**	**D.B.**	**40C**	**D.B.**
		70% RH	**80% RH**	**70% RH**	**80% RH**	**70% RH**	**80% RH**	**50% RH**	**60% RH**	**50% RH**	**60% RH**	**50% RH**	**60% RH**	**50% RM**	**60% RH**	**50% RH**	**60% RH**
STORAGE TEMP	15°C	-	-	-	-	-	-	2.77	7.0	16.8	23.3	34.5	42.7	56.4	66.4	81.4	96.5
	10°C	-	-	-	-	10.5	13.8	16.6	20.9	30.87	37.5	48.8	57.7	70.1	81.2	96.5	112
	5°C	-	-	9.6	12.0	22.8	26.2	29.0	33.5	43.7	50.5	62.1	70.6	83.9	95.4	111	127
	0°C	9.1	10.9	20.8	23.3	34.4	37.9	40.8	45.4	55.9	62.9	74.9	83.7	97.4	109	125	141
	-5°C	19.2	20.9	31.0	33.5	44.6	48.2	51.2	55.8	66.4	73.5	85.5	94.4	108	120	136	153
	-10°C	28.7	30.5	40.8	43.4	54.8	58.4	61.4	66.1	77.0	84.2	96.6	106	120	132	148	165
	-15°C	37.8	39.7	50.2	52.8	64.5	68.2	71.3	76.1	87.2	94.6	107	116	131	143	160	177
	-20°C	46.1	48.0	58.8	61.5	73.4	77.1	80.4	85.3	96.6	104	117	127	141	154	171	189
	-25°C	55.1	57.1	68.0	70.8	82.9	86.8	90.1	95.1	107	114	127	137	152	165	183	201
	-30°C	64.2	56.2	77.5	80.1	92.6	96.5	99.8	105	117	125	138	148	163	177	195	213
	-35°C	73.3	75.3	86.7	89.6	102	106	110	115	127	135	149	159	174	188	207	225
	-40°C	83.3	85.4	97.1	100	113	117	121	126	138	147	161	171	187	201	220	231

DB = Dry bulb temperature, RH = Relative humidity

2.10.10 Ancillary cooling loads

Ancillary loads come from fans, lights, production machinery, defrost equipment and pumps.

The duty required to offset the heat generated by ancillary loads is equal to the power absorbed by each load.

It is important to note that auxiliary heat loads from devices such as pumps, fans or lights have a "multiple energy penalty". Electricity is first consumed to operate the ancillary equipment and then further electricity is consumed by the refrigeration system in order to remove the heat generated.

It is beneficial to reduce auxiliary loads to a minimum. In the case of production machinery or process pumps, it is beneficial to have the driving motor outside the refrigerated space.

It is important to remember that small electrical motors i.e. motors up to 3.0kW are inefficient (particularly single phase motors) and can produce more energy/heat gain than the motor name plate rating, or absorbed power.

2.10.11 Margins (commercial equipment)

All loads must be assessed and added together, after which suitably sized equipment can be selected. It is normal practice to select equipment so that it has sufficient capacity to cope with the daily load without operating continuously. This will leave sufficient time for defrosting.

Daily running times to allow for defrosting will depend if defrosting is active (electric or hot gas) for temperatures below 3°C. Passive defrosting can be used for room temperatures above 4°C when room air is circulated through the cooler with the refrigeration condensing unit stood down.

Passive off cycle defrost	16/18 hours condensing unit running time per day
Active electric or hot gas defrost	18/20 hours condensing unit running time per day

It is recommended that a safety margin is added to the capacity of the equipment to cover for heat loads that may be higher than anticipated.

The accuracy of information available will depend on the margin required. This can vary from 5–10%. Small commercial applications would normally have a margin of 5%.

2.11 PLANT ROOM DESIGN

2.11.1 Introduction

For the efficient operation of equipment in a plant room there should be:

- adequate ventilation to remove heat dissipated by compressors or any heat exchangers positioned in the room
- sufficient space around components for ease of maintenance
- access to the critical features of compressors like sight glasses and service valves
- lifting equipment for handling major pieces of equipment

The capacity and number of fans required to achieve adequate ventilation will depend on the shape and size of the room and amount of heat required to be removed.

To reduce the energy consumption of ventilation equipment fans can be thermostatically controlled by the use of a thermostat so they only operate when required determined by plant room temperature.

Two main factors must be taken into account:

- a good overall air flow throughout the plant room area
- a suitable inlet or outlet vent to prevent any restriction of the air flow to, or from, the fan

When air cooled condensing units have been installed with the air flow from the condenser discharging into the plant room, adequate means for removing this quantity of air must be made available. This can be achieved by air extract fans, or exhaust ducts.

Failure to provide adequate ventilation will reduce the operating efficiency of both compressors and condensers.

High plant room temperatures will:

- increase the suction gas temperature entering the compressor reducing cooling capacity
- reduce the heat exchange capability of air cooled condensers increasing their operating pressure and lowering the system efficiency

2.11.2 Ventilation requirements for condensing units

To estimate the total air volume required, consult manufacturer's data to establish the heat rejection for each condensing unit and find the total heat rejection, Q. If this data is not available the heat rejection may be estimated as follows:

Assume a temperature rise across the condenser of 10K.

The following calculation will establish the air quantity required for ventilation purposes.

Total Heat Rejection (kW) = m x sp ht x TD

where:

m	= mass flow rate of air kg/s
sp ht	= specific heat of air = 1.0 kJ/kg °C
TD	= temperature rise or change across condenser

$$\text{Air volume } V(m^3/s) = \frac{Q \times \text{sp vol}}{\text{sp ht} \times TD}$$

where: sp vol = specific volume of air in m^3/kg

Alternatively

$$\text{Air volume } V(m^3/s) = \frac{Q}{\text{sp ht} \times \text{density} \times \text{TD}}$$

where: density = density of air in kg/m^3

An alternative method would be to select the ventilation fan based on the total air volume of all the condensing units. Air values for the condenser fans of condensing units are available from manufacturer's data.

It is important to remember that condenser fans which form part of condensing units are rated at free air conditions. Due consideration should therefore be given to the layout and type of louvres that are used to introduce air into plant rooms, and also the free area available for fresh air to the condensing unit.

There may be situations that require the use of noise control equipment. Noise consultants should be approached as early as possible if this situation forms a requirement of the contract.

2.11.3 Ventilation requirements for compressors

Where remote air cooled condensers are utilised, the amount of heat dissipated into the plant room by compressors will vary.

As a guide the following information can be used.

Semi hermetic and hermetically sealed compressors are normally suction gas cooled and dissipate small quantities of heat to a plant room. Taking between 5% and 10% of the electrical power input of the motor is appropriate. Heat inputs from oil separator vessels and discharge gas pipework must be added.

Fan cooled, direct drive and belt driven compressors dissipate more heat from the high side of the compressor. Heat is also dissipated from the drive motor. The smaller the motor the greater will be the proportional heat dissipation.

2.11.4 Ventilation fan control

Utilising one air circulation fan with on/off control may lead to unacceptable swings in temperature within the plant room. Not all the equipment will operate at the same time due to load changes and the varying ambient temperatures.

It may be advisable to install a number of air circulating fans and control via a multi-step thermostat or alternatively a speed controlled fan/s.

2.11.5 Leak detection equipment

This should be installed as required in the F-gas directive.

The manufacturer of leak detection equipment should be consulted with regard to the position of sensors. Reference should be made to BS EN 378-3:2016 Section 7 and 8 *Alarms and Detectors*.

2.11.6 Ventilation – escape of refrigerant

Reference should be made to BS EN 378-3:2016 Section 5 *Machinery Rooms*

It should be noted that one air change is equivalent to the plant room volume (assuming the plant room does not contain any equipment etc).

When leakage of refrigerant has occurred and the amount of refrigerant in the atmosphere has reached the maximum exposure level, all the fans should operate. Evacuation of personnel from the plant room to a safe place is required.

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Part 1 Introduction	Oct 2009	Under revision
Part 2 System Design and Component Selection	Jul 2019	2nd edition
Part 3 Safety Regulations, Standards and Directives	May 2009	Under revision
Part 4 System Installation	Oct 2008	Under revision
Part 5 System Commissioning	Oct 2008	Under revision
Part 6 System Maintenance and Service	Apr 2009	Under revision
Part 7 System and Component Decommissioning and Waste Disposal	Dec 2008	Under revision
Part 8 Refrigerants and Retrofitting	Dec 2008	Under revision
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British Refrigeration Association (BRA)
2 Waltham Court, Milley Lane, Hare Hatch, Reading, Berkshire, RG10 9TH United Kingdom
Tel No: + 44 (0)118 940 3416 - Fax No: + 44 (0)118 940 6258
Email: info@feta.co.uk

Institute of Refrigeration (IOR)
Kelvin House, 76 Mill Lane, Carshalton, Surrey SM5 2JR
Tel No: +44(0)2086477033 - Fax No: +44 (0)20 8773 0165
Email: ior@ior.org.uk
Registered charity 1166869

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Published By:

British Refrigeration Association
2 Waltham Court, Milley Lane,
Hare Hatch, Reading
Berkshire, RG10 9TH, UK
Tel No: + 44 (0)118 940 3416
Fax No: + 44 (0)118 940 6258
E-mail: info@feta.co.uk
https://www.feta.co.uk/associations/bra

Institute of Refrigeration
Kelvin House, 76 Mill Lane,
Carshalton, Surrey, SM5 2JR
Tel No: +44(0)2086477033
Fax No: +44 (0)20 8773 0165
E-mail: ior@ior.org.uk
https://ior.org.uk
Charitable Incorporated Organisation no: 1166869

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